TL
1
S6
Vol. 61

SAE TRANSACTIONS

Society of Automotive Engineers, Inc.
29 West 39th Street New York 18, New York

Robert Cass
President

John A. C. Warner
Secretary and General Manager

B. B. Bachman
Treasurer

Norman G. Shidle
Editor

E. D. Allen
Managing Editor

All articles appearing in SAE Transactions are indexed in "Engineering Index."

Copyright 1953, Society of Automotive Engineers, Inc.

Price: $3 to members, $7 to public and college libraries and U. S. Government Agencies, $10 to nonmembers. Foreign: $3 to members, $8 to public and college libraries, $11 to nonmembers.

Photolithographed reproductions can be had of any article published in SAE Transactions. Prices, on quantities of 100 copies or more, varying with number of pages and illustrations, can be obtained upon request from SAE Special Publications Department, 29 West 39th Street, New York 18, N. Y.

THIS volume of SAE Transactions has been numbered 61.

The last volume of the annually issued SAE Transactions was No. 54 (published in 1946).

The intervening years are covered by Vols. 1-6, inclusive, of SAE Quarterly Transactions, which was discontinued with the October 1952 issue.

From now on, SAE Transactions will be issued annually as a single, bound volume.

SPARK-PLUG FOULING STUDIES

R. J. Greenshields, Shell Oil Co.

Paper presented at SAE National Passenger-Car, Body, and Materials Mtg., Detroit, March 4, 1952.

SPARK-PLUG fouling has always been an operational problem in the aviation industry. This problem has been particularly acute in aircraft operating under lean-mixture cruise conditions in attempts to obtain long-range and/or maximum economy. The problem has manifested itself in severe engine roughness requiring premature termination of the flight or resulting in failure at take-off. The nature of this fouling in aviation engines is now believed to be understood, and practical corrective measures have been demonstrated in laboratory engines and significant improvement has been obtained in field tests now in progress.[1]

With the use of high-output, high-compression ratio automotive engines as recently introduced by most of the major automobile manufacturers, a similar spark-plug fouling problem has been found to exist when high-speed or high-power operation is attempted after a period of city-type driving. In some cases the fouling is so severe as to decrease top speed or to prevent proper shifting of the automatic transmissions used in conjunction with certain of these engines. With the trend toward higher

[1] See *SAE Journal*, Vol. 60, May, 1952, pp. 65-73; June, 1952, pp. 37-40: "Aviation Spark-Plug Fouling," by V. E. Yust and E. A. Droegemueller.

THE acute operational problem of spark-plug fouling in both the aviation and automobile industries is discussed in this paper, with special reference to the use of a tricresyl phosphate scavenger mix, which has been found to permit the laboratory aviation engine to run for 62 hr without fouling.

The mechanism by which tcp relieves spark-plug fouling is believed to be the prevention of formation of metallic lead, but additional factors include the formation of lead salts having low electrical conductivity; the reduction in deposition of free carbon, and the production of soft, powdery deposits, which slough off easily, compared with the usual hard, glazed form.

The successful use of tcp, as indicated in both laboratory and field tests, points to a future in which spark-plug fouling may become one of the minor causes of ignition failure.

The Author

R. J. GREENSHIELDS (M '35) is director of the Wood River Research Laboratory of the Shell Oil Co. He joined Shell in 1933 and since then has had extensive experience in the development and testing of automotive and aircraft fuels and lubricants. In 1943 he became chief research engineer and in 1948 assumed his present position. Mr. Greenshields obtained his B.S. degree in mechanical engineering from the University of Illinois in 1932.

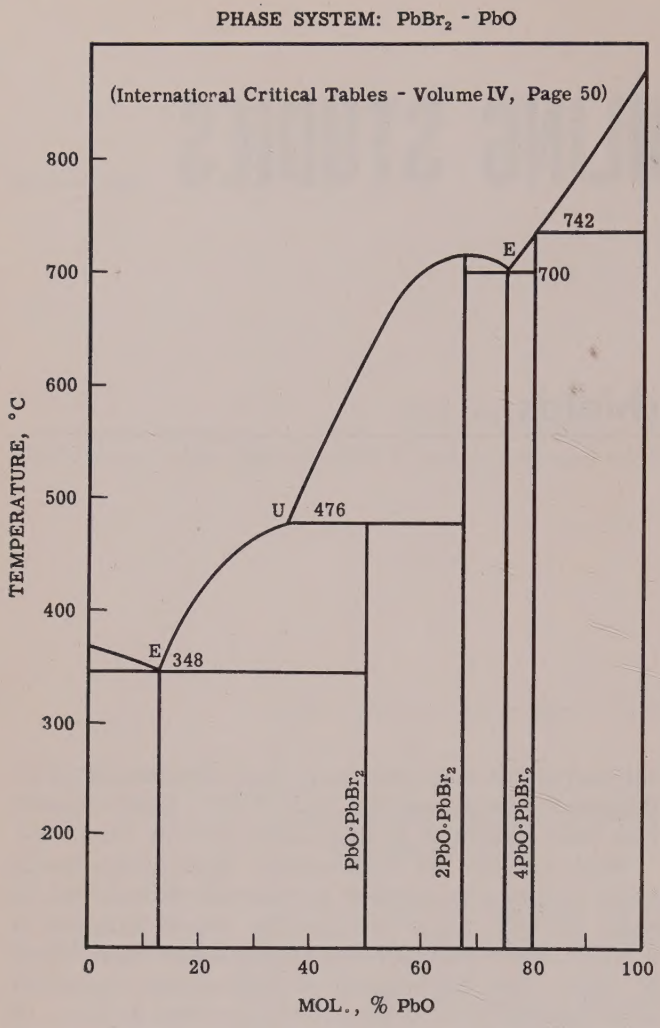


Fig. 1 – Phase diagram for lead bromide-lead oxide system

compression ratio and greater specific output, and the consequent increased demand on ignition system performance, it is expected that spark-plug fouling will soon become a limiting factor.

From our background of experience in the aviation investigation and our studies to date with the automotive spark-plug fouling problem it appears that the nature of the fouling is the same in the two types of equipment and that the mechanism of fouling is quite similar. Although this paper concerns mainly the automotive problem, our extensive examination of fouled and unfouled aviation spark plugs makes it desirable to include part of this work, and the discussion of mechanisms of fouling will contain combined analytical data obtained from both aviation and automotive engines.

The interpretation of reports of fouling from the field has oftentimes been complicated by structural failures of the ceramic which were not evident until the plug had been sectioned for deposit analysis. However, in this discussion, plug fouling is defined as that condition where deposits alone on the firing end of the plug cause sparking failure. Three types of fouling are generally recognized:

1. A deposit bridging of the spark-plug gap whereby a direct short-circuit is created. This may be caused by an increase in operating output which fuses deposits previously laid down, or it may result from a release of deposits from some other part of the engine such as the exhaust valve or piston crown. Spark-plug position, combustion-chamber design, plug gap clearance, and the distribution of tel and scavengers are factors affecting this type of failure. The occurrence of a bridged gap is in general highly erratic and accounts for only a low percentage of spark-plug failures.

2. A second type of failure is commonly referred to as low shunt resistance. This results from deposits on the ceramic core which at elevated temperatures provide an alternative path for the electrical discharge. It is characteristic of this type of failure that the conductivity of the deposit increases with temperature. When the resistance of the deposit falls below a certain critical value (which will vary with ignition systems and engine operating conditions) the potential developed across the plug electrodes is not sufficient to ionize the spark gap.

3. The third type of failure is a surface phenomenon which is manifested either as a corona-type discharge over the whole ceramic surface or as tracking down a narrow path along the ceramic. This type of failure is due to the presence of a highly conductive material distributed discontinuously over the surface. Visual evidences for tracking are not always seen but continued tracking may make a visible path through the deposit and cause a direct electrical short-circuit almost analogous to gap bridging.

The conditions under which spark-plug fouling shows up in automobile operation almost always involves a history of a short period of high output (highway driving) following extensive periods of low output and idling (city driving). This fouling results in poor acceleration, reduced top speed and generally rough idle.

Analysis of Combustion-Chamber and Spark-Plug Deposits

Combustion-chamber and spark-plug deposits have been extensively examined in order to find the reason for spark-plug fouling. The introduction of tel into gasoline to improve antiknock quality of the fuel introduces the possibility of a wide variety of combustion-chamber deposits. In aviation Ethyl fluid a sufficient quantity of ethylene dibromide is present to theoretically convert all of the lead into lead bromide, whereas in automotive mix there is 1.5 times as much halogen as is theoretically required to convert all of the lead into lead halide (1.0 theory ethylene dichloride and 0.5 theory ethylene dibromide). In contrast to aviation fuels which have low sulfur content, automotive gaso-

lines generally contain up to 15 times as much sulfur as is theoretically required to convert all the lead to lead sulfate.

Although lead bromide would be expected to be the primary lead deposit from the combustion of aviation gasoline, and lead halides and lead sulfate would be expected from the combustion of automotive gasolines, these simple compounds are frequently not present at all, or if present make up only a minor proportion of the total lead salts in the combustion chamber. To establish the cause and mechanism of spark-plug fouling it was first necessary to make a comprehensive study of the chemistry of the formation of inorganic lead deposits found in engines operated on leaded gasolines.

Heretofore, the examination of combustion-chamber deposits has normally consisted of a conventional elemental chemical analysis. The composition of the deposits was in turn calculated, assuming the simplest form of the lead salts such as halides, sulfates, and where necessary, oxides. In the subject investigation an X-ray spectrometer was used to attempt to identify the actual chemical phases present in the combustion-chamber deposits. The preliminary results with the X-ray spectrometer indicated that some of the simple compounds of lead previously mentioned were not present, but that X-ray diffraction patterns were frequently obtained which were unlike any of the patterns in the ASTM card index file.

The phase diagram for the lead bromide-lead oxide system as reported in International Critical Tables is shown in Fig. 1. This diagram indicates the presence of three intermediate compounds. Synthetic mixtures of lead bromide and lead oxide, of the appropriate compositions according to the phase diagram, were prepared and fused together. The X-ray diffraction patterns of these synthetic mixtures showed that four intermediate compounds $PbO \cdot PbBr_2$, $2PbO \cdot PbBr_2$, $4PbO \cdot PbBr_2$ and $6PbO \cdot PbBr_2$ were formed. The X-ray diffraction patterns of these synthetic compounds were in turn used in identifying the aviation engine deposits. The phase diagram for the lead sulfate-lead oxide system also taken from International Critical Tables and given in Fig. 2 shows the presence of three intermediate compounds. Again synthetic mixtures were prepared by fusing together appropriate amounts of lead sulfate and lead oxide, and obtaining X-ray diffraction patterns. The X-ray data showed that the three intermediate compounds were $PbO \cdot PbSO_4$, $2PbO \cdot PbSO_4$ and $4PbO \cdot PbSO_4$. The X-ray diffraction patterns of these synthetic samples were used in the identification of combustion-chamber deposits from automotive engines.

The X-ray diffraction analysis of deposits obtained in typical aviation and automotive engines for both intermediate and higher output operation are summarized in Table 1. It should be noted that the occurrence of a pure compound is the exception rather than the rule, and that the simultaneous presence of several chemical phases complicates interpretation of the analytical data. The lead oxybromides are the major constituents of deposits

Table 1 – Typical X-Ray Diffraction Analyses of Deposits from Several Engine Locations
(Major constituents listed first)

	Aviation Engine	Automotive Engine
Induction System	$PbBr_2$	PbX_2[a]
Intake Valves,		
Intermediate Output	$2PbO \cdot PbBr_2 + PbO \cdot PbBr_2$	$PbSO_4 + PbX_2$
High Output	$2PbO \cdot PbBr_2$	$PbSO_4 + PbO \cdot PbSO_4$
Piston Crown,		
Intermediate Output	$PbO \cdot PbBr_2 + 2PbO \cdot PbBr_2$	$PbSO_4 + PbX_2$
High Output	$2PbO \cdot PbBr_2$	$PbSO_4$
Spark-Plug Cavity,		
Intermediate Output	$2PbO \cdot PbBr_2 + PbO \cdot PbBr_2$	$PbX_2 + PbSO_4$
High Output	$2PbO \cdot PbBr_2$	$PbX_2 + PbSO_4$
Spark-Plug Ceramic Tip,		
Intermediate Output	$2PbO \cdot PbBr_2$	$PbSO_4$
High Output	$6PbO \cdot PbBr_2 + 2PbO \cdot PbBr_2$	$PbSO_4 + PbO \cdot PbSO_4$
Exhaust Valves,		
Intermediate Output	$2PbO \cdot PbBr_2 + PbO \cdot PbSO_4$	$PbO \cdot PbSO_4 + PbSO_4$
High Output	$4PbO \cdot PbSO_4 + 2PbO \cdot PbBr_2$	$4PbO \cdot PbSO_4 + PbO \cdot PbSO_4$

[a] PbX_2 represents a solid solution of lead chloride and lead bromide, with a chlorine to bromine ratio of approximately 2/1, as in the fuel.
Note: Lead scavenger was 1.0 theory ethylene dibromide in aviation engine; 0.5 theory ethylene dibromide plus 1.0 theory ethylene dichloride used in automotive engine.

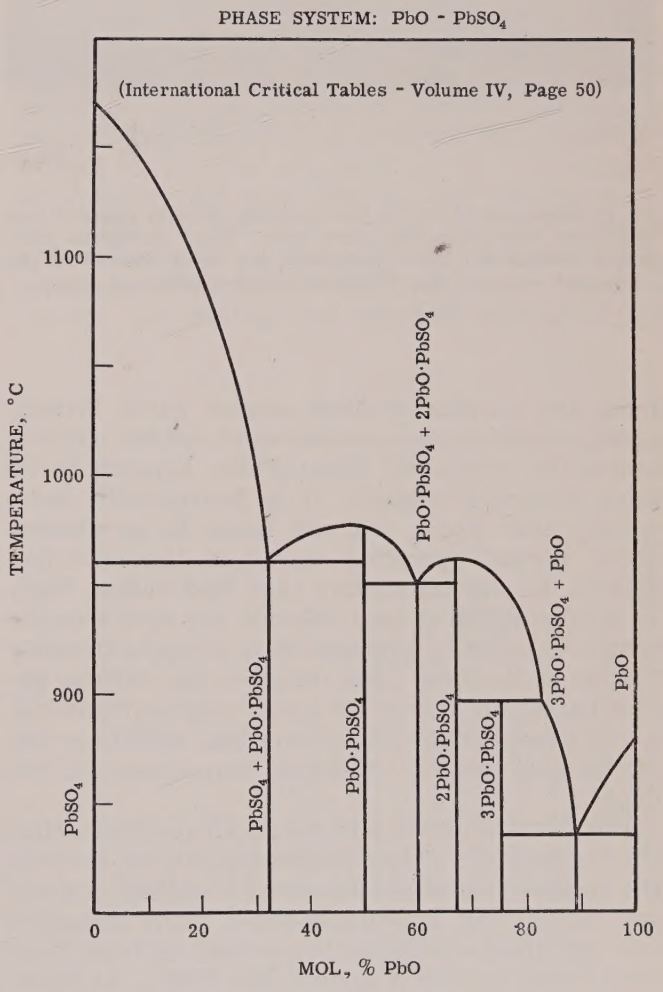


Fig. 2 – Phase diagram for lead sulfate-lead oxide system

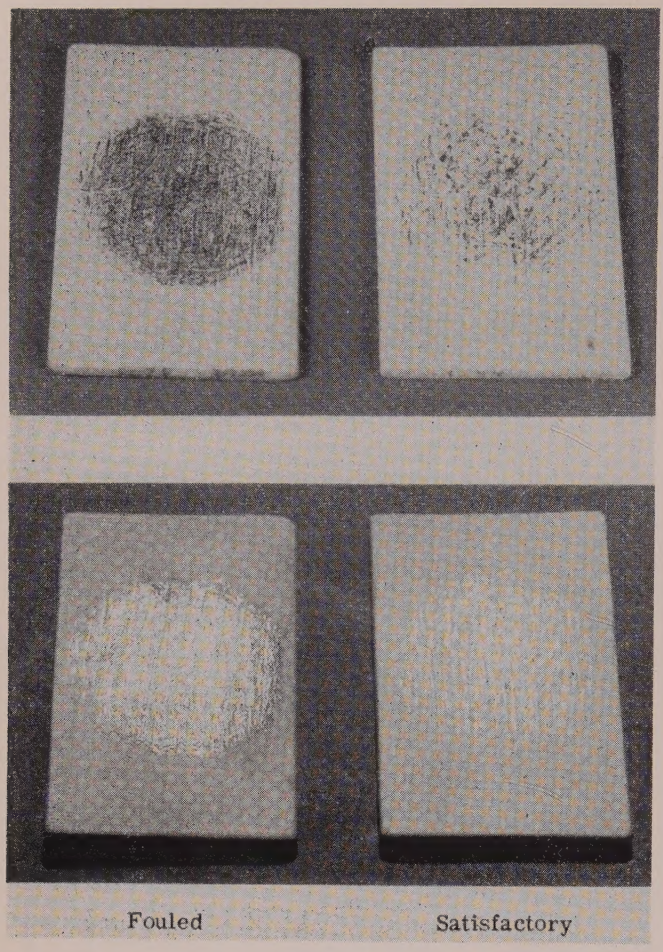


Fig. 3 – Appearance of metallic lead specimens physically separated from fouled and from satisfactory spark plugs. Upper photograph under general illumination. Lower photograph with lights adjusted to give specular reflection from freshly burnished metallic lead surface

from the various aviation engine parts investigated, except in the exhaust valve. Of the oxybromides the compound $2PbO \cdot PbBr_2$ appears to be most generally present. It is particularly noteworthy that $PbBr_2$ was not found to be present under normal operation except in the induction system. In the automotive case lead halide, PbX_2 (a solid solution of lead chloride and lead bromide with a chlorine to bromine ratio of approximately 2/1, as in the fuel) does occur on the various engine parts, but only in the spark-plug cavity was it a major constituent. Elsewhere lead sulfate or the oxysulfates are the principal components of the deposit.

Further examination of the results indicates that the temperature of the condensing surface controls the composition of the deposits for a given scavenger content. In aviation engines under extended idle operating conditions the coolest surfaces have been found to collect appreciable $PbBr_2$. At intermediate temperatures the lead oxybromides are deposited. The ratio of lead oxide to lead bromide increases with the temperature of the surface. At the high temperatures at which exhaust valves normally operate lead oxysulfates begin to appear, while at abnormally high temperatures lead oxysulfates alone are able to persist. The same trends also hold for the automotive engine deposits. In this case, however, due to the comparatively high sulfur content of the fuel, the lead halides are largely replaced by lead sulfate. Lead sulfate in turn is converted into $PbO \cdot PbSO_4$ and $4PbO \cdot PbSO_4$ by loss of SO_3 on the hottest engine surfaces or by reaction with oxyhalides in the solid state.

Mechanism of Spark-Plug Fouling

The deposits from both fouled and satisfactory spark plugs were examined for possible differences which could explain the fouling. The major constituent of the aviation spark-plug deposits was found to be $2PbO \cdot PbBr_2$ regardless of whether or not the spark plug was fouled. $PbO \cdot PbBr_2$ was also present in both cases though in smaller concentrations. Under high-output operation the compound $6PbO \cdot PbBr_2$ was observed to be present on both the fouled and the satisfactory spark plugs. Thus these lead oxybromides did not appear to be responsible for fouling of the spark plugs. Furthermore, the weight of the deposits found on the plug was no indication of plug performance.

In view of the similar X-ray analysis results on the desposits from fouled and satisfactory plugs, and also the lack of correlation between the quantity of deposit and fouling, the deposits were subjected to additional physical and chemical analysis. Laboratory-fouled plugs exhibited one unusual physical characteristic, namely, the presence of a grey-black glazed deposit on the nose of the plug. Subsequent work showed that these deposits contained metallic lead. A special procedure was devised to check for the presence of metallic lead since the X-ray spectrometer is not satisfactory for detection of low concentrations of metallic lead in the presence of much larger amounts of lead salts. This special procedure consisted of grinding the deposits under toluene on a clean, unglazed porcelain surface. Minute particles of lead, if present, are caught in the pores of the porcelain. The ceramic plate was in turn examined under a microscope for material of metallic luster. The larger particles were confirmed to be metallic lead on the basis of malleability, metallic conductance, melting point, and the preparation of derivatives. The results of analysis on deposits from typical fouled and satisfactory laboratory aviation plugs are shown in Table 2. The test for metallic lead was positive on fouled plugs and negative on the satisfactory plugs. Thus the presence of metallic lead appeared to correlate with fouling. Further, on the fouled plugs metallic lead was generally present in a higher concentration on the end of the ceramic than on the side of the ceramic core, while the

Table 2 – Composition of Aviation Spark-Plug Deposits
(Laboratory test plugs – Conventional scavenger)

Operating Time, hr	Fouled Plug 4	Satisfactory Plug 24
Quantity of Deposit	Light	Heavy
Appearance – Nose Deposit	Gray-black Some globules	Yellow-gray No globules
Metallic Lead Test	Positive	Negative
X-Ray Diffraction Analyses:		
Ceramic Core Tip		
Major Constituent	$2PbO \cdot PbBr_2$	$2PbO \cdot PbBr_2$
Minor Constituent	$PbO \cdot PbBr_2$	$PbO \cdot PbBr_2$
Side of Ceramic Core		
Major Constituent	$2PbO \cdot PbBr_2$	$2PbO \cdot PbBr_2$
Minor Constituent	$PbO \cdot PbBr_2$	$PbO \cdot PbBr_2$

Table 3 – Composition of Automotive Spark-Plug Deposits
(Laboratory test plugs – Conventional scavenger)

Operating Time, hr	Fouled Plug 71	Satisfactory Plug 71
Quantity of Deposit	Medium	Medium
Appearance – Nose Deposit	Yellow-gray Some metallic smears	Gray-white No metallic smears
Metallic Lead Test	Positive	Trace
X-Ray Diffraction Analyses:		
Ceramic Core Tip		
Major Constituent	$PbSO_4$	$PbSO_4$
Minor Constituent	$PbO \cdot PbSO_4$	$PbO \cdot PbSO_4$
Side of Ceramic Core		
Major Constituent	$Pb(X)_2$	$Pb(X)_2$
Minor Constituent	$PbSO_4$	$PbSO_4$

occurrence of metallic lead near the bottom of the plug cavity was erratic.

Deposits of spark plugs from large multicylinder aircraft engines troubled with spark-plug fouling in the field have been examined. As with the laboratory findings there were no significant differences in the nature or amount of lead salts from fouled and satisfactory plugs – lead oxybromides constituted the major portion of the deposits in both instances. In all cases the test for metallic lead was also made, and although metallic lead was frequently found on the satisfactory plugs, fouled plugs from comparable operation always exhibited much greater quantities. Fig. 3 is a photograph showing the relative amounts of lead in composite samples taken from a number of fouled and satisfactory spark plugs from the same engine. It is not too surprising that traces of metallic lead were found on spark plugs which were operating satisfactorily, since these plugs, having already operated as long as those which were definitely fouled, might readily be expected to be in the process of fouling. Furthermore, fouling would depend on the manner of distribution of the metallic lead as well as on the total quantity present.

A large number of aviation spark-plug deposits from the field were analyzed chemically for metallic lead in an attempt to correlate the amount of metallic lead in the deposit with the behavior of the plug in the engine. The data are summarized in Fig. 4. No critical amount of metallic lead marking the division between fouled and satisfactory plugs was found, but the data do indicate a definite trend. Thus, as high as 1.7% metallic lead content was found on plugs which were still operating but no deposit from a fouled plug contained less than 1% metallic lead. These data indicate that fouling depends not only on the amount of metallic lead present but that particle size and distribution are equally important. For example, it is obvious that a globule of metallic lead of appreciable size, located at the side or base of the ceramic in a noncritical area, would not of itself cause the spark plug to malfunction.

Trends similar to those found for the aviation plugs were also observed for automotive spark-plug deposits, as will be seen from the analytical data given in Table 3. Here again the bulk of the lead salt deposits were identical on the fouled and satisfactory plugs – the deposits consisting essentially of high-melting lead sulfate and lead oxysulfate of very low electrical conductance. A difference between the fouled and satisfactory plugs did exist in the metallic lead contents. In view of the analytical data on both the aviation and automotive spark-plug deposits, it is considered that the nature of fouling is essentially the same in the two cases.

The mechanism by which metallic lead is formed on the spark plug, subsequently causing misfire, is postulated to be as shown diagrammatically in Fig. 5. This mechanism involves (1) deposition of lead salts on the plug ceramic due to incomplete scavenging; (2) admixture of carbonaceous material with the lead salts during periods of idle or rich mixture, or extended operation at such lean mixtures (low output) that incomplete combustion occurs; (3) heating the resultant mixture by in-

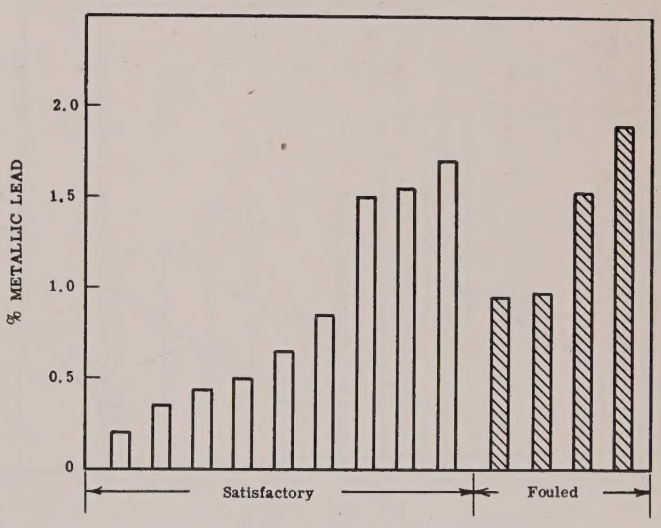


Fig. 4 – Metallic lead contents of deposits from fouled versus satisfactory spark plugs

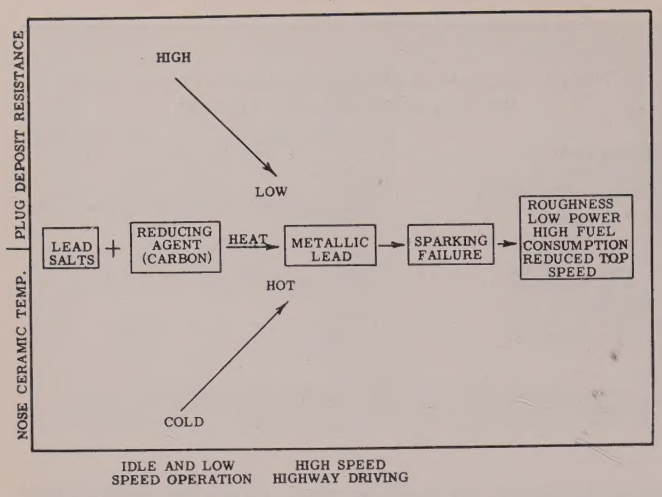


Fig. 5 – Postulated mechanism of automotive plug fouling

creased output operation, as characterized by high-speed highway driving or hill-climbing (take-off or climb conditions in the aviation application) with the result that a certain proportion of the deposit is reduced to metallic lead; (4) misfiring of the spark plug, due to a perhaps discontinuous but excellent electrically conducting path offered by the metallic lead particles.

The above mechanism, which fixes metallic lead as the primary cause of spark-plug fouling of the type currently experienced, satisfactorily explains all of the operational experiences and analytical data on both aviation and automotive spark-plug deposits. For example, this mechanism will explain

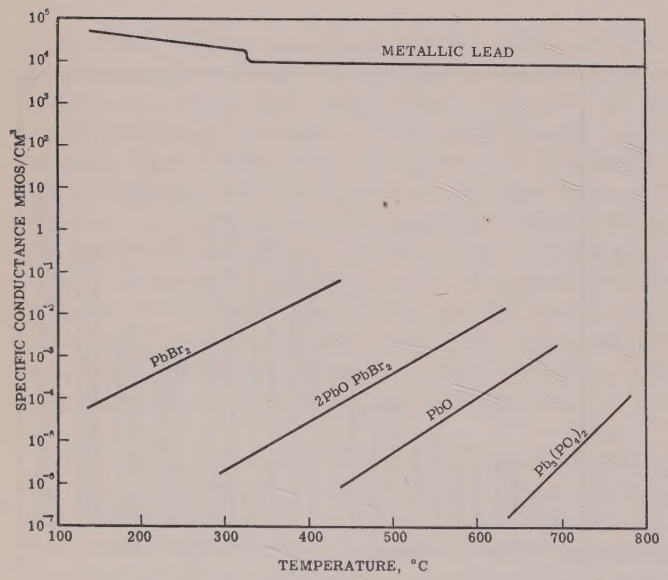


Fig. 6 – Electrical conductivity of metallic lead and certain lead salts

the greater incidence of fouling in those cars operated for a high percentage of the time under city driving conditions. It also explains the marked reduction in fouling time with increased tel content of the fuel.

In addition, experiments designed specifically to test the hypothesis have without exception supported the metallic lead theory. Several tests were run in the laboratory aviation engine with a view toward establishing the effects of carbon on fouling time. The addition of 1% by weight of colloidal graphite to leaded fuel (4.0 cc tel per gal) was found to decrease spark-plug fouling time from 4 to 2.75 hr. Appreciable amounts of metallic lead were visible in the deposit but little evidence of residual graphite was found on the plugs. To accelerate the formation of carbonaceous deposits within the engine itself, additional tests were conducted employing 5 min of rich mixture, low-output operation during each half-hour of engine time. This resulted in a fouling time of 1 hr compared to 4 hr when using continuous simulated cruise. Fouling always occurred within 1 min after return from idle to cruise output. Examination of the plug both visually (microscopically) and analytically indicated the presence of copious quantities of metallic lead globules, with no evidence of excessive carbonaceous deposits. On the other hand, when the plugs were removed for examination at the end of the rich mixture cycle, they were found to be sooty in appearance and the deposits contained only traces of metallic lead. It must be concluded that the carbon, which was in greatest proportion in the deposit just at the end of the rich idling period, was insufficient by itself to cause fouling but led to fouling by its ability to reduce lead salts to metallic lead on heating of the mixed deposits during subsequent medium to high output operation.

An alternate mechanism which has been proposed is that fouling is due to lowered electrical shunt resistance of the lead salt deposits at elevated temperatures. In this connection the electrical conductivities of several typical lead salts and metallic lead are given in Fig. 6. It is apparent that metallic lead is more conductive by several orders of magnitude than the most conductive of the lead salts. That electrical conduction of the lead salts themselves is not the mechanism involved in the current fouling problem, is further indicated by the analytical data previously presented which shows that identically the same lead salts are found on the fouled as on the satisfactory plugs from the same engine, and indeed cases have been found where a 4-fold increase in the usual amount of the lead salt deposit of the same composition failed to produce fouling.

Development of New Lead Scavenger

Accepting the metallic lead theory as the cause of spark-plug fouling, several methods of correct-

ing the difficulty are available: (1) prevent as far as possible by proper scavenging, manifold distribution, and lowered lead content of the fuel, the accumulation of reducible lead salts on the spark-plug ceramic; (2) prevent or minimize by increased head temperatures, appropriate fuel/air ratios and reduced idling time, the amount of carbon deposited with the lead salts. This may also be accomplished by admixture of an additive in the fuel which will promote oxidation of the carbon before reduction of lead salts occurs; (3) convert the residual unscavenged lead salts to compounds more difficult to reduce to metallic lead; (4) combine the metallic lead into a compound which is not harmful to spark-plug functioning.

Recognizing that the conventional halide scavengers remove the bulk of the lead from the combustion chamber,[2] the investigation of lead scavengers was centered around getting rid of the remainder of the lead oxidation products or rendering them relatively harmless. In other words, efforts were made to supplement the action of the normal halide scavengers. As a first attempt to prevent the accumulation of reducible lead salts on the spark-plug ceramic in the laboratory aviation engine, additional ethylene dibromide (0.2 theory) was supplied in the fuel so that the total scavenger content was 1.2 theories. This, however, produced only a small increase in the time for fouling. Hundreds of compounds, including materials which were synthesized specifically for the program, were run in various combinations and concentrations to determine their effect on spark-plug fouling time. The use of phosphorus compounds was studied in considerable detail. Tricresyl phosphate (hereafter referred to as tcp) was extremely effective in minimizing spark-plug fouling. Whereas the conventional scavenger gave a spark-plug fouling time of about 4 hr in the laboratory aviation engine, the special tcp scavenger mix permitted satisfactory operation for 62 hr with no evidence of fouling. The same supplemental tcp scavenger added to motor gasoline was also found to effectively reduce the occurrence of automotive spark-plug fouling.

The mechanism by which tcp relieves spark-plug fouling is believed to be the prevention of formation of metallic lead but additional factors may be recognized:

1. The formation and deposition of metallic lead is minimized. Physical analyses of spark-plug deposits from tcp fuel have revealed little or no metallic lead to be present. Further evidence that tcp minimizes metallic lead came from tests on plugs previously fouled which were cleaned of all deposits except those on the tip of the center electrode ceramic, leaving a deposit that runs high in metallic lead content. The quantity of metallic lead present was found to be decreased substantially by the use of tcp fuel on the basis of change in deposit weight and deposit analysis.

2. Lead salts having low electrical conductivity are formed. The use of tcp in the fuel results in

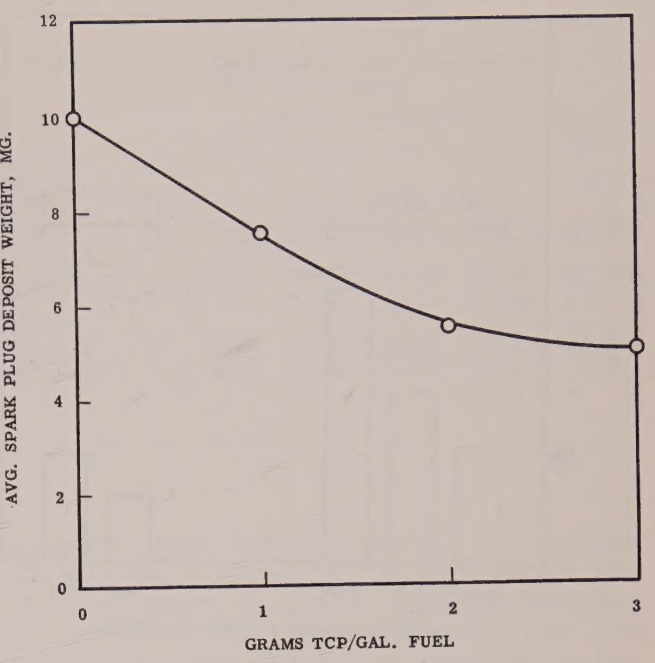


Fig. 7 – Effect of tcp on spark-plug carbon deposits (unleaded fuel)

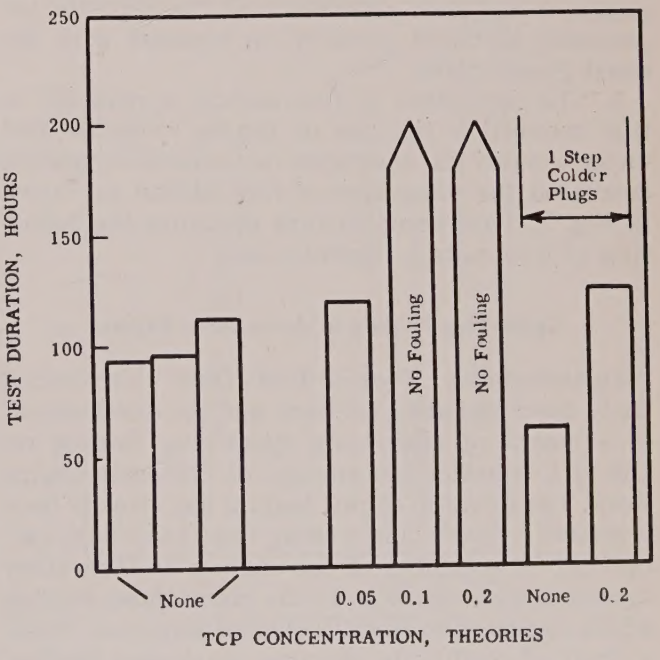


Fig. 8 – Effect of tcp on spark-plug fouling in engine A, a 1949 V-8 overhead-valve type

[2] See ASTM Bulletin No. 154, October, 1948, pp. 53-56: "Better Lead Scavenging Needed for Aviation Gasolines," by E. A. Droegemueller.

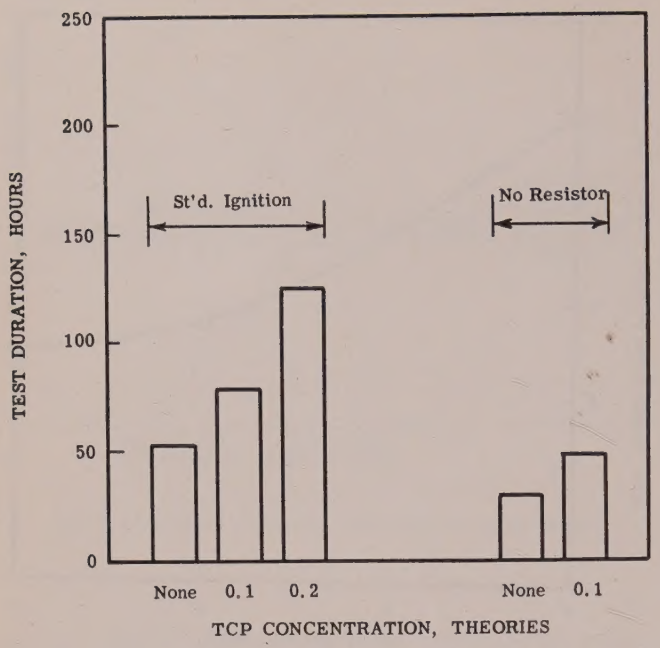


Fig. 9 – Effect of tcp on spark-plug fouling in engine B, a 1951 V-8 overhead-valve type

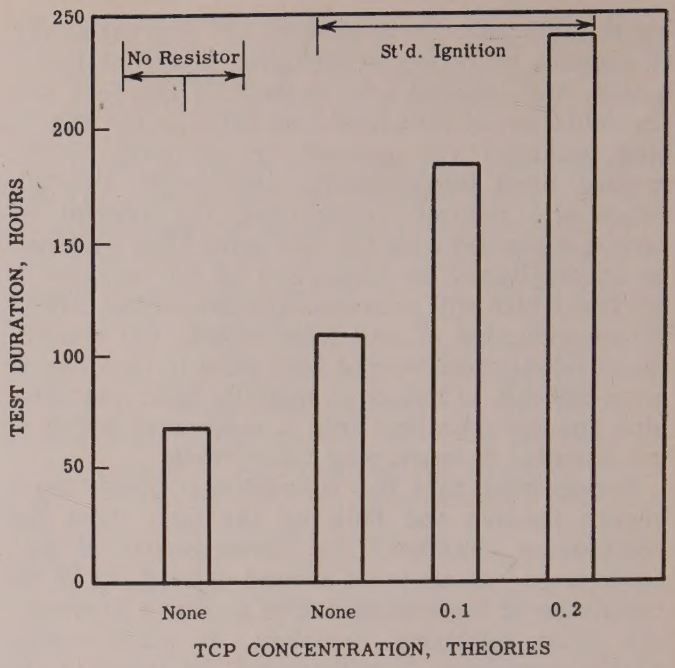


Fig. 10 – Effect of tcp on spark-plug fouling in engine C, a 1951 inline-8 overhead-valve type

the formation of lead phosphates and lead complexes such as lead oxyphosphate and lead halophosphate which have low electrical conductivity. The conductances of certain of these materials are given in Fig. 6.

3. Deposits slough off readily. The use of tcp has a physical effect on combustion-chamber deposits. It has been observed that the deposits are generally soft and powdery, in contrast with the usual glazed form.

4. The deposition of free carbon is reduced. In this connection the use of tcp in unleaded fuel under conditions conducive to carbon formation decreased the deposition of free carbon as shown in Fig. 7. This would in turn minimize the reduction of lead salts to metallic lead.

Spark-Plug Fouling in Automotive Engines

Dynamometer Engine Test Data – The previously described analysis work and the development of a means of alleviating spark-plug fouling required a considerable amount of full-scale engine work. The aviation engine testing has already been described.[1] Spark-plug fouling tests have been carried out in a number of car engines on laboratory dynamometer test beds. In the main, those engines which appeared to be critical were employed. After a study of conditions affecting spark-plug fouling, it was apparent that cyclic operation of the engine was necessary and that idle periods followed by light-load operation would bring about fouling of the spark plugs in a relatively short time with reasonably consistent and valid results. The pertinent test conditions used on four engines are shown in Table 4. It will be noted that a cycle consisting of 1 min of idling followed by 9 min at approximately 30 mph level road load, was used for engines A, B and C with spark-plug fouling occurring in about 100 hr or less. For engine D, a 6-cyl L-head design, similar cycling and loading did not produce fouling at 200 hr and conditions were changed to a longer idle period and a shorter, very low output period. Under these conditions spark-plug fouling was obtained in a reasonable time. As has previously been discussed, the principal type of spark-plug fouling dealt with in this paper is controlled materially by engine operating conditions and, in particular, by air/fuel ratio, low-power output and low-operating temperatures followed by periods of high power and high temperatures.

Spark-plug fouling was determined by operating the engine at full throttle 3200 rpm and noting engine roughness, power loss and, in several cases, backfiring. These checks were made at 8-hr intervals, and then more frequently as the test reached the critical point.

The data obtained are shown in graphical form in Figs. 8, 9, 10, and 11. Considering engine A, a V-8 overhead-valve design, spark-plug fouling occurred in 94, 95 and 112 hr, using a commercially available motor gasoline containing 1.5 ml tel per

gal. This is illustrative of the reproducibility to be expected in these tests.

There was little effect for the tcp concentration of 0.05 theory; however, no fouling occurred at 200 hr with 0.1 and 0.2 theories of tcp. These results are very similar to those obtained in aviation engine work at low concentrations where it was found that 0.2 theory of tcp was optimum. It should be mentioned that the results of the concentration studies discussed here are for fuels containing 1.5 ml tel per gal and may not be applicable at widely different tel contents.

In this particular engine the performance of spark plugs one step colder than normal was investigated. As was to be expected, the plugs became fouled in about half the time when using the colder plug. The use of 0.2 theory tcp in the fuel doubled the life of the spark plugs under these conditions.

Considering the data on engines B and C, shown in Figs. 9 and 10, similar trends are in evidence. The results shown in these figures are, in many cases, the average of several tests. Engine B, a V-8 overhead valve, fouled plugs on the normal premium gasoline at 52 hr. The use of 0.2 theory tcp increased the satisfactory operation time of the plugs to 126 hr. With engine C, an in-line 8-cyl overhead valve, fouling occurred without tcp at 110 hr and, with 0.2 theory, this was increased to 240 hr.

Both of these engines came equipped with nominal 10,000 ohm resistors in the secondary of the ignition circuits. Removing these resistors materially reduced spark-plug fouling time in both engines. Thus it appears that the use of a resistance in the secondary circuit will extend spark-plug life somewhat.

Based on the laboratory cycling tests, a fair estimate of the value of tcp as a supplementary scavenger can be obtained. Since the spark-plug fouling time in any engine varied with the type of operation used, direct averaging of the results may not be meaningful. However, under the conditions employed it can be said that the use of 0.1 theory tcp increased the spark-plug life about 75%, and 0.2 theory tcp at least doubled the satisfactory operation period of the spark plugs.

In the course of the above work the ability of tcp to clean up fouled spark plugs was investigated. Engines which had fouled plugs after running on standard premium gasoline were continued on test with various concentrations of tcp being added. It was found that 0.1 theory had no effect. On the other hand, with spark plugs which were not completely shorted out changing to a fuel containing 0.2 theory tcp would, in most cases, bring the partially fouled plugs back to normal performance.

Performance of Tcp in Service – Before beginning any field testing with tcp in either aviation or in automotive fuel, various physical and chemical tests were run to assure that the performance with respect to antiknock quality, stability, corrosion and rubber-swelling characteristics would not create new problems. Antiknock quality of the fuel with as much as 0.2 theory tcp was not impaired in the ASTM Aviation and Supercharge Methods nor in the ASTM Research and Motor Methods. Further, tcp has essentially no effect on fuel inspection properties except for the apparent gum values. The gum values by the present test methods are increased because of the low volatility of the additive though the increase can be accounted for by the quantity of tcp added to the fuel. It may be of interest that induction systems were, in general,

Table 4 – Operating Conditions for Spark-Plug Fouling Tests in Automotive Engines

	Simulated City Operation				Low Power
	Engine A	Engine B	Engine C	Engine D	Engine D
Test Cycle					
Idle, min	1	1	1	1	2
Power, min	9	9	9	9	8
Idle Conditions					
Speed, rpm	500–600	500–600	500–600	500–600	500–600
Manifold Vacuum, in. Hg	19–20	19–20	18–19	19–20	19–20
Power Conditions					
Speed, rpm	1500–1600	1500–1600	1500–1600	1500–1600	1500–1600
Manifold Vacuum, in. Hg	16.5–17.5	17–19	16–17	14–16	18–19
Bhp	12–13	12–13	12–13	9.5–10.5	4–4.5
Temperature					
Coolant, Out, F	135–140	135–140	135–140	145–150	120–130
Oil Sump, F	155–165	170–180	170–180	170–180	140–150
Carburetor Air, F	80–100	80–100	80–100	80–100	80–100

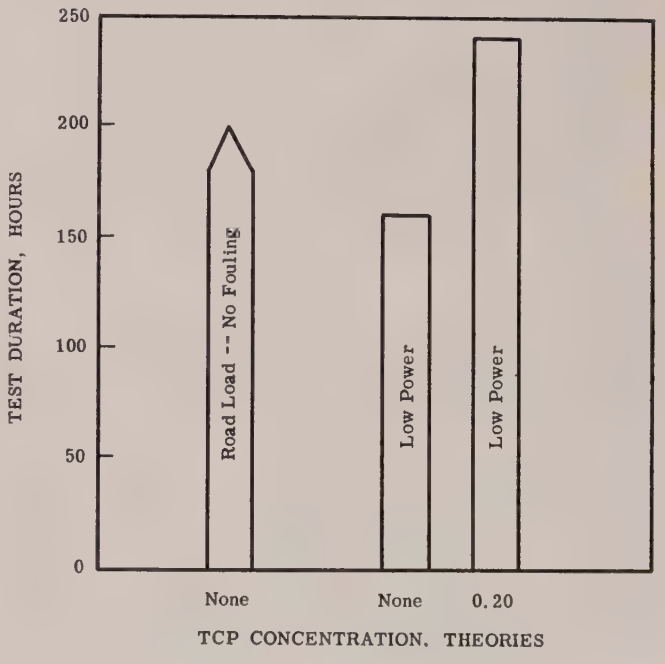


Fig. 11 – Effect of tcp on spark-plug fouling in engine D, a 1949 6-cyl L-head type

Fig. 12 – Spark plugs after 20,000 miles of operation on fuel with regular scavenger plus 0.1 theory tcp

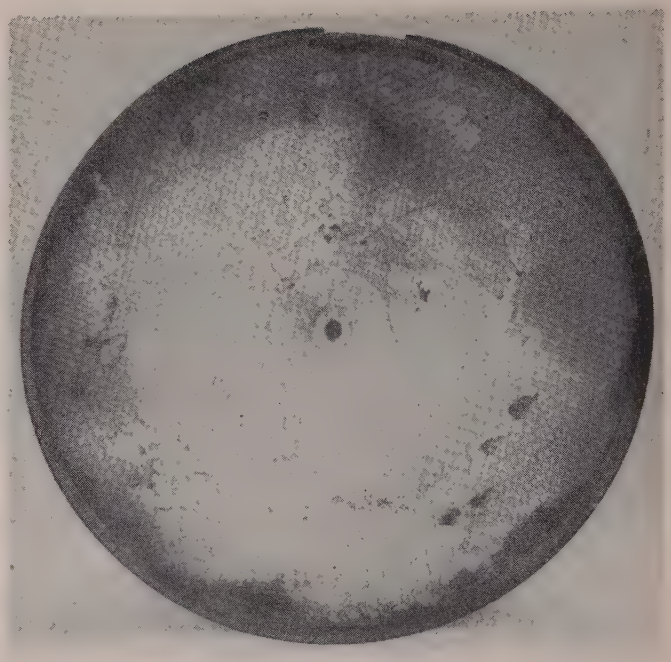


Fig. 13 – Piston crown deposits after 15,000 miles of operation on fuel with regular scavenger plus 0.2 theory tcp

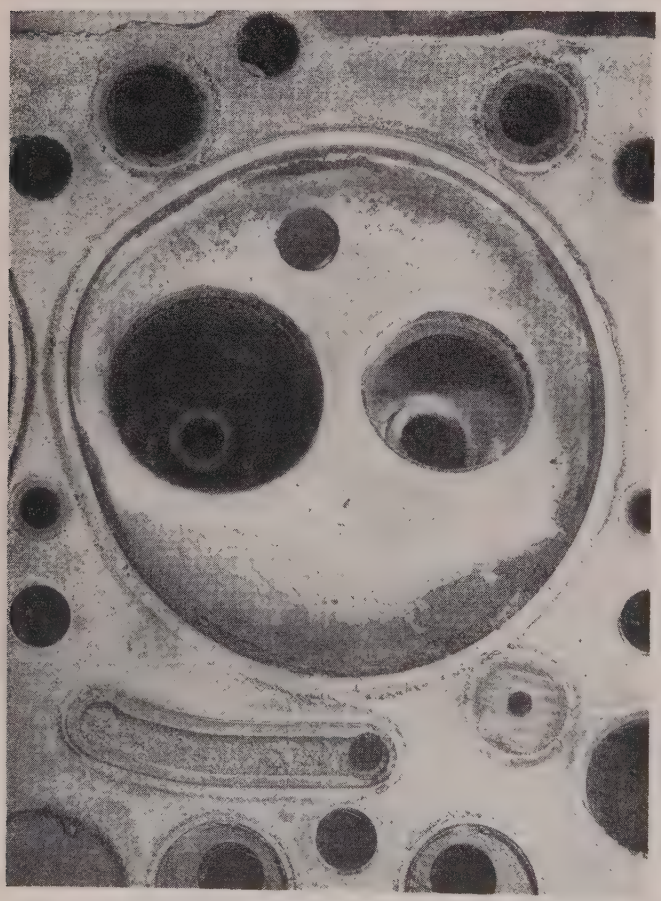


Fig. 14 – Cylinder head deposits after 15,000 miles of operation on fuel with regular scavenger plus 0.2 theory tcp

cleaner and that lacquering of various engine parts was somewhat decreased when fuel containing tcp was used. With regard to corrosion and rubber swelling, no deleterious effects could be found.

To investigate the overall performance of tcp in automotive engines on the road, eight passenger cars were put on an accelerated service test. These cars were all late models and included those engines which would be expected to give spark-plug trouble. A route of 150 miles in length in Central Illinois was chosen. The operation of the vehicles was designed to be normal for cross-country driving since the route included the usual number of towns and open stretches of highway. On the runs the cars averaged 50 to 55 mph for the entire distance but did not go above a 75-mph arbitrary limit. Three of the vehicles were operated on premium gasoline containing 0.2 theory tcp and each was driven 300 miles per day. Two of these have completed 15,000 miles each with no spark-plug changes. Testing on the third was discontinued at 10,000 miles because of a broken piston. The other five passenger cars have each been operated 600 miles per day on the above mentioned route on a premium fuel containing 0.1 theory tcp. This group of five cars has completed 20,000 miles each. Three have experienced no spark-plug difficulty. One had spark-plug gap increases of from 0.009 to 0.014 of an in. at 17,700 miles. These plugs were regapped and continued satisfactorily to the 20,000-mile mark. Fig. 12 is a photograph of a representative plug from this engine showing its exceptionally clean condition. The spark plugs from the other

engines inspected likewise had very little deposit on the ceramic or in the cavity. It may be well to point out here that, especially in aviation work, the quantity of deposit was in no way related to spark-plug fouling. Plugs have been obtained from service testing of tcp where the cavity was partly filled but the performance was entirely satisfactory. On the other hand, spark plugs from engines using normal leaded fuels were found to be fouled, but there was hardly any visible deposit on the ceramic of the center electrode. The fifth car experienced spark-plug failure at 17,600 miles. Engine misfiring occurred at 65 mph. The gap on the plugs was normal and no visible mechanical failure of the ceramics was apparent. The exact cause of the failure has not yet been ascertained.

Fleet tests of vehicles in normal service are underway in several different locations. A total of 25 passenger cars have accumulated 95,000 miles. A group of 46 trucks in local haul service have accumulated 125,000 miles. Six clipper trucks in long haul service have accumulated 186,000 miles. To date there have been no plug changes which could be attributed to fouling.

Inspection of the engines which have been on accelerated tests to date has shown that the quantity of combustion-chamber deposits was about normal. Figs. 13 and 14 show the piston crown and a portion of the cylinder head of one of the engines which had completed 15,000 miles. Here it will be seen that the combustion chamber is uniformly covered with a white deposit with a material absence of the "corn flake" type of buildup which is usually found. The uniformness and physical character of these combustion-chamber deposits may be such that they suppress or prevent preignition caused from this source. Preignition was not noticed by any of the drivers of the cars on the accelerated tests; however, preignition had been reported on a number of these cars when in normal service before the tests began. This same phenomenon was noted in laboratory engines wherein preignition was recorded on several engines running on regular leaded fuel. The preignition disappeared after changing to fuel containing tcp, and again returned when changed to regular fuel.

The octane requirement increase for the engines involved in the accelerated tests was checked periodically and was considered normal, being comparable with other data in hand on previous operation of these vehicles.

From the limited field tests to date it may be concluded that tricresyl phosphate is alleviating the spark-plug fouling difficulty and it would seem that the laboratory tests reported above, wherein tcp gave about 100% increase in spark-plug life, are being borne out in service. Thus it may be that either 0.1 or 0.2 theory tcp will extend the service life of a spark plug from the standpoint of fouling, so that other factors may become the predominating causes of ignition failure.

DISCUSSION

Spark-Plug Fouling Cause of Air Disasters

—S. M. Udale
Holley Carburetor Co.

THE absolute necessity for the use of lead in spark plugs has now been removed because the water injection system is available. In the air, water is used to raise the airplane horsepower up to 3500.

The simultaneous use of lead and water injections can lead to major catastrophes. For example, assume a plane is taking off, or assume a plane is landing, and the pilot changes his mind and opens up the throttle suddenly. In both cases the extra power which is made available is dependent on water. If the water should fail, even slightly, temperatures may go over 621 F, the critical temperature at which lead melts. In that event there is a possibility of a sudden shorting of the plugs.

The second crash of the three at Elizabeth, N. J., the crash at La Guardia airport in New York, where the plane landed in the water, and the Mill Hill accident in England — all can be attributed to lead glazing of the plugs.

That the Mill Hill accident was definitely due to lead glazing was indicated by the full reports of the court findings which were published following investigation of the disaster. The following is a quotation from the magazine *Aeroplane:*

"In its list of conclusions and findings the court points out that the primary contribution to the accident was the failure of the starboard engine shortly after take-off. This failure could not have been foreseen or guarded against by any greater exercise of vigilance that could reasonably have been required of the operators and their staff," the report adds.

"Secondary causes were harder to assess, the report says, but among them are or may be . . . glazing of lead deposits on the sparking plugs on the port engine while under conditions of high power output leading to ignition trouble and so to a loss of power sufficient to destroy the aircraft's ability to maintain level flight.

"Amplifying its findings on the sparking plugs, the report says that the cleaning process to which they were subjected was sandblasting with the plug in an assembled state. This had not removed the deposit from the insulators except at the tip. Erosion of the electrodes caused by excessive sandblasting was observed on some of the plugs and in a few cases to such an extent as to render them unfit for further service. There was some evidence that up to six plugs on the port engine were not gas tight at the gland. The court was of the opinion that the condition of the plugs in the port engine was likely to cause the failure of the aircraft to maintain height whatever engine revolutions and boost were selected. It recommended an alteration of the technique of cleaning plugs."

In the event it is decided that it is not wise to use lead with water injections, there will be no interference with the payload, as the plane will still be able to get off the ground with the full load by use of the water injection system. However, the cruising speed will be reduced. For example, if it now takes four hours to fly from New York to Miami, it may take seven hours without using lead.

It is my thought that not enough attention has been given to the advisability of not using lead with the water injection system on the larger transport planes. A load of 30,000 tons of lead was recently obtained from Mr. Winston Churchill. The Atomic Energy Commission needs all the lead we can save by not using it to pollute the air.

Stress Testing

The Author

H. C. JOHNSON (M '47) is project engineer engaged in the testing and development of automobile bodies with the Fisher Experimental and Development Section, General Motors Corp. He joined Fisher in 1943 as a stress engineer at their Memphis Aircraft Plant, and has been in the Experimental and Development Section at Detroit since 1945.

DESIGN practices for body engineers have been established through years of experience in body building. Recently, however, the increasing emphasis on material economy and manufacturing costs has led to new research and development projects to improve the structural and mechanical efficiency of the body.

This extensive research has developed the type of structural engineer who specializes in design and analysis of body structure. Since theoretical and empirical formulas used in mechanical, civil, and aeronautical engineering, do not always apply to automobile body structures, it has become necessary to rely on structural testing as a means of supplementing available information.

A technique of complete structural testing, in conjunction with body and road tests, has enabled the engineer to evaluate the rigidity contributed by the body to the complete car. This data has shown the importance of considering the body and chassis frame as a unit.

The testing technique which is currently used enables the body designer and stress analyst to obtain information on the body structure in relation to the complete car. Evaluation of present production body structure as related to the complete car unit is especially useful as a basis for comparison of new body designs.

Some of the main factors which have influenced the development of body testing will be discussed before describing the test methods. Road and laboratory tests first were correlated so that road conditions affecting body structure could be duplicated in the laboratory.

A general procedure for laboratory tests was formulated from the following observations of road tests:

1. Failure of structural members due to severe

SINCE theoretical and empirical engineering formulas do not always apply to automobile body structure, reliance must be put on structural testing to supplement available information. In this paper the author presents a technique for complete car structural and road testing in conjunction with body tests for such purposes as:

1. Aiding development of experimental body designs.

2. Assisting in the standardization of quality control methods.

3. Solving special problems in fabrication.

4. Furthering the improvement of body-frame unit structural efficiency.

5. Investigating the use of new materials.

6. Determining safety factor of body.

The search for adequate methods of determining the safety factor led to the development of the roll-over test. It is accomplished by towing a car so that the wheels on one side go up a ramp at a 15-deg angle, and at a speed of 50 mph. The car is rolled in such a manner that the windshield header receives the greatest impact.

Methods in Body Structures

H. C. Johnson, Fisher Body Division, GMC

Paper presented at SAE National Passenger-Car, Body, and Materials Mtg., Detroit, March 6, 1952.

road conditions are not frequent. The failure of structural members is defined as the buckling or fracture of the major structural members of the body.

Uniform deflection of all component parts of the body is desirable; that is, each component part should not be weaker than any other component part. Load deflection and angular twist rates of the body-chassis frame unit, as well as of individual structural members, are used for comparison of the component parts. Testing to failure generally is of no value in determining rigidity of the body structure.

2. Failure of welded joints and sheet metal panels due to body shake and metal fatigue were caused by severe road test conditions.

Structural tests which simulated conditions of the joints as found in the body structure were conducted in the laboratory. Road durability tests were generally used to determine whether any failure of joints existed.

Metal fatigue of body panels was usually evident in durability road tests. However, conditions that cause metal fatigue were duplicated by laboratory methods on the body-chassis frame unit.

3. Body openings indicated a tendency to distort from the original positions after durability road tests.

Conditions found on the road were simulated in the laboratory by twisting the body-chassis frame in the manner imposed by severe road conditions. It was possible to observe the movements of body openings, structural members, and body panels. Comparison can be made between present production body structures and those incorporating new design.

Test Methods

The body is considered a part of the body-chassis frame unit. Therefore, it is necessary to find the structural qualities, first of the chassis, and second, of the body-chassis frame unit. With this information, it is possible to evaluate the rigidity that the body contributes to the body-chassis frame unit. The third step determines the structural qualities of the complete car. With the three steps, the rigidity contributed by the frame, body, and front-end sheet metal can be evaluated.

Structural testing used in stages of body development determines the value of new body design. The deterioration of the body-chassis frame can be determined after durability road tests.

Distortion tests conducted on the body-chassis frame unit in the laboratory by simulating severe road conditions, enables the engineer to observe reactions of the body structure, leading to studies of efficiency in fabrication, welding, and new design.

The test methods are described in the order in which the tests are usually conducted.

Chassis-Frame Structural Tests

Bend and torsion tests are conducted on the frame for the purpose of obtaining load-deflection

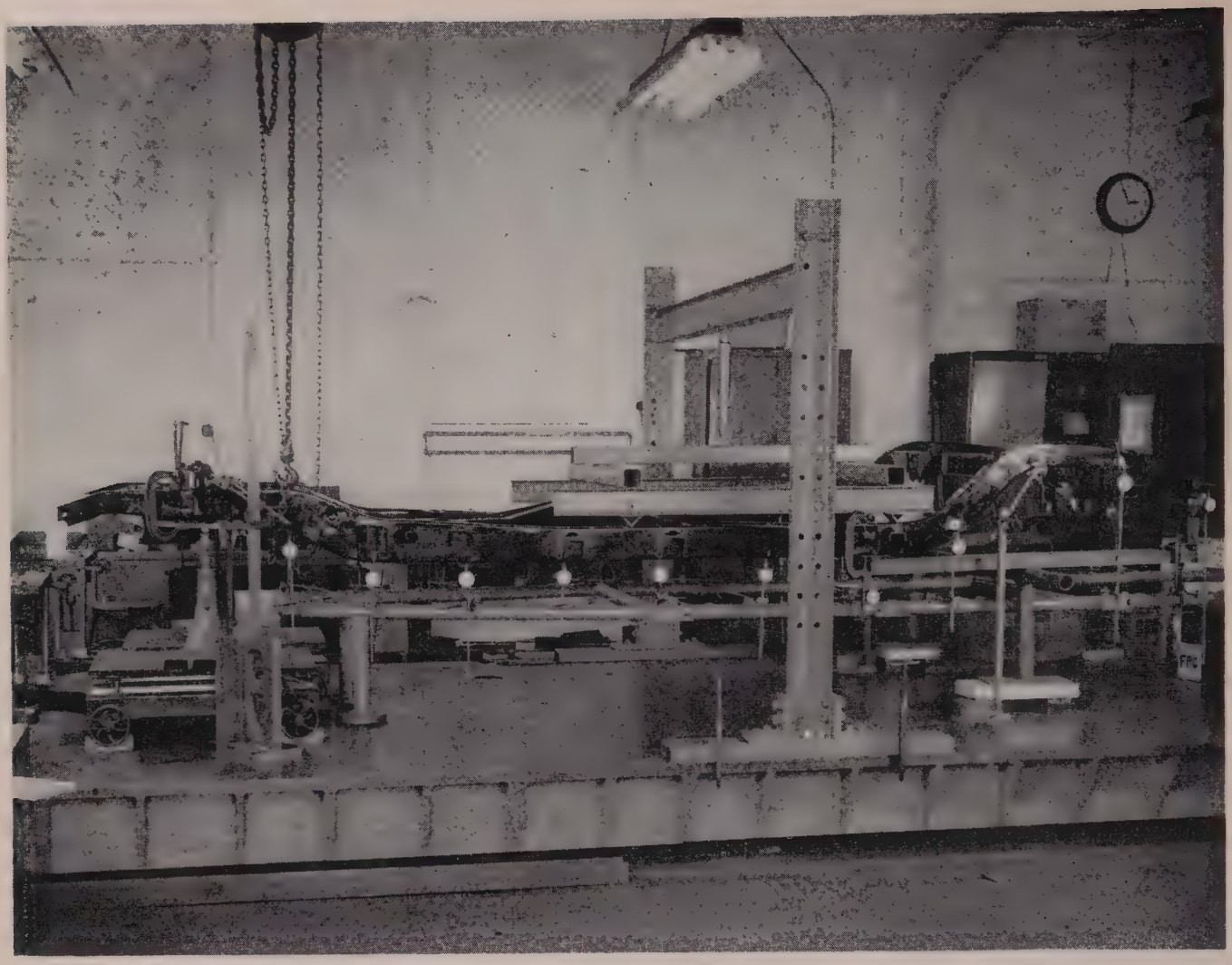


Fig. 1 – Setup for chassis frame bend test

and angular twist rates which determine:

1. By comparison of test results, whether the frame was more or less rigid than similar frames of the same wheelbase and car make.
2. Rigidity values to be used in conjunction with similar rigidity values on the body-chassis frame unit and complete car.
3. The effect on the rigidity of the frame caused by variations of metal gages and the shape of the frame.

Rear and stiffness tests are used for the purpose of evaluating the sag of the rear end of the frame with respect to the rear axle.

A brief description of the test methods used is given here.

Bend Test – From experience with bend tests of frames in conjunction with body-chassis frame tests, it is possible to detect changes in the frame that will increase the structural efficiency of the body-chassis frame unit. For example, if durability tests of the complete car reveal metal fatigue failures at the dash-brace location, bend test curves probably would indicate a weakness in that area. By increasing the metal thickness at that point, or changing the shape of the frame, a recheck of the frame would probably show that the deflection was uniform with respect to the overall curve. It is to be noted that normal curve deflections have been established by correlating road tests with laboratory results.

Bending stiffness correlation with road testing reveals that reduced bending stiffness introduces low-frequency vertical shake, and increased bending stiffness introduces high-frequency shake. With knowledge of bending stiffness values, it is possible to control some of the desired conditions necessary for good ride performance. Frame rigidity tests were not conducted for the purpose of changing the design of the chassis frame, but to obtain information that would be valuable in the structural design of the body.

A specific example of this was brought to the

attention of engineers when it was suggested that certain body mounts attached to the chassis frame be omitted between the front pillar and rear pillar locations. The first reaction was that it would weaken the overall bending stiffness of the complete car, thereby decreasing the durability of the car. However, road test information revealed the possibility of reducing vertical shake.

Instrumentation readings for the chassis frame on fairly smooth roads indicated constant vertical displacement of the frame at frequency ranges of approximately 750 cycles per minute. Especially high displacement at the point of maximum deflection was noted between the front and rear pillar locations. It was suggested that by omitting the body bolts in this area, the vertical shake in the body could probably be reduced. Further tests proved that the rigidity and durability of the complete car were satisfactory even though these body bolts were omitted.

The instrumentation test results did not reveal any change of vertical displacement, as anticipated. It was found, however, that there was a considerable reduction of road noise in the body.

In bend tests the chassis frame is simply supported at the front and rear axles. Frames using rear leaf spring suspension are supported from steel channels attached to the spring hangers at the rear axle location. This arrangement is necessary to simulate the support conditions of the complete car at the rear axle. A flexible attachment to the steel channel at the rear-rear spring hanger is

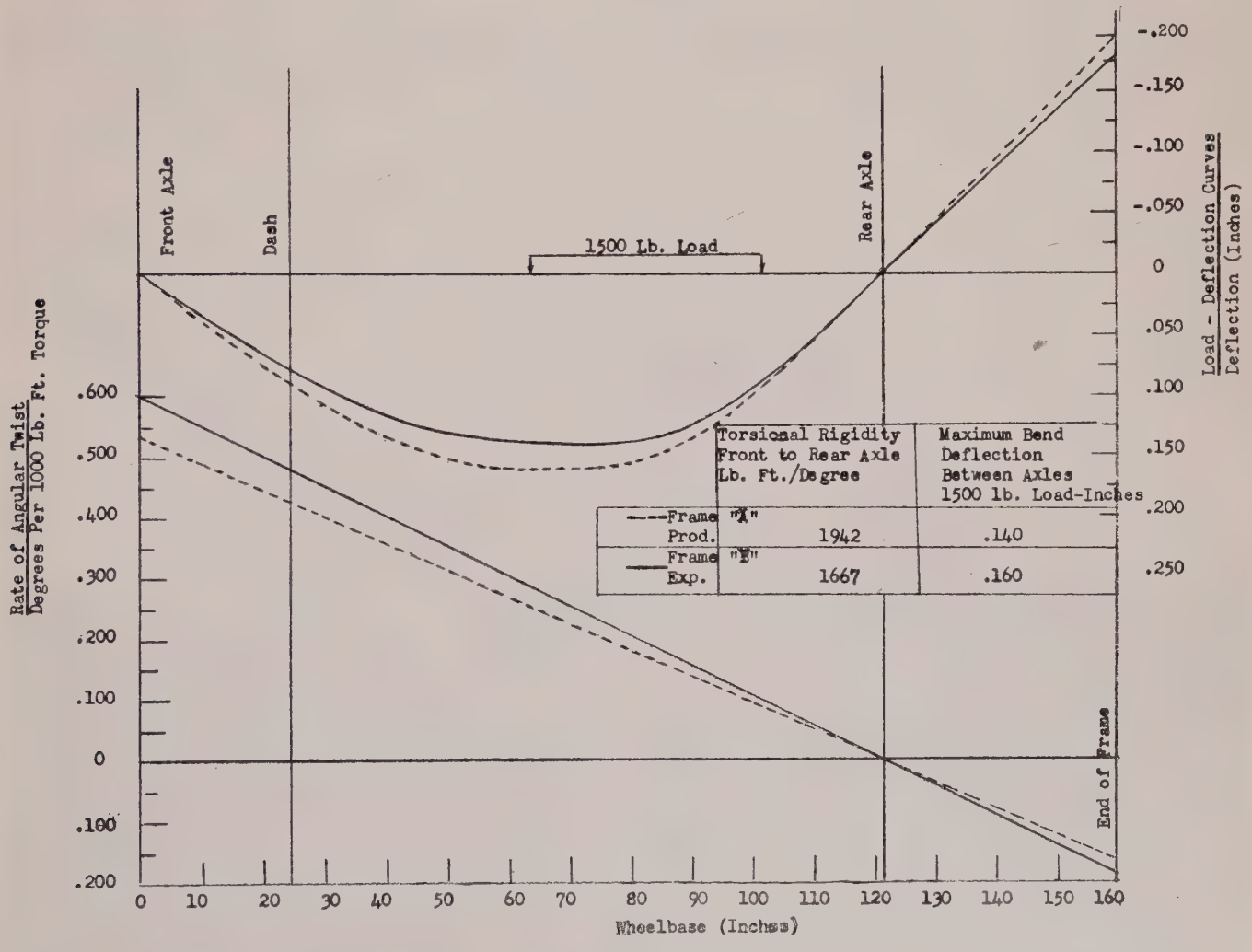


Fig. 2 – Bend and torsion curves showing comparison of production and experimental chassis frames

used to prevent restraint of the frame under load conditions.

Dial indicators graduated in one-thousandths of an inch are attached to an indicator support frame which is supported from the floor. The stems of the dial indicators are placed at the axle and body bolt mount locations on the lower surface of each side rail. The photograph of the bend test setup is shown in Fig. 1.

Torsion Tests – Road tests have shown the importance of torsional rigidity with relation to ride performance of the car. The reaction felt in riding in a car with low torsional rigidity is similar to the swaying experienced while driving over an uneven road. It is also possible, however, to have excessive torsional rigidity. For example, if one front wheel of a car rebounded from a depression in the road, the frame and body would not absorb the effects of the bump, resulting in a high-frequency shake in the body of the car.

Frame and body-chassis frame rigidity can be changed so as to produce favorable reactions with the spring suspensions. An increase or decrease in frame torsional rigidity may be achieved by relocation of body mounts, rearrangement of crossbars and "X" members, change of metal thickness.

The torsion test was conducted with the same type of rear axle support and dial indicator arrangement used in the bend test.

Platform scales are placed at the front axle location. The side rails of the frame forward of the centerline of front axles are supported from the scales by means of jacks. Weight is placed on the frame in order to hold it to the support at the rear axle, while twisting the frame at the jacks.

The torsional rigidity is expressed in terms of lb-ft torque per deg. Deflection and rate of angular twist curves are shown in Fig. 2.

Rear-End Sag Test – Rear-end sag tests were developed so as to control the rigidity of the rear end of the frame and body-chassis frame unit. By testing cars which gave satisfactory results on the road, it was possible to set up rear-end sag deflection rates with which new frames and bodies could be compared.

This test is conducted by using the arrangement utilized in the bend test, with the exception that a 400-lb load is applied at the center of the rear-end crossbar.

The test results are expressed in inches of deflection of the rear end of the frame with respect to the centerline of the rear axle.

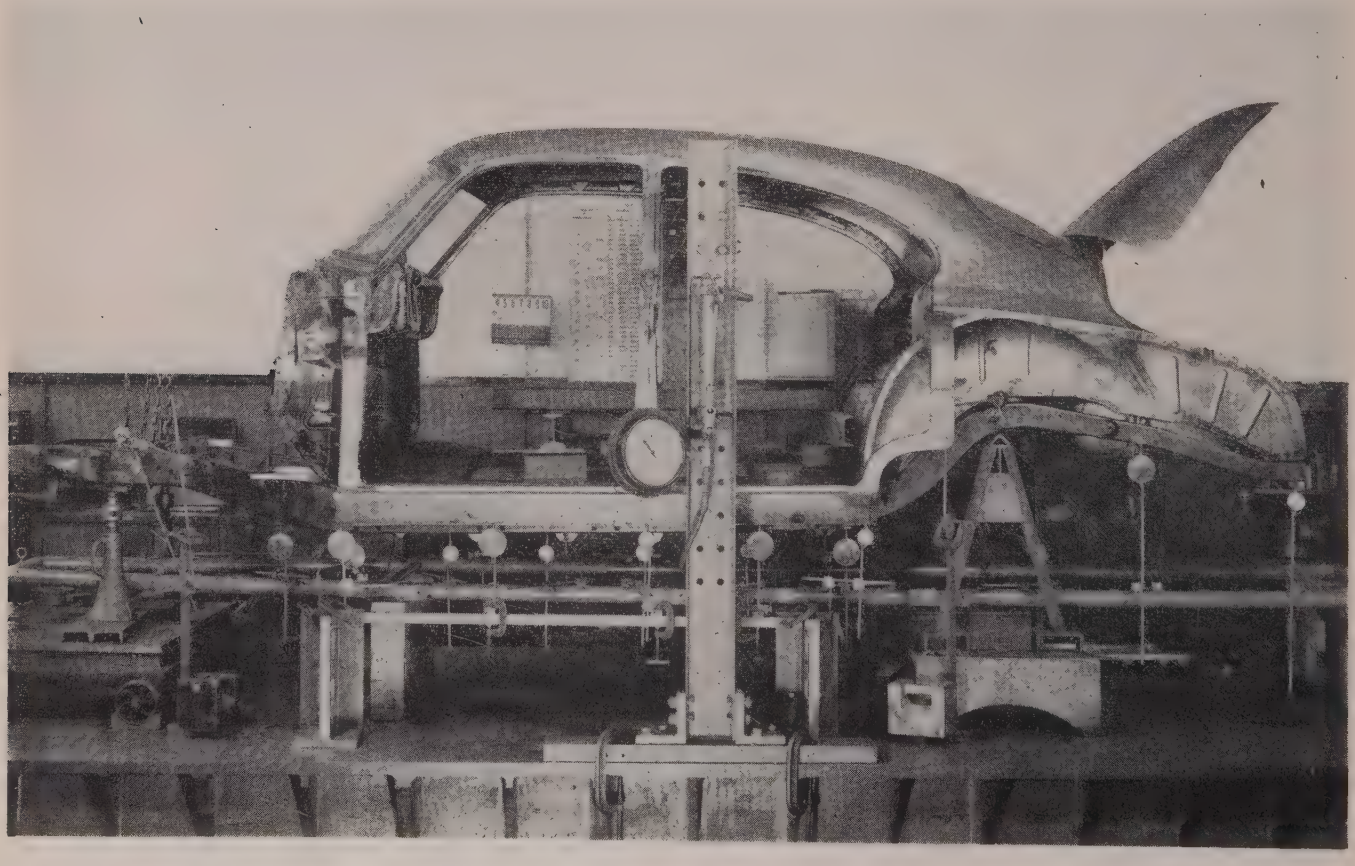


Fig. 3 – Setup for body-chassis frame bend test

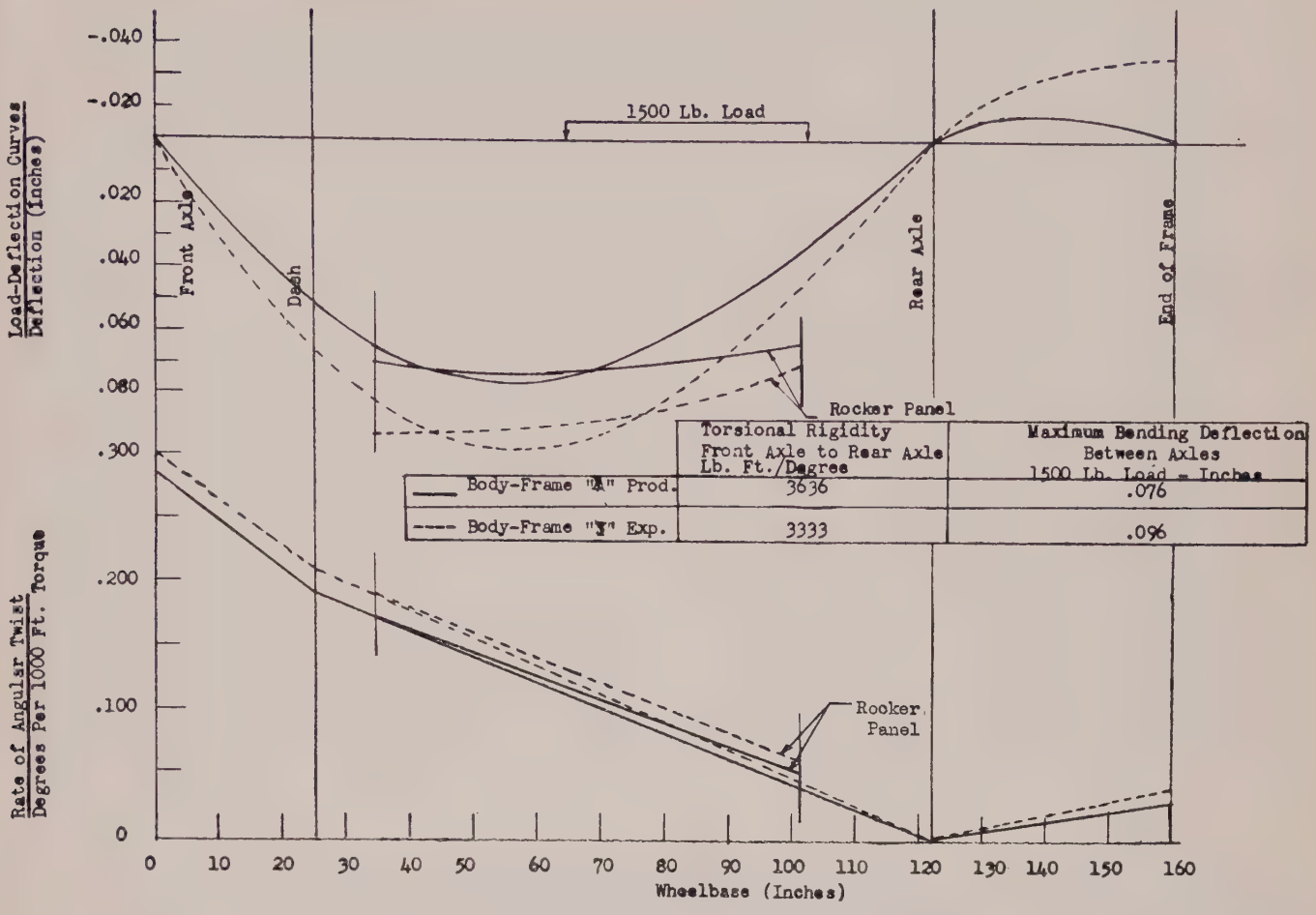


Fig. 4 – Bend and torsion curves showing comparison of production and experimental body-chassis frame units

Body-Chassis Frame Unit Structural Tests

The test procedure is the same as that described for the chassis frame, except that additional indicators are used to obtain deflections of the lower surface of the rocker panels. These rocker deflections indicate bending stiffness of the rockers with respect to the frame rigidity values of the body-chassis frame unit. It is to be noted that the windshield glass and back light glass are installed before the test, inasmuch as the glass increases body rigidity approximately 5%. The doors and deck lid remain in an open position during the test. The test results are used in the comparison of similar units, and for evaluation in reference to the complete car.

The rear-end sag test is conducted in the manner already explained for the chassis frame. The test results are used for comparative purposes. After the correlation of this test data with road test data, it is possible to set up limits of acceptability for rear-end sag. (See Figs. 3, 4, and 5.)

Some of the special investigations conducted in conjunction with the body-chassis frame unit tests are as follows:

Body Shim Study – The body bolts, which are used to attach the body to the chassis frame, are shimmed with various materials such as steel, tire carcass, or mechanical rubber shims. It is possible to obtain rigidity values which show the effect of the different types of shims on the body-chassis frame unit. This information is used in conjunction with road tests.

For example, while riding over a rough road the ride performance of the complete car may be harsh and unyielding. Changing to a softer shim often materially reduces this condition which can probably be defined as a high-frequency shake. However, a softer shim would also reduce the bending stiffness and torsional rigidity, and increase the low-frequency of the complete car. It should be kept in mind, therefore, that considerable reduc-

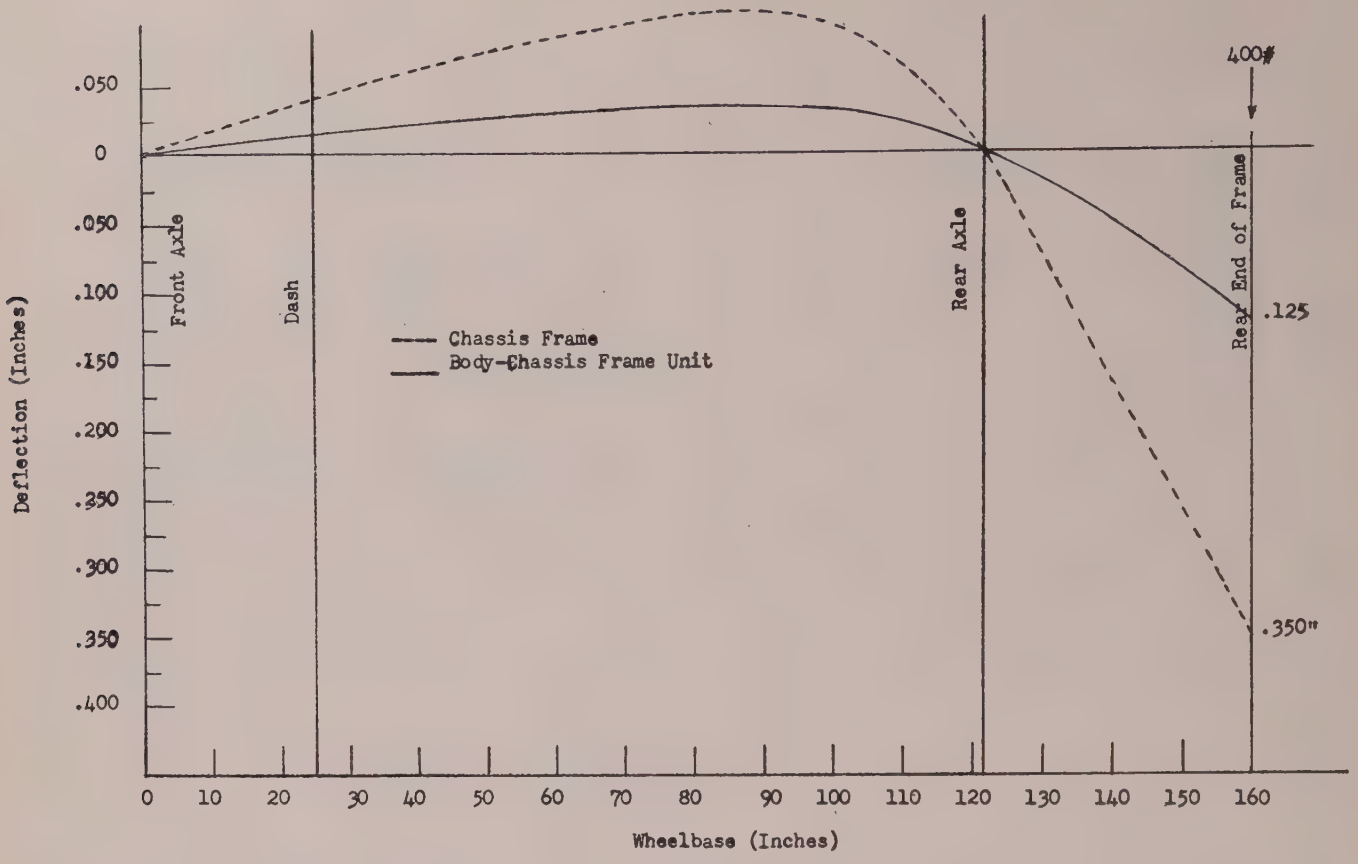


Fig. 5 – Rear-end sag deflection curves

tion in rigidity will affect the durability qualities and the low-frequency shake of the body and chassis frame.

However, the test results showing rigidity of the body-chassis frame due to body shims, are not useful unless they are needed to solve a condition encountered in road tests.

Structural Design Changes – Since the body-in-white is used in the body-chassis frame unit, it is convenient to interchange structural members for the purpose of evaluating their effect on the rigidity of the body-chassis frame unit. This information is again used for comparative purposes in conjunction with other tests, such as shake-rig distortion and road tests.

In connection with body-chassis frame tests, detail tests of structural members are conducted by simulating as nearly as possible the assembled conditions of the members. The desired deflections under simulated loads will then be obtained before assembling into the body for a final check.

Strain Gages – Strain gage readings are used to determine magnitude and direction of load in structural members caused by bend and torsion test conditions.

A knowledge of the magnitude and direction of load in certain structural parts of the body enables

Table 1 – Distortion Test Results

Description	Production X	Experimental Y
Angular Twist, deg		
1. Front Axle to Rear Axle	0.253	0.253
2. Dash on Frame to Rear Axle on Frame	0.159	0.154
3. Dash on Body to Rear Axle on Body	0.124	0.089
Distortion, in.		
1. Front Pillar Diagonals, average	0.039	0.042
2. Center Pillar Diagonals, average	0.027	0.015
3. Rear Pillar Diagonals, average	0.079	0.032
4. Windshield Opening Diagonals, average	0.007	0.021
5. Front Door Opening Diagonals, average	0.065	0.053
6. Rear Door Opening Diagonals, average	0.052	0.043
7. Backlight Opening Diagonals, average	0.008	0.038
8. Deck Lid Opening Diagonals, average	0.005	0.005
9. Rear Seat Back Opening Diagonals, average	0.035	0.012
10. Roof Top Diagonals, average	0.042	0.022
11. Dash Diagonals, average	0.009	0.004
Lateral Bending – at Base of Windshield Pillar	0.002	0.000
Vertical Bending – Roof Rail – Top of Windshield Pillar	0.020	0.016
Relative Movement – Body to Frame at Rear Axle	0.019	0.034

the stress analyst and body designer to determine efficient structural design.

Stress Coating – The application of stress coating is used to determine stress concentration of body panels and joints due to the bend and torsion test conditions on the body-chassis frame unit.

Body-Chassis Frame Unit Distortion Tests

Movements of the body of a car as observed while traveling over a rough road are simulated by twisting the body-chassis frame unit alternately on a specially designed shake rig. The distortion of the body enables the engineer to locate any abnormal movement.

These tests are made by twisting the chassis frame alternately at the spring housings of the front crossbar, with the frame at the rear axle remaining stationary. Alternate twisting of the frame is accomplished by lowering left spring housing and raising the right and vice versa.

The rear axle attachment is rigidly supported by the stationary base of the shake rig, using the channel supports attached to the rear spring hangers for leaf spring type of rear suspension. The spring housings at the front crossbar are fastened to the movable platform of the shake rig. The shake rig is moved vertically, with the sides moving up and down alternately. The stroke is adjusted for approximately 0.250 deg angular twist of the chassis frame. This twist simulates severe road conditions and causes distortion in the body which can be measured and observed.

Comparison of similar body movements also discloses weld deficiencies and poor assembly of body parts. It is essential that the experimental body is built to specifications so that errors in fabrication will not influence excessively the general reaction of the body.

Localized distortions that are considered abnormal are corrected before removing the body-chassis frame unit from the shake rig. Body-to-chassis mount studies are also conducted by varying the shim conditions and body bolt torques.

Fig. 6 shows test setup for distortion test.

The test results are recorded in Table 1.

Complete Car Structural Tests

The body-frame unit is assembled into a complete car for the final phase of testing. The series of structural tests conducted on the complete car determines the deterioration of the car in terms of rigidity caused by road durability tests. These tests also complete the phase of testing necessary to determine the rigidity contributed by the frame, body-chassis frame unit, and front end sheet metal.

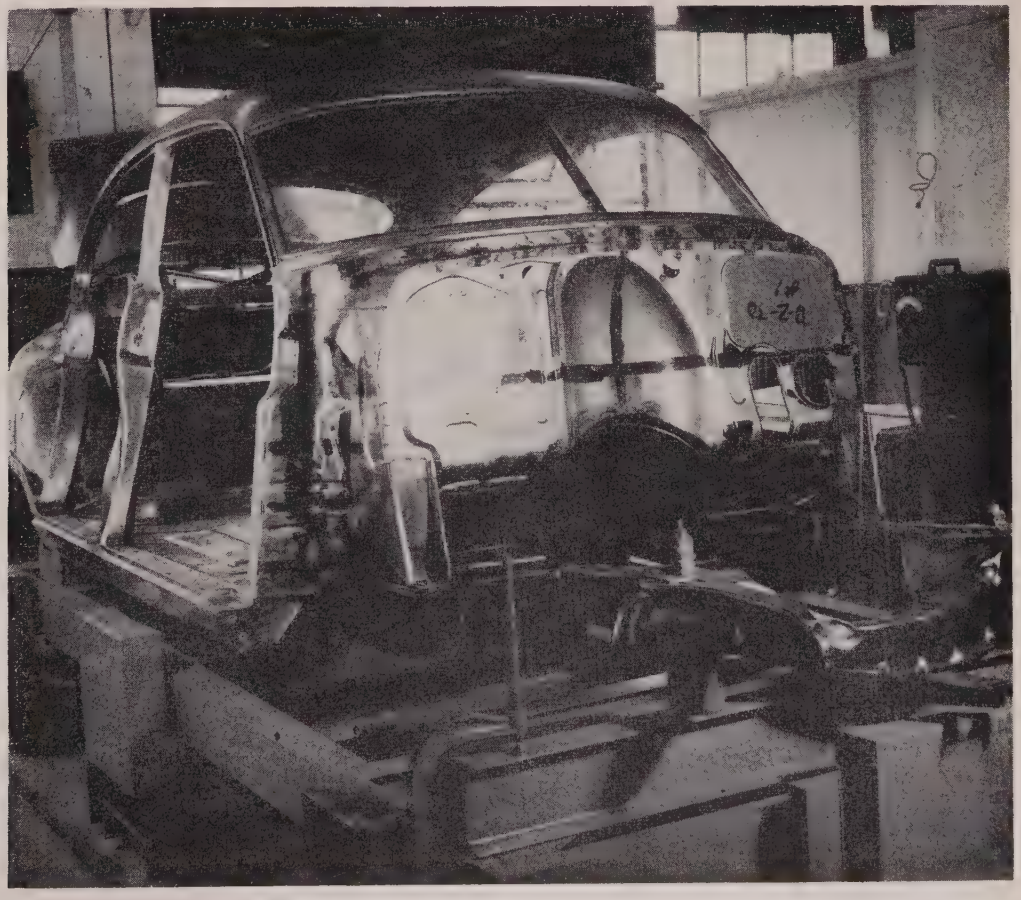


Fig. 6 – Setup for shake rig distortion test

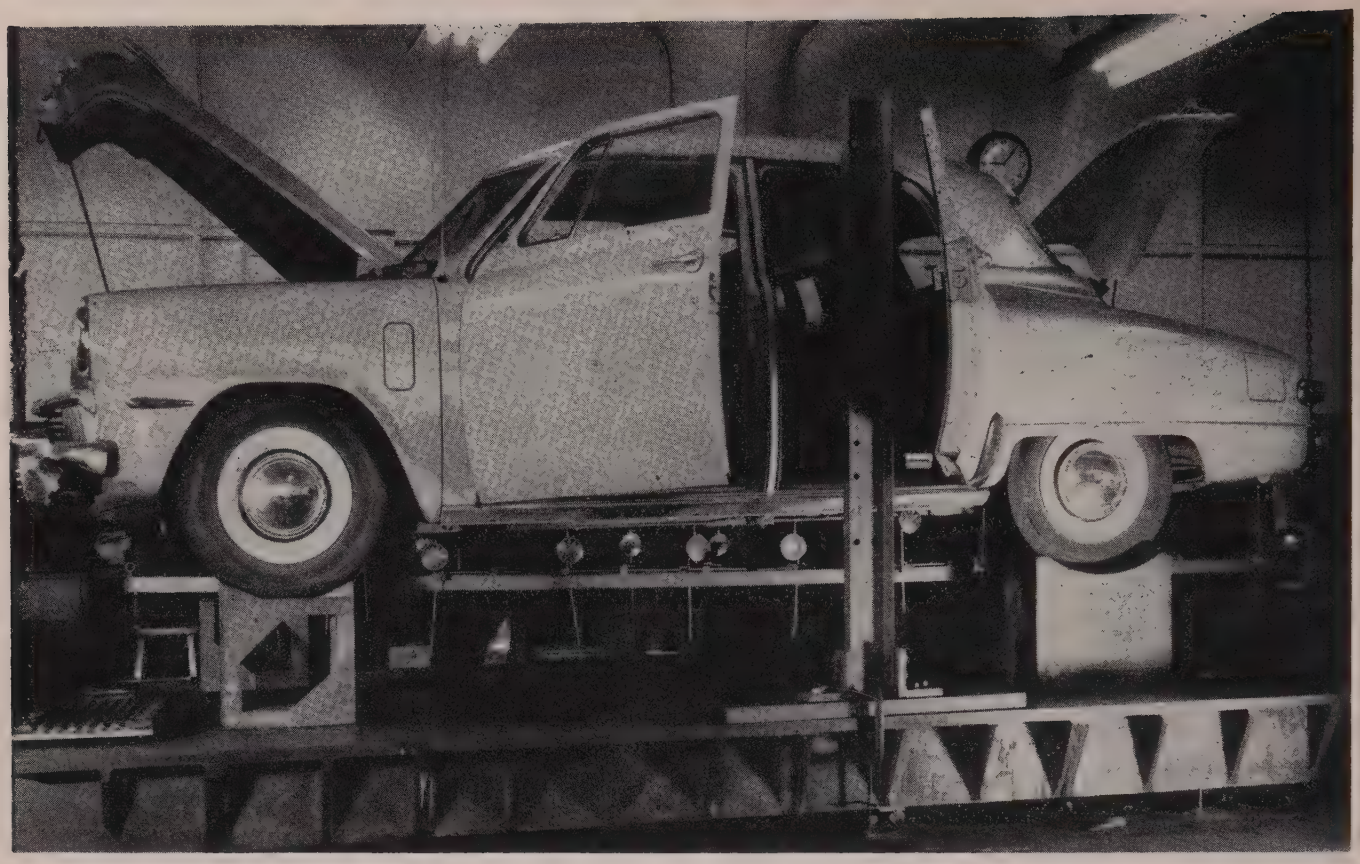


Fig. 7 – Setup for complete car bend test

An accumulation of bend and torsion test data covering complete cars of the same wheelbase and car make, together with durability road test data, enables the engineer to set up limits of acceptable rigidity. Usually, failures of body structural members would be detected by a decrease in bend and torsional rigidity values.

Bend and torsion tests of the complete car are conducted in such a manner that the chassis frame and body-chassis frame tests and results can be compared directly with the complete car test results. (See Table 2.)

Bend Test – Test methods are planned so as to require a minimum removal of body and chassis parts. The spring suspensions and wheels provide supports for the frame at the front and rear axle locations.

The support frame for the indicator is attached to the center of the front crossbar and at rear bumper brackets. Supported in this manner, the dial indicators remain in one plane. Any bending of the chassis-frame is registered by the dial indicators. The axle locations are considered as zero deflections.

By conducting bend tests with the spring suspensions and wheels removed, and supporting the frame in the same manner as for the body-chassis frame unit, it is found that the test results are comparable with those with the suspension and wheels in place. (See Fig. 7.)

Torsion Tests – Torsion tests are conducted in the same manner as the body-chassis frame test except that the spring suspensions and wheels are in place. The rear suspensions are blocked out by placing a steel block between the frame at the centerline of axle and rear axle housing. A load is placed in the car to hold the frame rigidly on the steel blocks while twisting at the front axle location.

Body and Frame Contour Check – The permanent set of the body frame is determined after road

Table 2 – Test Results of Chassis Frame, Body-Chassis Frame Unit, and Complete Car

Description	Torsional Rigidity Front Axle to Rear Axle, lb-ft per deg		Maximum Bending Deflection with 1500-Lb Load, in. between axles	
	Production X	Experimental Y	Production X	Experimental Y
Chassis Frame	1942	1667	0.140	0.160
Body-Chassis Frame Unit	3636	3333	0.076	0.096
Complete Car	4085	4000	0.050	0.052

durability tests. Before the road durability tests, frame and body check points are scribed on the complete car and measured from a surface plate with the car supported at the front and rear pillar locations on the frame. After durability tests, the car is again checked in the same manner.

Repeated checks of durability test cars enable the engineer to set up limits of acceptable permanent set. (See Fig. 8.)

Durability Road Tests

Many miles of driving over roads presenting all types of travel conditions were necessary before a method of road testing could be developed which would amplify and accelerate body test conditions. A main purpose was to determine road conditions that would twist the body and distort body openings. A cobblestone road approximately two miles in length, with irregularities in stones varying up to 5 in. in height, was found to give the most desirable road conditions.

It was determined that the body structure in a test car should complete 2000 miles of cobblestone road at an average speed of 30 mph with satisfactory results.

Test cars with new design of major structural body members are subjected to 5000 miles of cobblestone road for the purpose of attempting to fail the body structure.

Measurements made before and after durability tests are as follows:

1. Rubber weatherstrip pressure.

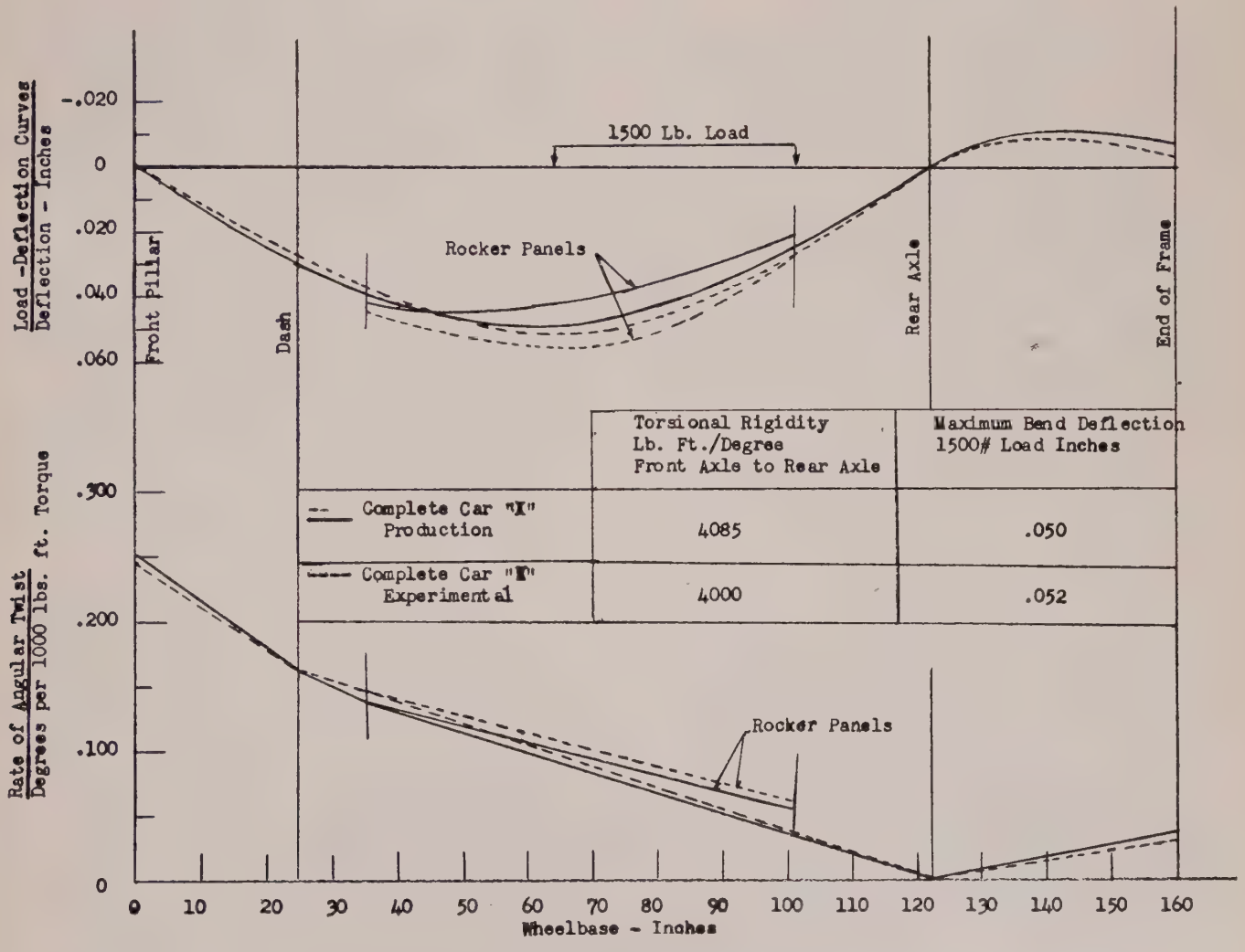


Fig. 8 – Bend and torsion curves for complete cars, showing comparison of production and experimental models

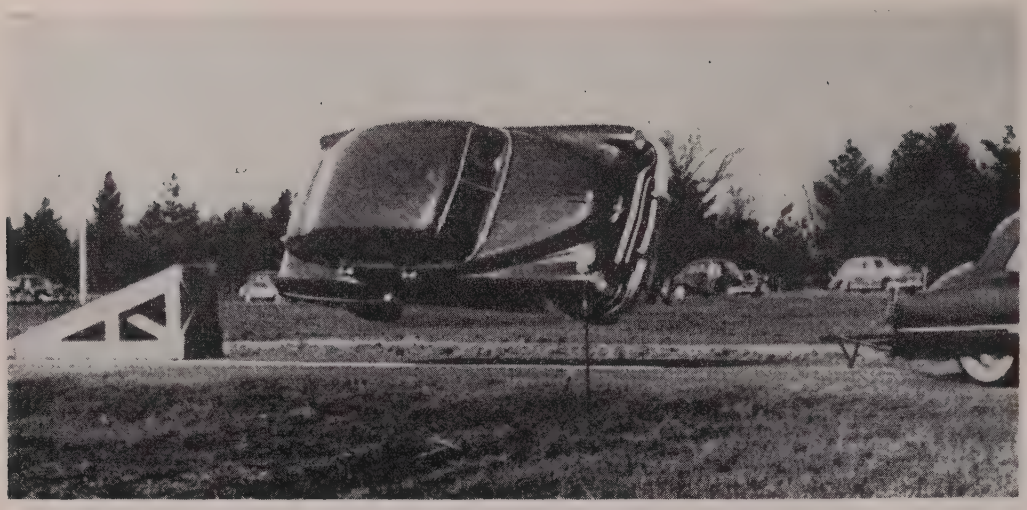


Fig. 9 – Beginning of roll-over test

2. Door and deck lid lock release.
3. Door sag with respect to lock pillar.
4. Deck lid opening measurement.
5. Instrumentation tests of ride performance.

The test car is also inspected daily for body and chassis failure. All failures are repaired when found. Repeated failures are investigated. As a rule, one failure is not considered sufficient to indicate a design failure.

Many structural tests, before and after durability tests, indicate that 3 or 4% reduction in rigidity of the complete car is generally considered normal.

Electrical instrumentation for indicating vibration has become an effective method for solving many body vibration problems, such as mirror shake, door flutter, and localized body part vibrations.

Roll-Over Test

For the purpose of determining the safety factor of the body, a roll-over test has been devised. In this test, the car is rolled without occupants, in such a manner that the windshield header receives the greatest impact. The roll-over is accomplished by towing a car so that the wheels of one side of the car go up a ramp with a 15 deg angle and approximately 3½ ft high, at a speed of 50 mph. The towing car misses the ramp, and is disengaged from the towed car as one front wheel of the towed car travels up the ramp. By many trials it has been possible to control the roll-over so as to obtain the desired condition. (See Figs. 9 and 10.)

Conclusion

The above described testing technique is a valuable asset in the development of experimental body designs. It is also important in the following phases of engineering and manufacturing of bodies:

1. Standardization of quality control methods.
2. Solving of special problems in fabrication.
3. Improvement of structural efficiency of body-frame units.
4. Special investigations for utilization of new materials in body structure.

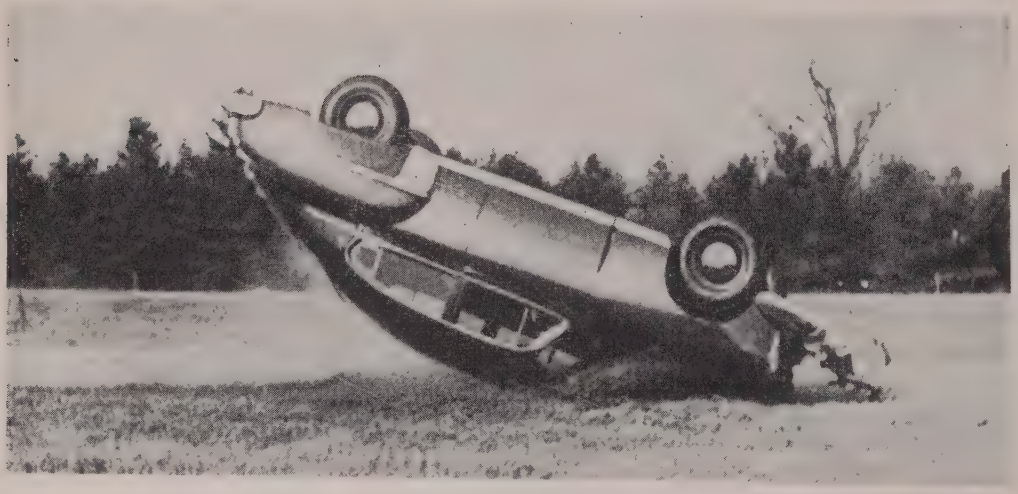


Fig. 10 – Finish of roll-over test

STEERING OF TRACK-TYPE VEHICLES

William W. Henning, International Harvester Co.

Paper presented at Earthmoving Industry Conference, SAE Central Illinois Secion, Peoria, Apr. 9, 1952.

THE track-type vehicle was stimulated into a position of importance by the military requirements of World War I, and since has been the subject of continual development for both military and commercial application. Vast strides have been made in the design of tracks and suspensions, making possible higher road speeds and more efficient utilization of power. Track machines are now serving in many divergent fields, both as prime movers and as transportation vehicles. As the speed of track-laying vehicles increases, and as the requirements for increased productivity become more pressing, the importance of controllability and efficient power transmission become more evident, necessitating careful consideration of the problems of steering mechanism design.

It is not possible within the limits of this paper to comprehensively cover the general subject of steering of track-type vehicles, therefore the discussion will cover some of the more practical aspects of popular types now in use. In particular, it will deal with the mechanisms by which steering is accomplished in the full-tracked, 2-track, and laterally rigid machine, which represents the most prevalent present-day application of tracks, and is exemplified by the crawler tractor, military tank, landing vehicle tracked, and others. It will also deal with some of the common factors which are so often overlooked or improperly evaluated in choosing a steering mechanism for a particular vehicle.

If the design of the ideal steering mechanism

DURING the period between the two World Wars the track-type vehicle was stimulated into a position of prime importance both in military and civilian applications. This paper reviews the vast strides made in the design of tracks and suspensions, which in turn have led to higher road speed and more efficient utilization of power. However, with the increase of speed, the importance of controllability and power transmission becomes more evident, necessitating careful consideration of the problems of steering mechanism design.

The author describes an ideal system which, when and if designed would undoubtedly be universally adopted. But only those who have attempted to come to grips with the problem in a practical manner realize that the final solution will not be a simple one. The present result is that there are almost as many steering systems as there are different types of vehicles. The many reasons leading to such wide-range developments are covered in this paper.

WILLIAM W. HENNING (M '45) is divisional chief engineer, advanced engineering, Industrial Power Division, International Harvester Co. He received his B.S. in M.E. from Armour Institute of Technology in 1935, and joined International Havester that same year.

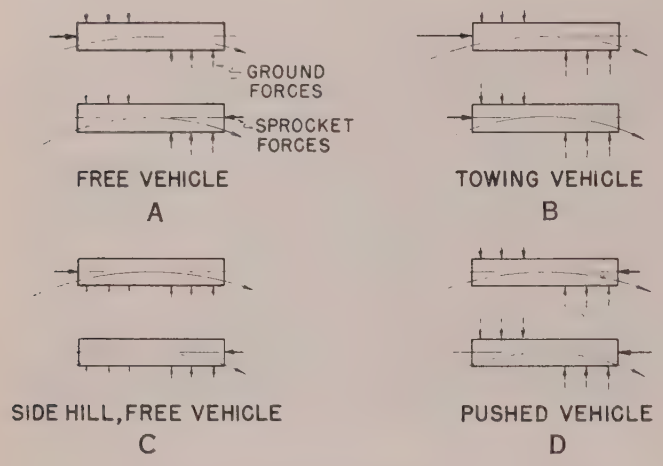


Fig. 1 – Steering forces during sustained turn at low speed

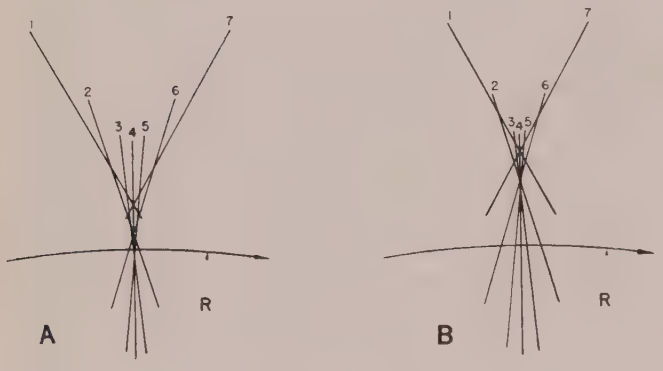


Fig. 2 – (A) illustrates motions relative to ground which grouser goes through during turn in which there is no longitudinal slip. (B) shows results of 5% slip superimposed on (A)

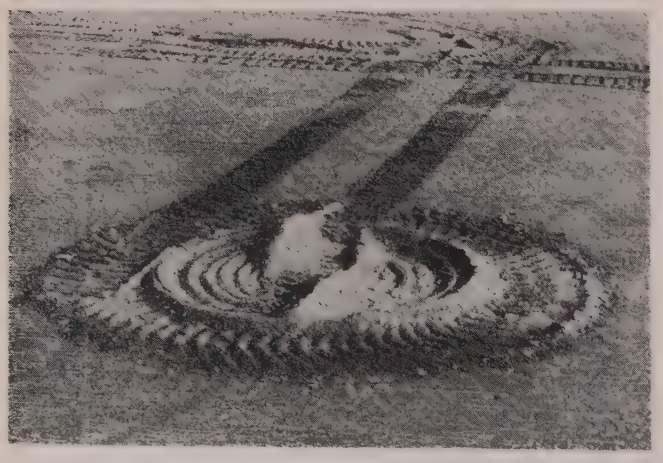


Fig. 3 – Pattern made by tractor making 360 deg pivot turn at slow speed, showing severe cutting of ground

were reasonably simple, straightforward, and commercially feasible, all track vehicles would undoubtedly by this time be steered with the same general type of device. Since this is not the case, it can be assumed that the problem is not so simple, and the solution must be a compromise of various factors which do not have the same relative importance when considered for different vehicles.

Before examining the more significant types of steering mechanisms now in use, some of the factors which lead to the selection of a particular system for a given application will be considered.

In creating a machine, the designer attempts to furnish at reasonable cost a device of most favorable productivity, safety, durability, and ease of maintenance. Machine productivity is a combination of mechanical and operator efficiencies. Therefore, it is of utmost importance to not only produce a machine capable of high mechanical efficiency, but also one that is conducive to high operator efficiency. An operator obviously cannot get more out of a machine than it can mechanically deliver, but ease of control can increase the ratio of actual work done to possible work done. This ratio may be termed operator efficiency. It is always preferable to have machines operated by skilled personnel, but in too many cases, particularly in the military, extremely unskilled operators must be used. It is necessary that a machine be capable of being operated with some degree of efficiency and a high degree of safety by an inexperienced operator. In the final analysis, the factors of efficiency, safety, cost, and ease of maintenance must be evaluated against their relative importance in the particular vehicle being investigated. It will be seen later that efficiency and safety are greatly affected by the characteristics of the steering control system.

While some experimentation has been carried out using laterally deflectible track suspensions which provide guidance by laying the track pads in a desired course of travel, all present track vehicles, to the writer's knowledge, are of the laterally rigid type in which deviation from a straight path is by bodily sliding the track pads on the ground. Desirable or not, the expedient of simplicity has dictated this construction.

For any specific vehicle in any particular degree of turn, the power losses encountered at the ground will be independent of the mechanism provided to induce the turn. It will be shown, however, that this is not the only significant loss, but that other characteristics are of importance, and are dependent upon the steering mechanism, the design of which greatly affects the power losses under certain circumstances. The sliding, slipping, and turning of the track pads in the ground during steering, is a disorderly process as well as a power consuming one. Resistance of the ground to such movements is highly variable and may change substantially between the time the pad first touches the

ground and is finally lifted from it. The forces generated by the sliding of the pads during steering resist change in direction of the vehicle, and at slow speeds are genarally the dominating factor. Other factors are inertia, rolling resistance, degree and direction of grade, location of the center of gravity of the vehicle, ground contour, effect of towed loads, and so forth.

To better understand the types of forces involved, Figs. 1A to 1D are shown. Fig. 1A illustrates in a much simplified manner the types of forces generated during the steering of a free vehicle at low speed. The broken arrows on the tracks illustrate the resistance of the ground to turning, while the solid arrows illustrate the driving force on the track required to negotiate the turn. The driving forces are under control of the operator although in some cases he may not control them directly as forces but rather, by controlling sprocket speed, will cause these forces to generate automatically. In the same manner, the conditions of the tractor with a towed load (Fig. 1B), a free vehicle on a side hill (Fig. 1C), and a pushed vehicle (Fig. 1D), are illustrated. A comparison of these figures will clearly illustrate the fact that in order to negotiate the same degree of turn under various loading conditions, entirely different sprocket forces are required. This is an important point and explains the reason for the difficulty of control encountered in a steering system depending on a balance of frictional forces to control sprocket speeds. Fig. 2A illustrates the motions relative to the ground which an individual grouser goes through during a turn in which no longitudinal slip takes place. Fig. 2B is the result of 5% slip superimposed on Fig. 2A. It will be noted that the grouser moves back and forth over the same ground, cutting it up badly and changing its effective frictional coefficient during the turn. The similarity can be seen in the photograph, Fig. 3, which illustrates the pattern made by a tractor making a 360 deg pivot turn at slow speed, and which demonstrates the severe cutting of the ground both by the inner and outer tracks.

So far nothing has been said of the relative speeds of the two tracks on a vehicle. Depending on the internal mechanism of the machine, the steering forces may cause the required speed differential between the tracks, or a predetermined speed differential may cause the steering forces. While it may not be readily evident thus far in the discussion, a given vehicle turn is very closely regulated by the relative speed of the tracks, so that it can be stated that the problem of steering a track vehicle is one of regulating the relative speeds of the tracks. As the sprocket loads have been shown to vary with practically every condition of the ground and load distribution, any system in which the speed of tracks is regulated by control of the torque delivered to the driving sprockets, must require either a skilled operator

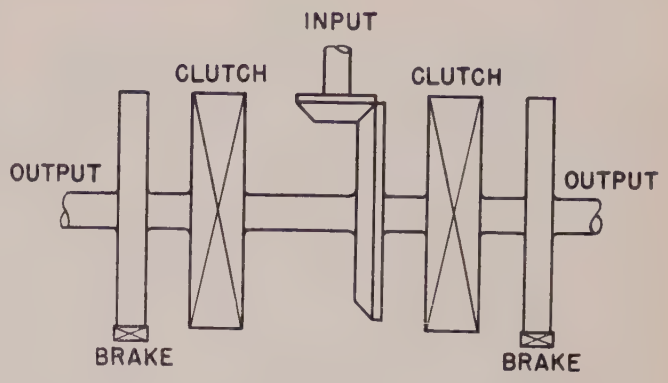


Fig. 4 – Steering clutch and brake system commonly used in commercial tractors

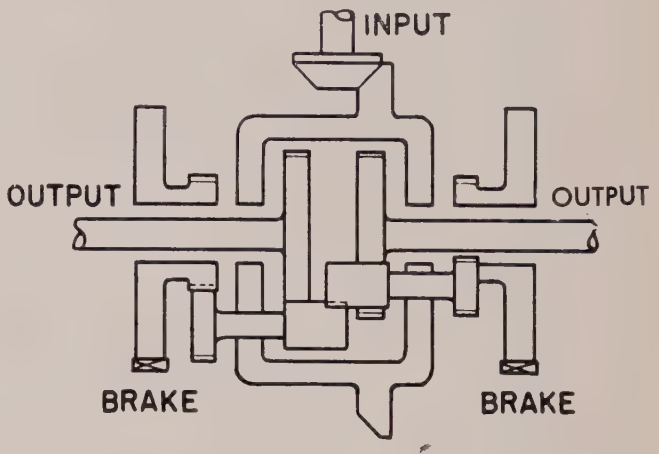


Fig. 5 – Controlled differential system used on both commercial and military vehicles

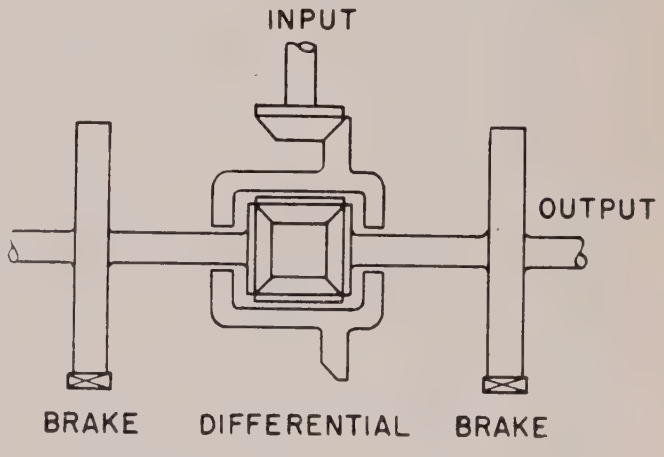


Fig. 6 – Braked differential has limited application due to lack of geared ratio turn

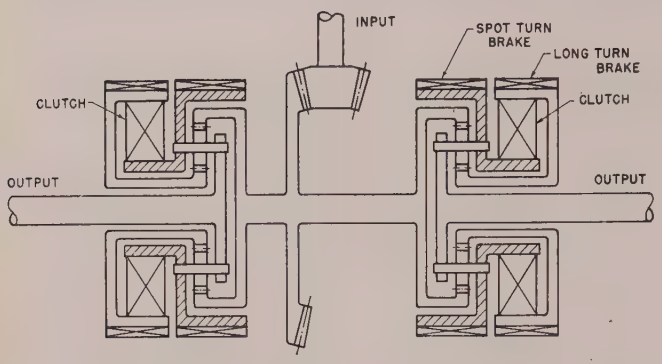


Fig. 7 – Newly developed Cletrac spot turn can make a completely geared turn at a fixed radius

or an intelligent mechanism. Therefore, the most desirable mechanism (from a control standpoint) will be one which controls track speeds at the operator's discretion, consistently and independently of the torque delivered. To those who have struggled with this problem, this is not new nor is the solution simple. This is evidenced by the fact that the following mechanisms, all different, are in current use in the United States:

Commercial:

Steering clutch and brake (see Fig. 4) includes Caterpillar, Allis-Chalmers, International Harvester, and others.

Controlled differential (see Fig. 5) such as Oliver Cletrac.

Braked differential (see Fig. 6).

Spot turn (see Fig. 7), the Oliver Cletrac.

Planet power (see Fig. 8), the International Harvester.

Military:

Cross drive (see Fig. 9).

Controlled differential.

These variations in steering system represent only a few which had been experimented with or built into vehicles in the United States and Europe. Each one is a compromise between the factors of function, durability, and simplicity, as dictated by the ultimate requirements of the consumers. As all of them are far from perfect, it is to be expected that a considerable amount of research and development effort is being directed toward the perfection of this type of mechanism.

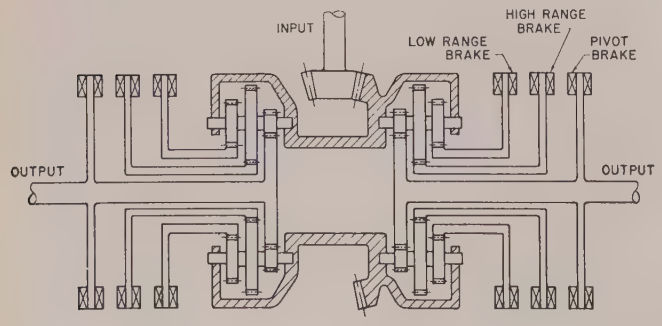


Fig. 8 – International Harvester's planet power steering system has much in common with the spot turn

Figs. 4 to 10 illustrate schematically several commercial types as well as some types used in various military vehicles. Each mechanism has its virtues and its faults.

The steering clutch and brake system, Fig. 4, is most commonly used in commercial tractors, and steers the machine by disconnecting one side either partially or fully, either allowing that side to slow down or forcing it to slow down by application of the pivot brakes, depending upon the combination of external forces acting on the vehicle.

The controlled differential, Fig. 5, has been used on a great many commercial and military vehicles, particularly during World War II, and depends upon the stabilizing forces of the ground contact and self-locking ability of the differential to maintain straightforward guidance. When turning, the application of a brake to one side of the planetary produces a geared turn of fixed ratio. It is evident that pivot turns are not possible and that the outer track speeds up as the inner one slows down.

The braked differential, Fig. 6, has had limited application due to the fact that like the controlled differential, it lacks straight-ahead stability and speeds up the outer track as the inner slows down, but at the same time does not have the advantage of a geared ratio turn. Pivot turns can be made but are not usually practical due to high power consumption.

The newly developed Oliver Cletrac spot turn (Fig. 7), and the International Harvester planet power (Fig. 8) steering systems have much in common theoretically, but differ radically in the means of accomplishing the objective. Each system has two geared ratios for each side which, in effect, is the same as having a 2-speed power shift transmission on each side of the tractor. In addition, the power may be disconnected and a brake applied to one side for pivot turns. It is noted that each system therefore can make a completely geared turn at a fixed radius.

The cross drive illustrated in its most recent version, Fig. 9, is a more complex device designed to produce high mobility in military transport vehicles, and combines the speed-change transmission as an integral part of the steering mechanism. It will be seen that this is not just a combination of two elements into one package, but is one which produces a very definite steering characteristic by the interrelationship of transmission ratio to steering ratio. For each transmission ratio, one geared

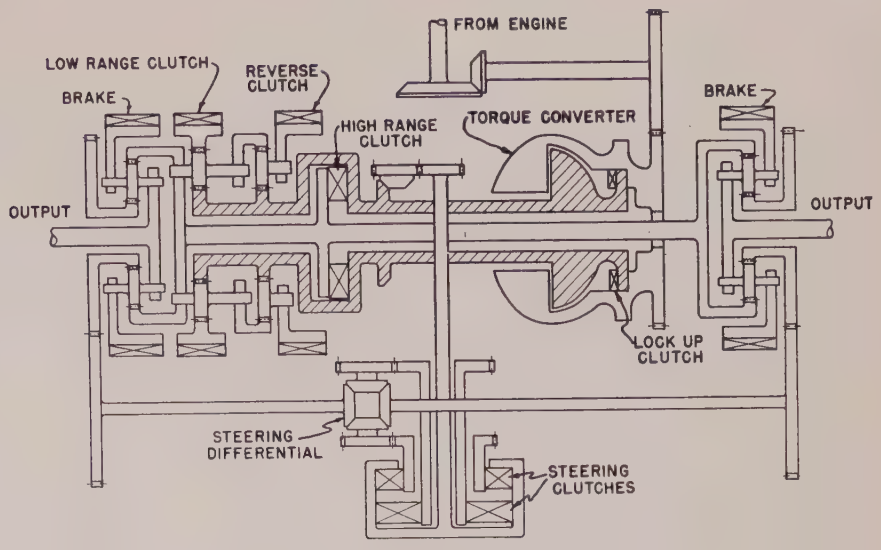


Fig. 9 – The cross drive is a complex device designed to produce high mobility in military transport vehicles

turning ratio is obtained which results in turning radii inversely proportioned to the transmission gear ratio. Pivot turns are possible, and with the transmission in neutral, the vehicle will turn about its own center with one track driving forward and the other driving reversed. Intermediate turning radii are possible by slipping of the steering clutches. Slowing of the inner track results in a corresponding increase in speed in the outer track.

The change in track speed during a turn is accomplished by combining the transmission output speed with an additive or subtractive speed from the steering differential through an epicyclic gear train. When driving straight ahead, the inherent resistance to steering provided by the tracks in the ground, is depended upon to maintain course.

The Merritt Brown system (Fig. 10) used in British tanks during World War II has some characteristics in common with the cross drive in that it combines the speed-change transmission with the steering mechanism, and regulates relative track speeds by combining, through an epicyclic gear train, the transmission output speed with that of an input-driven cross-connected differential, in a proportion dependent upon the slippage of the steering brake. It also has a geared turning radius for each transmission speed, and can turn on its own axis with the transmission in neutral.

Both the cross drive and the Merritt Brown systems have distinct advantages for high speed operation, as steering losses are low and steering sensitivity is less at higher speeds, due to the larger geared radius of turn in the higher gears. In addition, no power train disconnect is encountered during steering.

As the vehicles in which the systems have been installed have several different ultimate uses, no single yardstick for judging relative merits, even from a functional standpoint, can be used. One drive may be perfectly satisfactory in a low-speed vehicle, yet extremely dangerous in a high-speed one. One drive may be entirely satisfactory in a transporting vehicle, but inadequate in a towing vehicle. There are a few things, however, which in a very general way may be considered to be desirable in most cases. Some of these are very controversial and must be judged in the light of the ultimate application to a vehicle.

The chart shown in Fig. 11 has been prepared as a guide in evaluating general characteristics of various steering systems. The characteristics which are desirable in most applications are cross-hatched. It must be understood that such an evaluation, shown here for illustrative purposes, can be intelligently made only with a full knowledge of the details of the control system as well as the requirements of the vehicle. In addition to those shown, many other factors such as cost, ruggedness, serviceability, momentary power disconnect, mechanical efficiency, and so forth, must be considered in the final analysis. The chart has been prepared to compare the general properties of the mechanical system, rather than the sales features resulting from design refinements. Among the many factors to be considered are several which are associated with power delivery or absorption during steering. Power delivery is a most important item, and increases in magnitude with an increase in vehicle speed. Therefore, the relative importance of minimum power losses during steering should be judged less critically in a slow-moving vehicle than in a fast-moving one, but is of significant importance in all.

The foregoing discussion outlines in a general way some of the idiosyncrasies of track vehicles, and the forces which the steering mechanism must control. The ideal mechanism would take these varied forces, digest them efficiently, and force

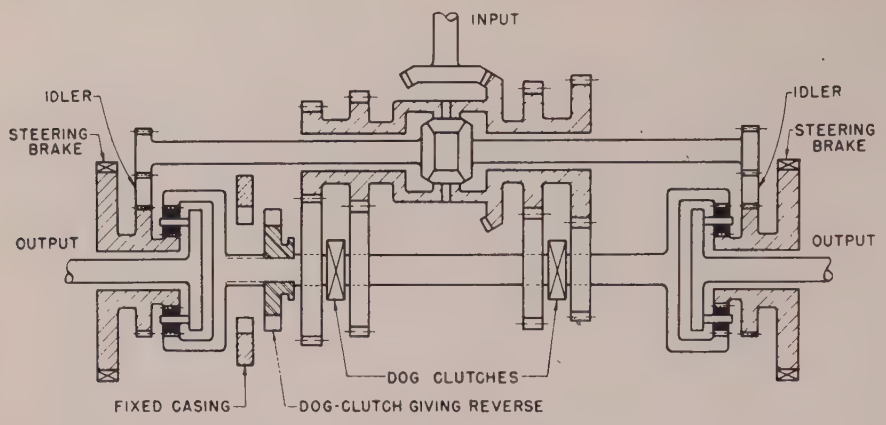


Fig. 10 – Like cross drive, the Merritt Brown system combines speed-change transmission with steering mechanism

the tracks to respond consistently to a given signal from the operator under all operating conditions. Such a mechanism is illustrated in Fig. 12. This mechanism, except for power transmission losses, would not dissipate power as it has no brakes, and would therefore deliver the maximum power to the tracks. Torque reversals due to reverse track loads would result in a circulation of power from one track to the other. This type of system is called a regenerative one, and several existing steering mechanisms have a point in their operating range at which they are completely regenerative. The controlled differential in its tightest turn, the planet power during its geared turn, and the spot turn during its geared turn, are regenerative. At high vehicle speeds, the power differential between tracks may be considerably in excess of the power of the engine, and a system which depends upon brake slippage to induce the speed differential would cause the vehicle to stop on a sustained turn.

A second important feature of this ideal steering mechanism is that sprocket speeds are controlled independently of sprocket loads. Thus, reverse steering common on friction-controlled steering mechanisms does not exist.

The third characteristic is that a continuous power train connection is made, and this is the most important factor in eliminating reverse or uncontrolled steering, as well as in maintaining mobility under difficult conditions of traction.

How close existing mechanisms approach the ideal performance can be evaluated by examination of the power characteristics and the design details of the control mechanisms. These details may be of extreme importance in the controllability of the machine. For instance, when making the transition between the speed ranges of the planetary, with a TD-24 planet power steering unit, there is a theoretical power disconnect but, in actual practice, the rotational inertia of the brake discs provide the stabilizing influence required to resist a change in track speeds during the split second shift.

It is possible by making a sufficient quantity of assumptions to reduce the steering characteristics of a vehicle to mathematical analysis. Applying the deductions, however, to anything except generalities is extremely risky if any degree of accuracy is expected. One such approach to mathematical analysis can be considered. In Fig. 13 is illustrated a vehicle turning about pivotal point A. The mean turning radius is R. In order to negotiate this turn, it is necessary that driving forces be applied to the tracks, resulting in some degree of slip. In this particular illustration, it is considered that the outer track, driving forward, has slipped opposite to the direction of travel of the vehicle, and the inner track, providing a resisting force, has slipped in the direction of the motion of the vehicle. The instantaneous centers of rotation of the inner and outer track shoes, are shown by a_i and a_o respectively.

It is evident from the vector diagram that:

$$V_{og} = V_o - V_{os} \tag{1}$$

and:

$$V_{ig} = V_i + V_{is} \tag{2}$$

and:

$$\frac{V_{ig}}{R - \frac{G}{2}} = \frac{V_{og}}{R + \frac{G}{2}} \tag{3}$$

Reducing this produces:

$$R = \frac{G}{2} \frac{V_o - V_{os} + V_i + V_{is}}{V_o - V_{os} - V_i - V_{is}} \tag{4}$$

For example, assume a free vehicle operating as follows:

$$G = 6 \text{ ft}$$
$$V_o = 3 \text{ mph}$$
$$V_i = 2 \text{ mph}$$

Inner track slips forward and outer track backward:

$$V_{is} = V_{os} = 0.10$$

$$R = \frac{6}{2} \frac{3 - 0.10 + 2 + 0.10}{3 - 0.10 - 2 - 0.10} = 3 \times \frac{5}{0.80} = 18.8 \text{ ft}$$

Now assume that this vehicle is pulling a load and conditions are as follows:

$$G = 6 \text{ ft}$$
$$V_o = 3 \text{ mph}$$
$$V_i = 2 \text{ mph}$$

Drawbar pull will tend to make outer track slip more and inner track slip less.

$$V_{is} = 0$$

$$V_{os} = 0.20$$

$$R = \frac{6}{2}\frac{3 - 0.20 + 2 + 0}{3 - 0.20 - 2 - 0} = 3 \times \frac{4.8}{0.80} = 18.0 \text{ ft}$$

Again let the load push the vehicle so that:

$$G = 6 \text{ ft}$$

$$V_o = 3 \text{ mph}$$

$$V_i = 2 \text{ mph}$$

Inner track slip is increased and outer track slip is decreased.

$$V_{is} = 0.20$$

$$V_{os} = 0$$

then:

$$R = \frac{6}{2}\frac{3 - 0 + 2 + 0.20}{3 - 0 - 2 - 0.20} = 3 \times \frac{5.2}{0.8} = 19.5 \text{ ft}$$

It will be noted that the turning radius without longitudinal slip would be

$$R = \frac{6}{2} \times \frac{5}{1} = 15 \text{ ft.}$$

The above examples are hypothetical, but are in general representative of the degree of turning radii variation experienced on level ground operation. Under the conditions of sustained turning with a fixed drawbar, the drawbar pull may increase the turning radius by producing a stabilizing moment as the hitch point moves outward from the center of turn, thus increasing the resistance to turning. This is generally the case in short radius turns.

It will be seen that for low levels of longitudinal slip on the track, the direction of forces has little effect on the actual turning radius with constant track speeds. This is borne out in practice where fixed ratio steering demonstrates consistent turning under normal operating conditions. On an extreme sidehill operation, on tight pivot turns, and with nonuniform track loading, track slip may become more significant and appreciably affect controllability of the vehicle. Several approaches to analytical reduction of track to ground forces and resultant motion have been suggested, but as track proportions, pad designs, ground conditions, and external vehicle forces vary so radically, performance predictions are extremely unreliable. It will suffice to say that in general, a high ratio of gage width to track length produces a readily steerable unit with low longitudinal slip due to turning, while an extremely low ratio would preclude short turns and demonstrate extremely high longitudinal slip due to turning. Grousers, which are highly aggressive longitudinally to the track, and less aggressive laterally to the track, aid in steering but at the same time lessen the vehicle's ability to operate on side slopes, or to control skidding when executing high-speed turns. Side slippage, either on sidehill operation or during high-speed turning, can easily offset the advantages of low power consumption during the turn. In the same way, extremely short tracks, by their very characteristic of low resistance to turn and consequent low power consumption, lose their ability to some extent to resist jackknifing when being pushed by a trailed load, especially when resistance steering is used or a power disconnect occurs during steering. It is evident then that the track vehicle is inherently handicapped by the fact that some of the same characteristics which provide stability guidance of the vehicle also increase the difficulty of turning, but to a lesser extent in short track vehicles. In operations on soft terrain, where resistance to steering is usually highest, grouser design has less effect on the power consumption required to negotiate a turn, as penetration is usually sufficient for the edge of the pad to be the governing factor in resistance to side slippage.

In considering what happens to a pad when it is laid in the ground during a turn, reference is again

	PIVOT TURN	AXIS TURN	REGENERATIVE STEERING	REVERSE STEERING WHEN PUSHED	RAD. OF TURN DET. BY FRICTION LEVEL	POWER TO BOTH TR'KS DURING TURN	OUTER TRACK SPEEDS UP
STEERING CLUTCH	YES	NO	NO	YES	YES	NO	NO
CONTROLLED DIFFERENTIAL	NO	NO	PART TIME	NO	YES	YES	YES
SPOT TURN	YES	NO	PART TIME	NO	PART TIME	PART TIME	NO
PLANET POWER	YES	NO	PART TIME	NO	PART TIME	PART TIME	NO
CROSS DRIVE	YES	YES	PART TIME	NO	PART TIME	YES	YES

Fig. 11 – Chart evaluating general characteristics of various steering systems with cross-hatching indicating those desirable in most applications

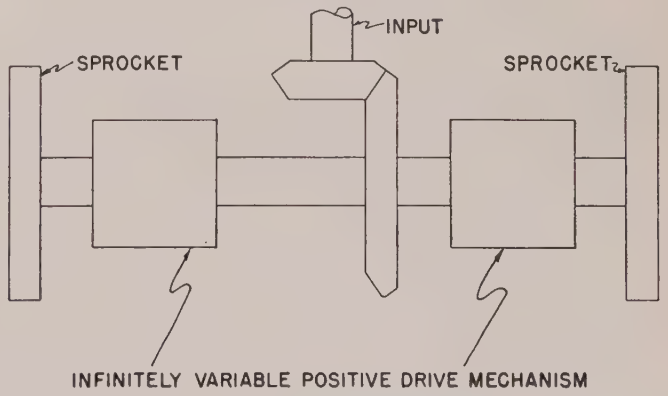


Fig. 12 – Ideal steering control mechanism would force tracks to respond to signal from operator at all times

made to Fig. 2B which illustrates the path followed by a pad from the time it is laid down until it is picked up. While it has been forced to move due to track slippage and angular velocity, it is not moving with the tractor. In effect, the pads on the ground have a forward velocity equal to the slip speed, and are rotated at the angular velocity of the tractor during the turn. From this it can be concluded that the power to turn is proportional to the angular velocity of the vehicle and to the slip speed of the grousers, but not to the forward speed of the vehicle.

The approximate magnitude of such steering losses can now be examined. Experience has shown that slip is of a low order under conditions where power losses are maximum (normal turn at high speed). For example, in considering a track being rotated about its center of pressure, it is assumed that the center of pressure is coincident with the center of area of the track print, and that the track length is sufficient so that the effect of angular resistance of the pads is small in proportion to the lateral resistance to sliding. Then the torque to rotate the tracks on the ground can be demonstrated to be $W \times \frac{1}{4}L \times \mu$, where W is the weight on the tracks, L is the length of the track on the ground, and μ is the coefficient of friction (Fig. 14).

It is fully realized that this oversimplified approach does not take into account the following factors:

(a) Longitudinal slip and its effect on the magnitude and direction of the frictional forces.

(b) Increase in the indicated value of μ with increase in total movement of the grouser over the ground.

(c) Angular creep in the track due to play in the suspension and the track joints.

(d) Resistance to rotational displacement of the grouser in the ground.

However, except for academic purposes, this approach renders satisfactory results. It is noted that according to this concept, steering is limited to a maximum ratio of track length to gage width of 2. Like many other approximations, this one is not useful when extended outside the field of normal practice, but the indication that large ratios of track length to gage width produces difficult steering is perfectly true, and very long tracks become capable of only limited control over the vehicle direction of travel.

The value of μ can be assumed to be anything between 0.1 and 1.0 depending upon ground conditions, but since average cross-country conditions, and maximum power to turn is being computed, 0.9 will be chosen. The following example will be examined: $W = 40{,}000$ lb, $L = 100$ in., $R = 50$ ft, $G = 80$ in., vehicle speed, 6 mph.

$$\text{Moment} = M = \frac{WL\mu}{4} = \frac{40{,}000 \times 100 \times 0.9}{4} = 900{,}000 \text{ in.-lb}$$

$$\text{Tractor angular velocity} = \frac{6 \times 5280}{2\pi \times 50 \times 60} = 1.68 \text{ rpm}$$

Power loss due to transverse slippage of track shoes

$$= \frac{900{,}000 \times 1.68}{63{,}025} = 24 \text{ hp}$$

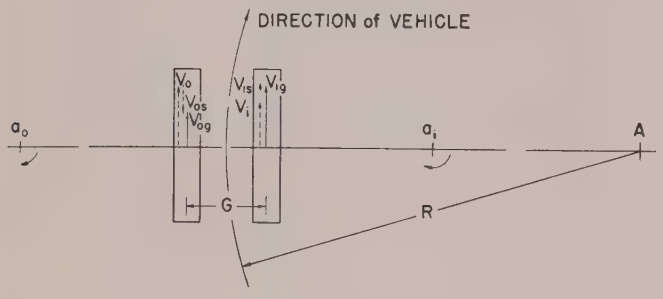
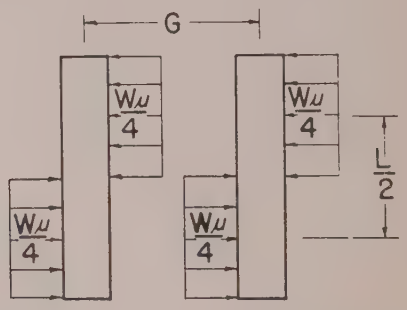


Fig. 13 – Illustrating a vehicle turning about pivotal point A, in which outer track, driving forward, has slipped opposite to direction of travel, and inner track, providing resisting force, has slipped in direction of motion of vehicle.

G = Track gage
A = Center of turn of vehicle
a_o = Instant center of outer track
a_i = Instant center of inner track
V_o = Sprocket pitch line velocity outer
V_i = Sprocket pitch line velocity, inner
V_{os} = Outer track slip velocity
V_{is} = Inner track slip velocity
V_{og} = Outer track velocity
V_{ig} = Inner track velocity



$$\text{RESISTANCE TO TURNING} = 2 \times \frac{W\mu}{4} \times \frac{L}{2} = \frac{WL\mu}{4}$$

$$\text{AVAILABLE MOMENT TO INDUCE TURN} = \frac{W\mu}{2} \times G$$

LIMITING CONDITION IS WHEN

$$\frac{W\mu}{2} G = \frac{W\mu}{4}$$

$$G = \frac{L}{2}$$

Fig. 14 – Approximate magnitude of steering losses indicates that slip is of low order under normal turn at high speed

If this machine were to make a pivot turn of R = 5 ft at 6 mph, the moment remains constant, but the angular velocity becomes 16.8 rpm, and the power consumption would be 240 hp.

To roughly calculate the power distribution of the above machine during the 50 ft radius turn, when the outer track slip is 0.12 mph backwards, and the inner track slip is 0.12 mph forward, or

$$\frac{V_{ig} + V_{og}}{2} = 6 \text{ (mean vehicle speed)} \qquad (5)$$

$$V_{is} = V_{os} = 0.12$$

then using (4):

$$R = \frac{G}{2} \frac{V_o - V_{os} + V_i + V_{is}}{V_o - V_{os} - V_i - V_{is}}$$

and reducing to:

$$V_o (2R - G) - V_i (2R + G) = 2R (V_{os} + V_{is}) + G (V_{is} - V_{os})$$

from (1): $V_o - 1.143\, V_i = 0.257$

from (2): $V_o = V_{og} + 0.12$

from (5): $V_i = V_{ig} - 0.12$

$V_o - 0.12 + V_i + 0.12 = 12$

$V_o + V_i = 12$

Solving simultaneously:

$V_o = 6.51$ mph

$V_i = 5.49$ mph

Neglecting rolling resistance, the sprocket forces to induce the above turn are $\frac{900{,}000}{80} = 11{,}250$ lb (from the second equation in Fig. 14).

This is a driving force on the outer track and a driven force on the inner track.

Outer track hp is $\frac{11{,}250 \times 6.51}{375} = 195.5$ delivered.

Inner track hp is $\frac{11{,}250 \times 5.49}{375} = 164.5$ absorbed.

It will be noted that with a 100% efficient regeneration of power from the inner track to the outer, the power dissipation is 195.5 —164.5 on 31 hp, as compared to 24 previously calculated. This calculation, however, includes longitudinal slip hp of $2 \times \frac{11{,}250 \times 0.12}{375} = 7.2$ hp.

If the power transmission efficiency is 0.9 and a regenerative system of steering were used, the engine requirement would be (again neglecting rolling resistance):

Outer sprocket input:	195.5/0.9 = 217 hp
Delivered to outer sprocket from inner sprocket:	164.5 x 0.9 x 0.9 = 133
Difference made up by engine:	84

If a nonregenerative system were used, the full 217 hp must be supplied by the engine, while the inner sprocket hp is being dissipated by brake slippage or other power absorbing means. This is, of course, only one case, and this ratio will not hold under other circumstances.

For example, if on the above conditions is superimposed a drawbar pull of 20,000 lb, and it is assumed that the slips change correspondingly, and $V_{is} = 0$, $V_{os} = 0.18$, then V_o will become 6.58, and V_i will become 5.42.

The outer sprocket force will become 11,250 + 10,000 = 21,250 lb.

The inner sprocket force will be 11,250 —10,000 = 1,250 lb.

Outer sprocket hp $= \frac{21{,}250 \times 6.58}{375} = 373.$

Inner sprocket hp $= \frac{1{,}250 \times 5.42}{375} = 18.1.$

In the above case, there is little advantage to a regenerative steering system. In actual operation, these and many other combinations are encountered which make the regenerative system more productive, but the above examples illustrate the important difference between transport vehicles and drawbar machines.

In many cases the power requirement will exceed the available engine power, and the power to turn must be made up from the inertia of moving parts. For this reason, a system which steers by slowing the inner track without speeding the outer track is at an advantage as more inertia is available at the critical moment. In addition, the decrease in average vehicle speed reduces the power requirement on drawn equipment during the turn.

As a steering system is basically a speed control system, it is only natural that a regenerative system could also serve to efficiently control the general vehicle speed. In the planet power system, the high-low ranges generally used for steering also serve as a power shift, thus greatly reducing the operator's effort in downshifting and upshifting to traverse a particularly difficult spot. If the ideal system could be designed, there would be no need for a speed-change transmission ahead of it, as it would offer a complete selection of vehicle speeds as well as turning radii.

While analytical methods are useful in understanding the mechanics by which various effects are produced by a steering system, actual operation often reveals greater differences in productivity than would be suspected from the calculations. The increase in mobility of a tractor which is obtained by maintaining power to both tracks while working in soft terrain or an extreme sidehill, is difficult to demonstrate on paper, but very evident to the man who is operating the tractor.

The designer today, creating track-type machines for the future, is confronted with more exacting requirements of performance, ease of control, and safety, and is closely examining the gains to be made by properly matching the steering system to the vehicle. This discussion is not designed to promote any particular system as the most satisfactory for any particular application, but rather to point out some of the factors which must be considered in making a selection of existing types or in creating new ones.

Supercharged Diesel Performance

THE maximum torque obtainable by supercharging the 4-stroke cycle diesel engine is limited by the durability of the parts and the efficiency of the complete system for crowding more and more air into the cylinders. Preignition, detonation, and abnormal spark gap resistance do not haunt the supercharged diesel engine.

The task of providing increased engine durability, like taxes, always will be with us. Supercharging markedly increases mechanical and thermal loads on critical engine parts.

The air induction system of the supercharged diesel is essentially a two-stage air compressor. Someone in an overenthusiastic moment labeled the first compressor stage a supercharger. The engine pistons function as the second stage. The

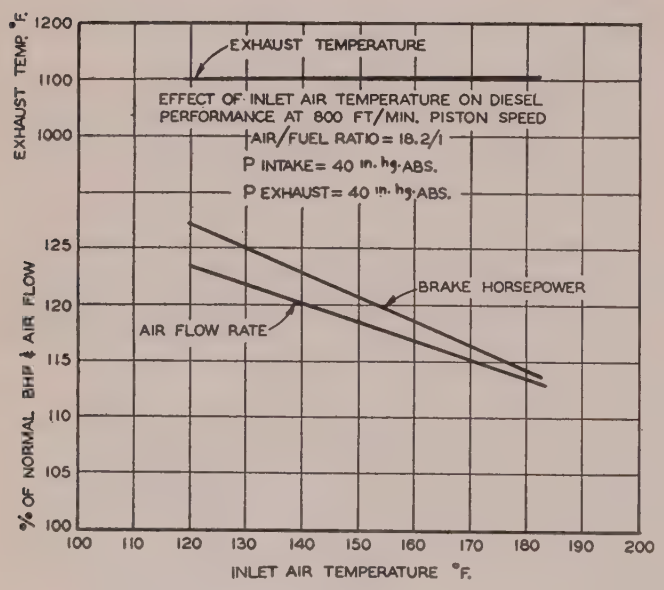


Fig. 1 – Effects of inlet temperatures with low piston speed

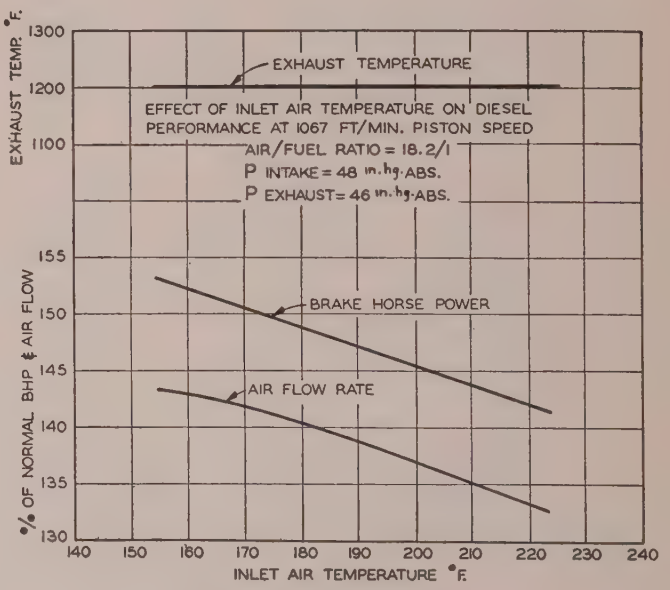


Fig. 2 – Characteristics at mean piston speed of 1067 fpm

versus Intake and Exhaust Conditions

Lloyd Johnson, Caterpillar Tractor Co.

This paper was presented at the SAE Annual Meeting, Detroit, Jan. 16, 1952.

usual problem facing an engine manufacturer is to select the most suitable types and sizes of superchargers to use with a line of engines. A knowledge of the characteristics of the engine as a second-stage compressor has been found most useful to Caterpillar engineers in the evaluation of proposed supercharging arrangements.

A choice had to be made between a prototype single-cyl test engine and a multicyl model for this study of the response of the engine to variations in inlet and exhaust conditions. The single-cyl test engine is a very useful laboratory tool, easy to modify and economical to operate. For the tests in question certain objections to its use arise. Do the strongly pulsating air and exhaust gas flows cause ram or suction effects which might appre-

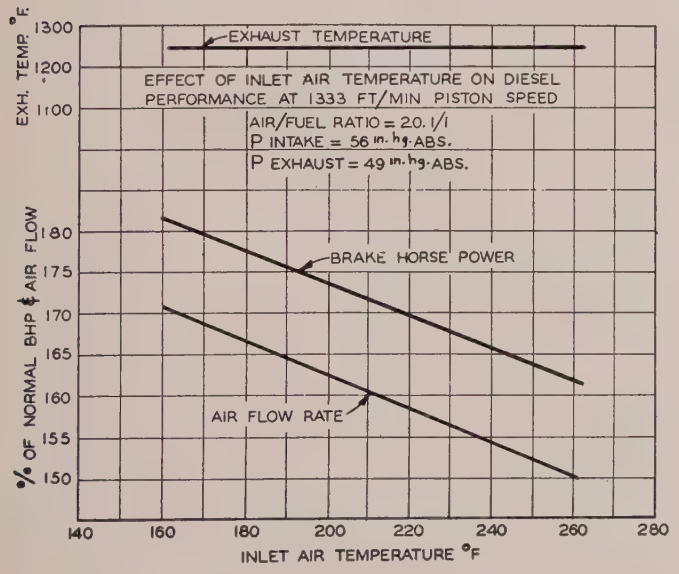


Fig. 3 – Characteristics at mean piston speed of 1333 fpm

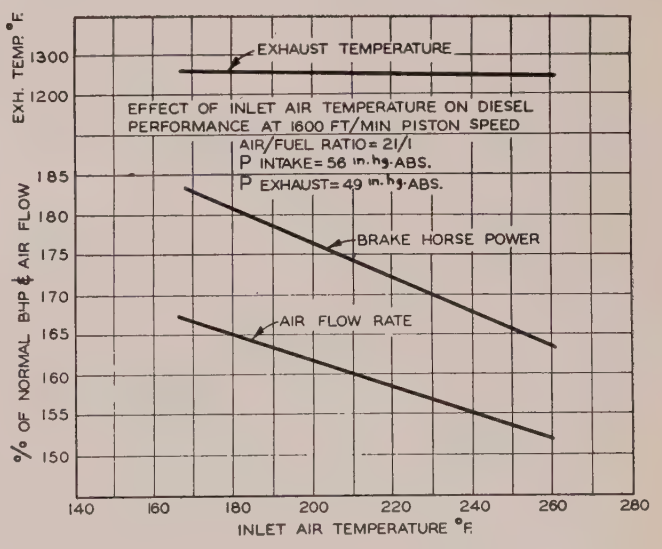


Fig. 4 – Characteristics at mean piston speed of 1600 fpm

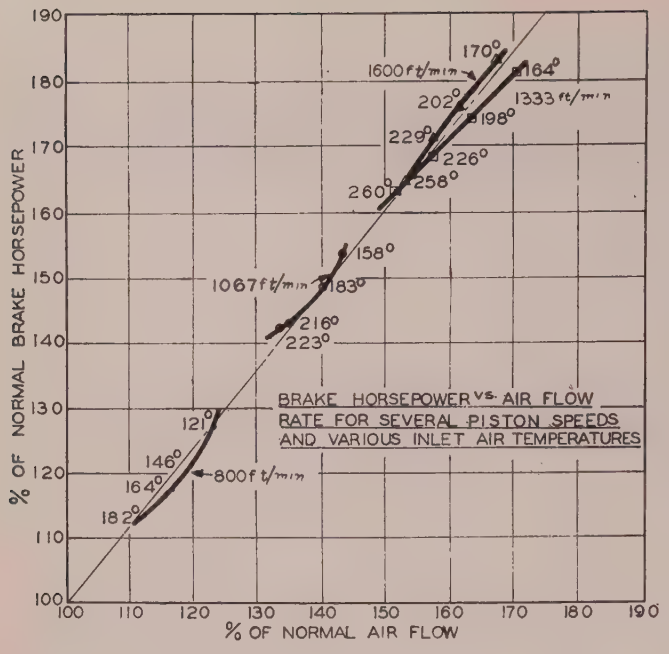


Fig. 5 – Cross-plot of data shown on four previous figures

ciably effect airflow and scavenging? To what degree are airflow measurements effected by pulsations at the metering station? Are multicyl engine manifold conditions duplicated? The magnitude and significance of such errors can be determined by either calculation or test. However, such questions were conveniently avoided by running supercharging tests on a multicyl engine which is typical of the engines for which supercharging is contemplated.

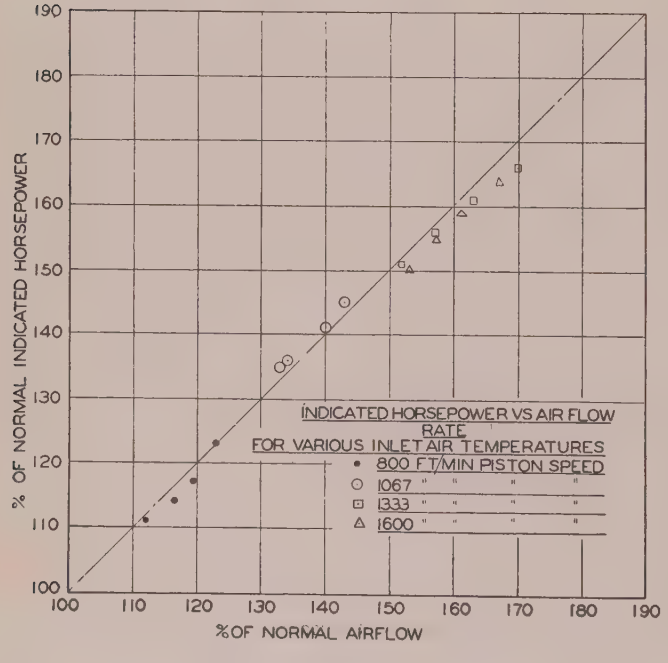


Fig. 6 – Effects of variations in combustion efficiency, temperature, and speed

There exist several suitable sources of air, at variable pressures and in different quantities, to feed the engine. The air can be taken from the factory's compressed-air system. This source is often used to provide boost to single-cyl engines running on endurance tests. Compressed air from this source would have to be metered in a closed system. Varying demands for air elsewhere on the line may cause annoying pressure fluctuations.

An independent, engine-driven compressor, while involving more equipment, can be used in conjunction with airflow metering equipment that has been used for years to measure airflow to unsupercharged engines. An adjustable bypass valve which permits recirculation through this compressor, provides a convenient vernier adjustment on flow and is particularly useful when a low rate of airflow is desired. A heat exchanger, between the compressor and engine under test, provides means to regulate air temperature.

Such a setup would be incomplete without a valve in the exhaust line to throttle exhaust flow to simulate the back pressure typical of exhaust turbine-driven superchargers. Properly located manometer connections and thermocouples and the usual dynamometer and fuel-flow measuring equipment complete the setup.

With inlet and exhaust conditions variable, some suitable fixed parameter had to be selected. Comparisons at a selected fixed horsepower would reflect variations in combustion efficiency with changes in available air. While such effects are interesting and informative, they do not indicate the potential of the engine. The density and color of exhaust smoke unfortunately are but a very rough indication of combustion efficiency, although it is often made a limiting factor for satisfactory service. In some instances our experience has led us to avoid exhaust temperatures above a certain valve measured in a certain place. This is very empirical and hardly a fair basis for comparing supercharging systems except as it affects exhaust turbine performance.

The absolute maximum power output, measured by one test procedure or another, while important, is used for such short periods, if at all, as to be of secondary importance. The power rating which an engine can sustain for hours at a time without damage, is important to owners of industrial and construction machinery. The limits on such sustained power output are cleanliness of combustion, mechanical loading of parts, and degree of thermal loading of such critical parts as pistons, rings, valves, and lubricating oil.

A study of experience with unsupercharged Caterpillar diesel engines revealed a reasonable constancy in combustion cleanliness and thermal levels with a given air/fuel ratio. Air/fuel ratio

THIS paper presents results which will answer many of the problems facing an engine manufacturer in the selection of the most suitable types and sizes of superchargers to use with a line of engines. Although performance curves of production model diesels are available, decisions are still needed in choosing peak supercharging pressures, drive means, and size and effectiveness of intercoolers, if any.

The author describes the use of a typical model to determine response to variation in intake and exhaust conditions, resulting in data which will assist in evaluating engine potentials with any system of supercharging. Thus, supercharger selection for a particular line of engines is aided by knowledge of engine characteristics as a second-stage compressor.

The Author

LLOYD JOHNSON (M '42) is a staff engineer in the Research Department of Caterpillar Tractor Co. Since 1945 he has worked on the design, development, and evaluation of experimental types of internal-combustion engines. Mr. Johnson joined Caterpillar's engineering graduate training program following his graduation from Iowa State College.

was chosen as a fixed parameter for each test. Performance at the selected air/fuel ratio is considered a reasonably practical measure of the usefulness of the engine under the intake and exhaust conditions selected for evaluation.

At the lower mean piston speeds an air/fuel ratio of 18 to 1 allowed operation just on the clean side of the upward swing in exhaust smoke density. At the higher piston speeds, air/fuel ratios leaner than 18 to 1 were selected. This was considered necessary to avoid exhaust temperatures over 1300 F, measured at the exhaust manifold outlet. Sustained operation at higher thermal loading of critical engine parts would result in inadequate service life by our standards. A temperature of 1300 F is on the high side of the limit that would be necessary for adequate service life of exhaust-driven turbines employing time-proved alloys of reasonable cost.

Mean piston speed, rather than rpm, has been chosen as a measure of velocity factors. This facilitates application of the results of such tests to engines in a line of several cylinder sizes. Since Caterpillar is concerned with supercharging for its line of essentially similar engines of various displacements, airflow and horsepower figures have been plotted as nondimensional ratios. The base value of airflow and horsepower in each instance has been the flow and bhp, respectively, of the same test engine operating unsupercharged at the same air/fuel ratio and mean piston speed. Inlet and exhaust were maintained at standard laboratory conditions of 29.7 in. Hg barometric pressure and 90 F inlet air temperature. The temperature of 90 F has been selected for routine performance tests by Caterpillar research as being a convenient inlet air temperature to maintain at any time of year by use of induction pipe heat exchangers. A temperature of 60 F air is costly to obtain in large volumes during a 95 F summer day. Besides, 60 F air makes things too easy for an engine on endurance test.

Discussion of Results

The five independent variables in such a test are inlet air temperature, mean intake manifold pressure, mean exhaust manifold pressure, air/fuel ratio, and mean piston speed. In Fig. 1, the effects of inlet temperature on per cent airflow, per cent bhp, and exhaust temperature are shown for a low piston speed of 800 fpm. Intake and exhaust manifold pressures are equal but above atmospheric. Note the practically constant exhaust tem-

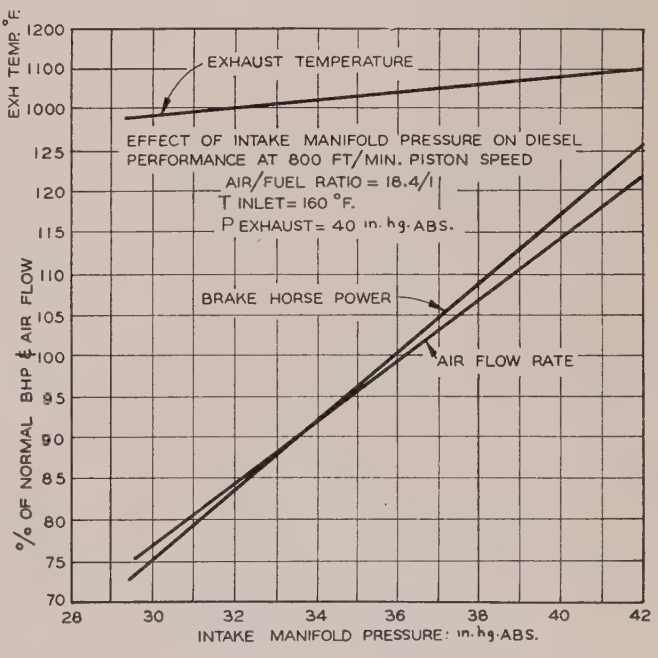


Fig. 7 – Typical effects of increased boost

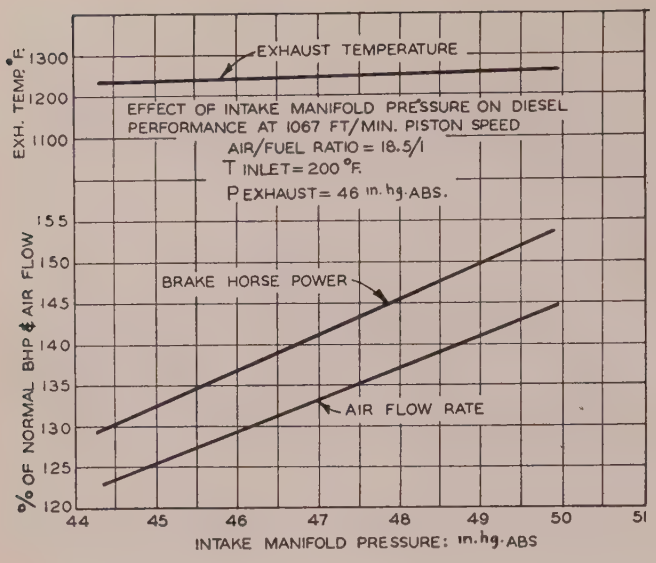


Fig. 8 – Effect of increased boost at 1067 fpm

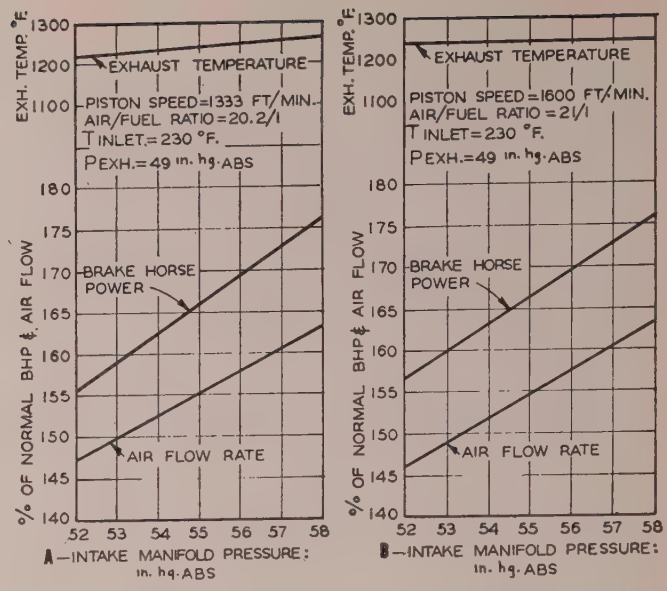


Fig. 9 – Effects of boost at 1333 and at 1600 fpm

perature, while bhp changes as a direct function of airflow.

Figs. 2, 3, and 4 show similar characteristics at mean piston speeds of 1067, 1333, and 1600 fpm. Note in each case that the exhaust temperature does not vary in spite of the varying bhp.

Fig. 5 is a cross-plot of the data shown in Figs. 1 through 4 showing, as we would expect, bhp changing about in proportion to airflow. Indicated hp was computed by adding to bhp, the engine friction hp determined for each speed and manifold condition. A plot of per cent airflow, as shown in Figs. 6, reveals some departure from a direct proportion which may be attributed to variations in combustion efficiency occurring with variations in induction air temperature and engine speed.

The next factor varied was intake manifold pressure. Fig. 7 shows, for 800 fpm mean piston speed, the increasing airflow, bhp, and small increase in exhaust temperature which are typical of the effect of increased boost. Figs. 8, 9A and 9B show similar characteristics at 1067, 1333, and 1600 fpm. Note that at 1600 fpm exhaust temperatures varied but slightly with increased load. Fig. 10 again cross-plots data to show the relation between bhp and air flow when airflow change is the result of changes in intake manifold pressure. Since each segment shown is for a different mean piston speed, the 100% figure for each segment differs. This eliminates the effect of speed on volumetric efficiency from this cross-plot.

Figs. 11, 12, 13, and 14 show the effect of exhaust manifold pressure on airflow, bhp, and exhaust temperature at the same four piston speeds. Again exhaust temperature remains unchanged, for a given speed and air/fuel ratio, while increasing back pressure restricts airflow and causes a drop in bhp. Fig. 15 cross-plots bhp against airflow as affected by exhaust back pressure. Whereas in the other cross-plots, the several curve segments fell fairly close to a common line from the 100% ordinates, the exhaust back pressure variation caused a reduction in bhp greater than can be attributed to airflow change alone. This additional loss of power resulted from the extra work the engine had to do during the exhaust stroke of the engine when exhausting at higher pressures.

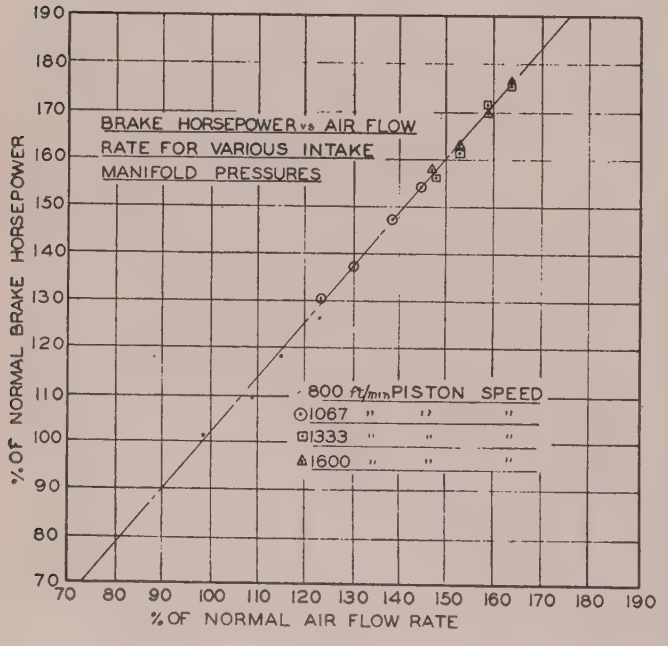


Fig. 10 – Result of changes in intake manifold pressure

To determine more accurately what proportion of the loss in power was due to extra pumping load, engine friction was determined at each speed with and without back pressure. Fig. 16 is a plot of the increase in fmep for increase in absolute exhaust manifold pressure, plotted for four piston speeds. On the average fmep increased 0.6 psi, per psi increase in back pressure. The increase in fmep would doubtless be greater were it not that during the early part of the exhaust period the back pressure increases useful work delivered to the piston. Fig. 17 shows indicated hp versus airflow for variations in exhaust back pressure. Indicated hp, arrived at by adding fhp to bhp, does not show the extra work lost during the exhaust period. The curve sections fall nearer to a common line.

The remaining variable not yet reported is air/fuel ratio. Fig. 18 shows the effect of air/fuel ratio on airflow, bhp and exhaust temperature at 1333 fpm piston speed. The increase in airflow with reduction in rate of fuel flow can probably be attributed to a reduction in temperature of residual gases and of combustion-chamber and cylinder-wall surfaces. The same trends are shown in Fig. 19, which shows the effect of reduced fuel rate at 1600 fpm piston speed. A study of the relation between airflow and bhp means little when air/fuel ratio is changed in a diesel engine. However, the variation in bhp as a function of fuel flow shown in Fig. 20 reveals that overall efficiency decreased with increased output through the range of the air/fuel ratios studied. The slope of the curves shows fuel requirements to increase more rapidly than the bhp. However, the overall efficiency of the engine with 56 in. Hg boost and 49 in. Hg back pressure can be seen to be better than that of the unsupercharged engine. The total per

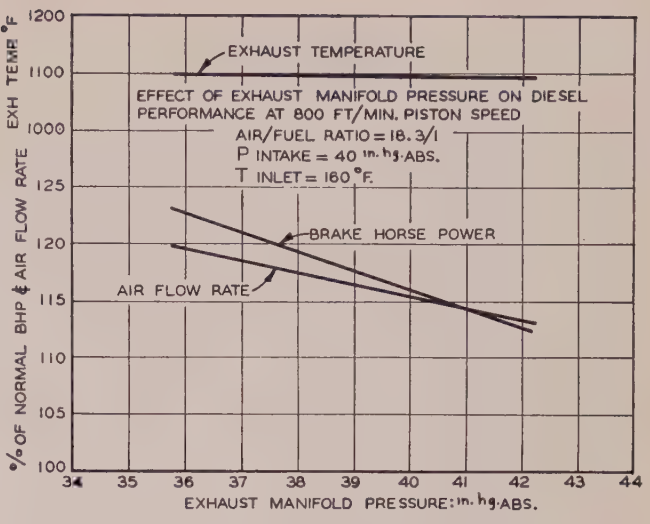


Fig. 11 – Effect of exhaust manifold pressure at 800 fpm

cent increase in fuel requirements is less than the total per cent increase in bhp.

If the variation in airflow, bhp, and exhaust temperature, as affected by each independent variable, can be plotted as straight lines without excessive error, the effect of manifold conditions, air/fuel ratio, and exhaust temperatures at a given piston speed can be expressed by simple linear algebraic equations. (See appendix for definition of symbols.)

$$w/w_a = a_1 p_i + b_1 t_i + c_1 p_e + d_1 (A/f) + e_1$$

$$\text{bhp}/\text{bhp}_a = a_2 p_i + b_2 t_i + c_2 p_e + d_2 (A/f) + e_2$$

$$t_e = a_3 p_i + b_3 t_i + c_3 p_e + d_3 (A/f) + e_3$$

For example, at 800 fpm mean piston speed, the equations for airflow, bhp, and exhaust tempera-

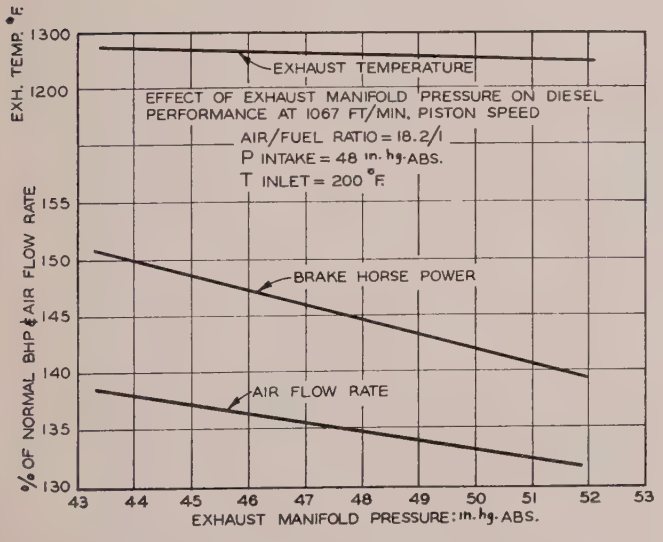


Fig. 12 – Effect of exhaust manifold pressure at 1067 fpm

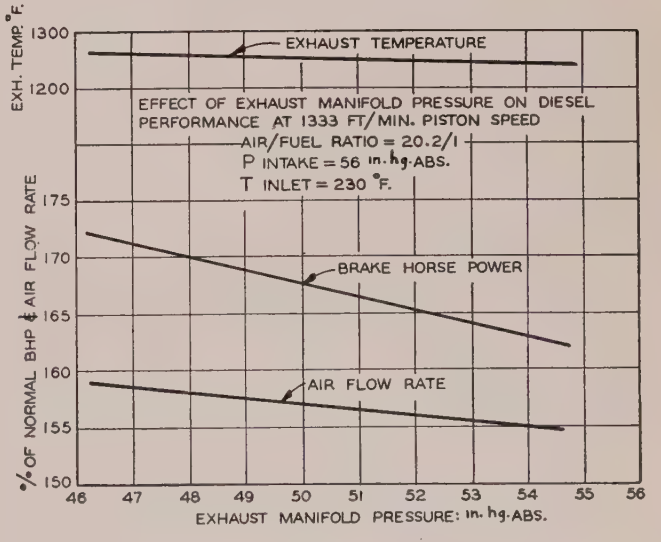


Fig. 13 – Effect of exhaust manifold pressure at 1333 fpm

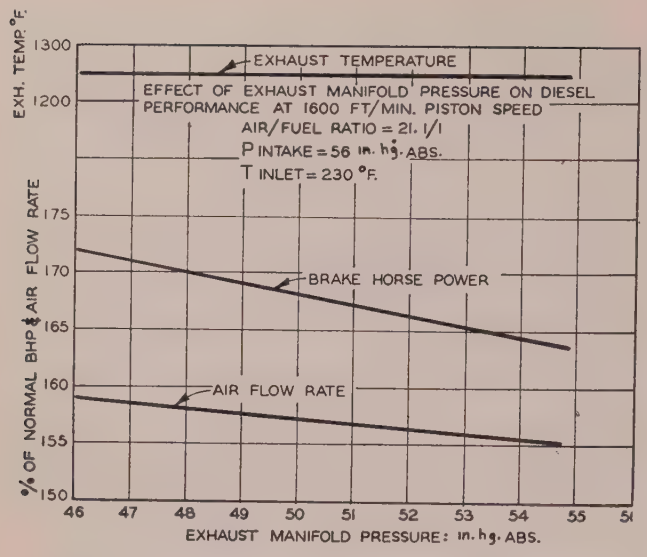


Fig. 14 – Effect of exhaust manifold pressure at 1600 fpm

ture as derived from the characteristic slope lines are:

$$\frac{w}{w_a} = \frac{3.7p_i - 0.017t_i - p_e + 0.67(A/f) + 23}{100}$$

$$\frac{\text{bhp}}{\text{bhp}_a} = \frac{4.2p_i - 0.021t_i - 1.7p_e - 2.7(A/f) + 99.2}{100}$$

$$t_e = 10p_i - 50\,(A/f) + 1605$$

Similar equations can be written for the other

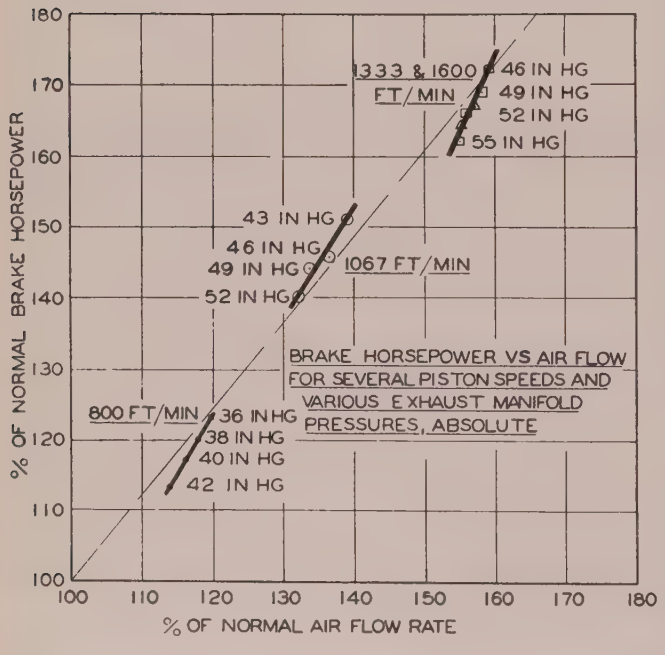


Fig. 15 – Cross-plot showing effects of exhaust back pressure

piston speeds. A temptation exists to combine the equations for several speeds into one equation with piston speed included as a fifth independent variable. This can be done if the effect of speed is nearly linear. With small valve overlap and short valve opening duration, volumetric efficiency was found to be almost a linear function of speed. Unfortunately combustion efficiency varies with changes in engine speed because of injection and combustion characteristics. For sufficient accuracy, the combined equations for airflow, bhp, and exhaust temperature involve second and third degree functions. Often it is simpler to use the equations for a single piston speed than to use the more complex equations worked out to cover all speeds.

With such curves and equations, an analysis of engine response to any variation in supercharger characteristics is simplified. The reduction in available engine power caused by back pressure incident to use of an exhaust turbine must be taken into account, just as must power used to drive a gear-driven compressor. The degree of boost that can be employed without adequate intercooling can be accurately evaluated on the basis of these characteristic curves. The rather minor influence of factors other than speed and air/fuel ratio on exhaust temperatures encourages us to greater use of supercharging as a means to increase power with but minor increases in thermal loading.

In using such characteristic curves to analyze various engine models, we must be aware that certain dissimilarities between engines are likely to exist. Variations in length of jacketed exhaust port, in exhaust manifold shape, and in manifold location can result in variations in per cent heat loss from exhaust gases after they leave the cylinder. A significant change in valve event timing or in port shape can cause changes in airflow. This has been shown by extensive investigations, such as those by Taylor, Livengood, and Eppes, of Massachusetts Institute of Technology. The use of constant air/fuel ratio as a measure of performance capabilities of the diesel becomes subject to question if large valve event overlap is used. As pointed out by P. H. Schweitzer of Pennsylvania State College, measure of intake airflow can be a misleading measure of useful volumetric efficiency when a part of the air can short circuit to the exhaust.

Conclusion

The performance curves of production model diesels are available to anyone, from the engine manufacturer's sales department. However, before such ratios are published, many decisions have to be made in selecting type and size of supercharger, drive means, peak supercharging pressures, and size and effectiveness of intercoolers, if any. Using a typical model in a company's engine line to deter-

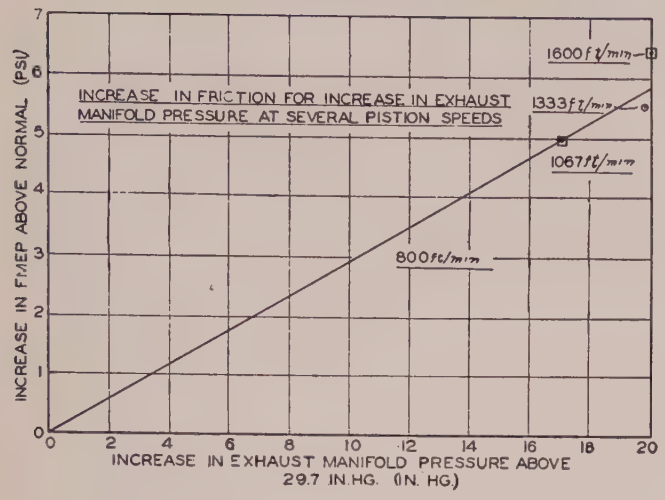


Fig. 16 – Plot for four piston speeds

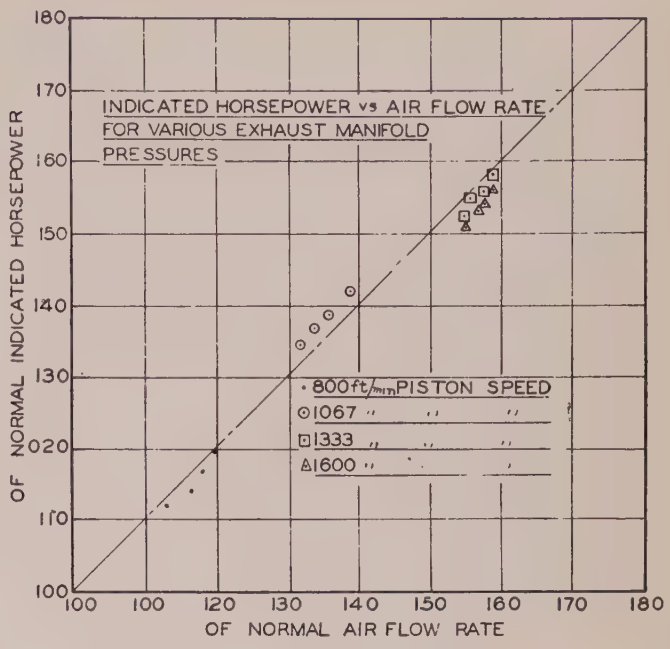


Fig. 17 – Variations in exhaust back pressure

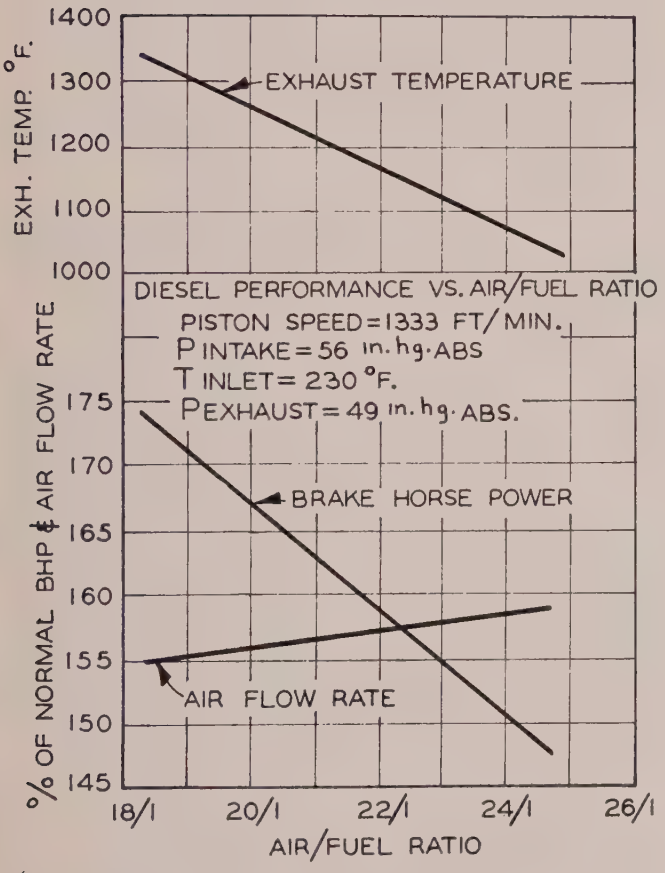


Fig. 18 – Air/fuel ratio variable at 1333 fpm

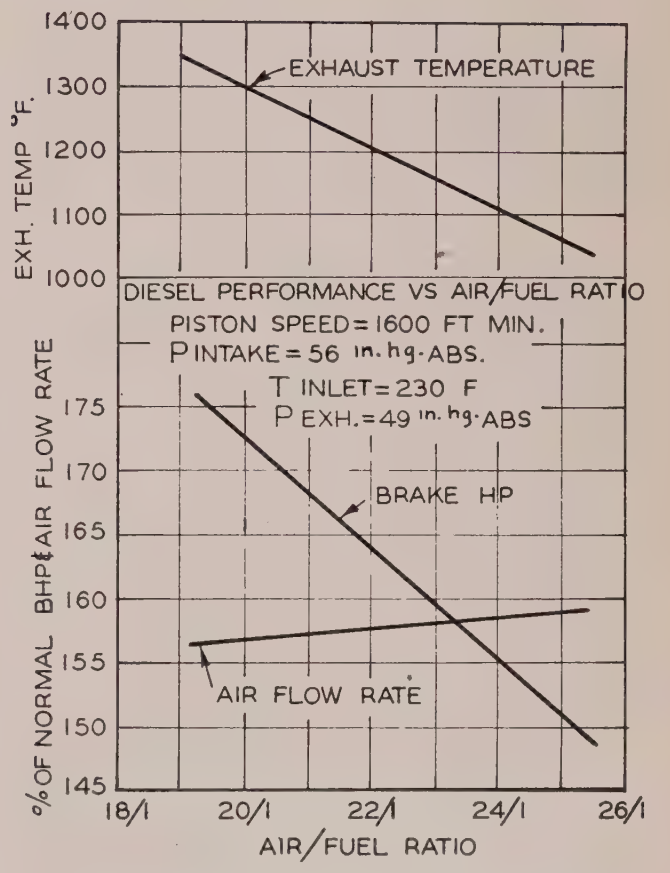


Fig. 19 – Effect of reduced fuel rate at 1600 fpm

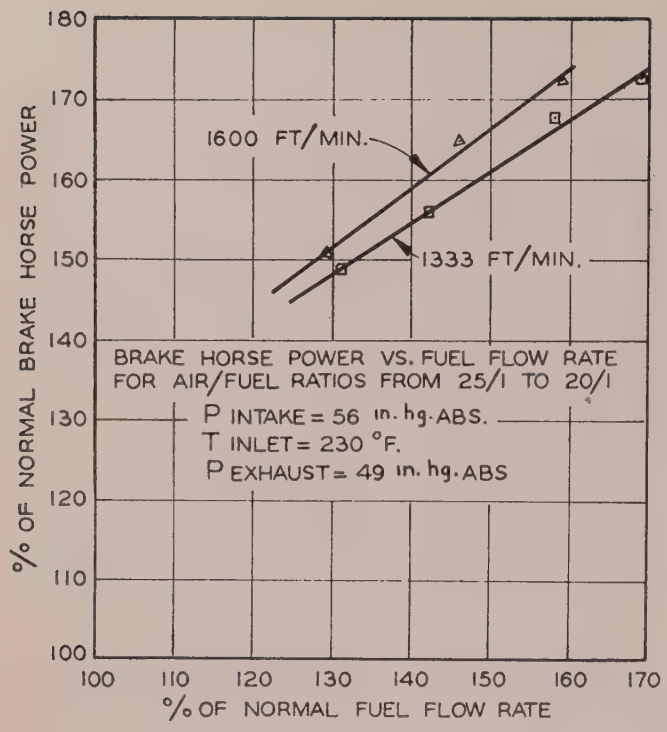


Fig. 20 – Variation in bhp as function of fuel flow

mine the response to variation in intake and exhaust conditions will facilitate evaluation of the possibilities of new engine sizes – also the potentialities of current engines with any of the several systems for supercharging.

APPENDIX

w = Airflow rate, lb per min

w_a = Airflow rate at designated engine speed and air/fuel ratio, naturally aspirated at 29.7 in. Hg absolute barometric pressure and 90 F inlet air temperature, lb per min

bhp_a = Brake horsepower at same conditions as for w_a

t_e = Exhaust temperature, F

p_i = Intake manifold total pressure, in. Hg absolute

t_i = Inlet air temperature measured in manifold, F

p_e = Exhaust manifold total pressure, in. Hg absolute

A/f = Air/fuel ratio, lb air per lb fuel

a_1, b_1, c_1, d_1, and e_1, linear coefficients of p_i, t_i, p_e, and A/f, respectively, defining the effect of each independent variable on airflow, and a constant

a_2, b_2, c_2, d_2, and e_2, linear coefficients, defining the effect of each independent variable on bhp, and a constant

a_3, b_3, c_3, d_3, and e_3, linear coefficients, defining the effect of each independent variable on exhaust temperature, and a constant

DISCUSSION

Describes Parallel Work In Engine Development

– Martin Berlyn
American Locomotive Co.

I HAVE been associated with certain engine developments which closely parallel the work described by Mr. Johnson. Because our work dealt with a direct-injection open combustion-chamber design, we never achieved the high percentage of air utilization mentioned by Mr. Johnson but, like Mr. Johnson, we did employ the sort of valve timing usually associated with naturally aspirated engines.

Our total valve overlap from air-opening to exhaust-closing was 44 deg, but since this was largely made up of cam ramp, the time-opening integral was of such small area as to be negligible from the viewpoint of air-bypassing during the overlap period.

We employed positive-displacement blowers for our tests and provided ourselves with independent control of inlet air pressure, inlet air temperature, and exhaust back pressure; we measured the total air going into the engine and assumed that this was the actual air retained in the cylinder for combustion.

Our air/fuel ratio was not the remarkable 18.2 to 18.5 recorded by Mr. Johnson but was between 22 and 23; our piston speed was 1750 fpm.

Our inlet pressure was around 62 in. Hg absolute and we worked with exhaust pressure from atmospheric up to values that we might expect from a turbosupercharger, and then went on up to back pressures giving zero, and even negative, pressure differentials.

We found that the measurement of exhaust temperature was not clear cut. We measured the temperatures in the exhaust port and at a point where the exhaust pipe entered a surge-tank; we considered this latter point as the "turbine nozzle."

Now here is something which does not show in Mr. Johnson's calculations; we found that, when we had atmospheric pressure in the exhaust system, the temperature measured at the turbine nozzle might be 200 to 250 F, higher than that measured at the exhaust port. As the exhaust back pressure was increased, both temperature readings also increased, but the reading at the exhaust port rose more rapidly (in relation to the increase of back pressure) than the reading at the turbine nozzle. When the differential pressure (inlet air and exhaust) reached about 12 in. Hg the two temperature readings were the same. Further increase of exhaust back pressure caused the port temperature reading to rise above the reading at the turbine nozzle.

The temperature rise along the exhaust tract, with zero or low exhaust back pressure, suggests burning of fuel in the exhaust system, but we do not believe that this was actually taking place.

Mr. Johnson has not stated at what point or points he measured the exhaust temperatures recorded in his plots; I should be interested to know just what he did.

We are not satisfied with the accuracy of our exhaust temperature indications, and I would be interested to hear Mr. Johnson's views on this point.

Similar Tests Offer Comparative Data

– C. B. Dicksee
A. C. E. Ltd.

I HAVE found myself in fairly close agreement with the author in most respects.

In our work we normally speak in terms of air used, rather than air/fuel ratio or excess air. Air used has the great advantage that many interesting relationships appear very nearly as straight lines instead of as hyperbolic curves. Another useful parameter is energy supplied per unit of cylinder volume as this enables one to compare engines regardless of size – it is, of course, another version of air used, but does not necessitate knowing the airflow.

I am a little surprised that the bhp lines in Figs. 1 to 4 of Lloyd Johnson's paper are such definite straight lines, because the temperatures all around the cycle will have been increased by the increase in intake temperature, and this will affect the efficiency. The fact that the air/fuel ratio has been maintained constant has reduced the effect of an increase in air temperature, but I should have expected a noticeable deviation from the straight line especially as the friction will be constant and represent a differing proportion of the ihp.

Again, in Fig. 5, there is practically no deviation from a straight line relationship between airflow and bhp despite an increase in piston speed of 2 to 1. Surely the mechanical efficiency should have altered sufficiently with double the speed to cause a marked departure, or are the higher piston speed figures obtained from different engines? The ihp will increase directly with the airflow as the engine speed increases, but the higher friction loss should cause some deviation in the case of the bhp. The paper suggests that some change in combustion efficiency takes place at higher speeds with a change in air temperature, but if this is the case the bhp should deviate even more from the direct relationship.

I was interested in the effect of increased back pressure on friction mean pressure, Fig. 17. In some tests we have made we found that 1 lb sq in. back pressure gave 1 lb sq in. loss in bmep, but your figure is more favorable than this, although with the engine firing the loss may be a bit higher than when motoring, as the increased back pressure will result in some increase in the volume of residual gases retained in the engine.

Questions Suggested by Test Equipment Used

– R. T. Sawyer
American Locomotive Co.

AS I studied Mr. Johnson's paper a number of questions arose about the test equipment used. He did not state the type of blower used or the nature of air connection to the engine. Was an air chamber used to equalize pressure to all cylinders, or a cooler of appreciable volume employed?

While the reader might assume that the engine tested was a precombustion chamber-type diesel, Mr. Johnson did not so state. I would be interested to know whether the engine tested was an in-line or V arrangement, how many cylinders, the arrangement of intake and exhaust manifolds, and the valve event timing.

Since the blower was driven by another engine, were the bhp figures of the engine under test corrected for the power needed to drive the blower in arriving at the power outputs shown?

Mr. Johnson touched on the subject of large inlet and exhaust overlap, inferring that the engine characteristics shown would not apply to large overlap. I feel this quite unfortunate as the experience of many builders of supercharged engines has shown improved scavaging and cooler, heavier air charge, as well as improved turbosupercharger performance resulting from increased overlap.

The last question I would like to ask is whether Mr. Johnson finds he can really use data from such tests to determine the type of supercharger he wants on a variable speed, variable load engine, as in a tractor?

Author's Closure To Discussion

MR. BERLYN'S comments are gratifying inasmuch as the experience he mentions so closely parallels ours. We haven't until recently made a practice of measuring exhaust temperatures in the ports from individual cylinders – a quite usual practice with large engines. On occasion we have had difficulty in reconciling exhaust temperature measurements of one test or engine model to those of another. In general we are reluctant to hang much faith on the accuracy of exhaust temperature measurements because of the pulsating nature of the hot gas flow and the presence of cooler walls of considerable mass.

Usually we can only make certain that what we read is the temperature of the thermocouple bead – not that of the gas. However, by consistently using a reliable type of thermocouple installed in the same place at the junction of the exhaust manifold and stack, we do learn by experience how hot a reading is still safe for highly heat-loaded engine parts. This temperature also is useful in predicting the operating temperature of nozzle and wheel blading of a turbosupercharger.

In answer to Mr. Sawyer's questions, the engine used was a 6-cyl, in-line, precombustion chamber diesel of 5¾ bore and 8 stroke. We have found that the results are applicable to our line of V-8 and V-12 engines which employ the same pistons, valves, cam timing, and injection equipment. The inlet manifold was of the trunk type with a heat exchanger of small volume between the blower and the manifold.

As Mr. Berlyn mentioned, the opening and closing portions of valve motion are at low velocity on a ramp which may be long or short, depending on valve train design and service requirements. We are very reluctant to attach appreciable significance to the actual opening and closing points specified for the convenience of service personnel. The overlap used in the engine under test was optimum for unsupercharged performance. Larger overlap, while increasing airflow appreciably, only slightly increases charge weight available for combustion. We have not found any appreciable piston-cooling effect from increased overlap. The bhp figures were not corrected for the power needed to drive the blower, as must be done for gear-driven superchargers. The extra airflow with large valve overlap increases gear-driven blower power requirements excessively.

One of our purposes in running such engine tests was to find out how far actual performance deviated from that we would expect from computations based on air density, known engine friction, and so forth. After we have the data from an actual engine test, we often find we can logically attribute the deviations to factors we would have neglected, in predicting performance on the basis of computations. Fig. 6 shows the expected straight-line relationship between ihp and airflow as the line has a slope of unity. The light broken line of Fig. 5 does not have a slope of unity. That such a line can be drawn is partially due to coincidence arising from the choice of boost and back pressure for each piston speed, such that bmep increased somewhat proportionally to fmep. That such a relationship exists under conditions typical of turbocharged engine operation is of interest but not conclusive.

Fig. 17 does not show the loss in net power output resulting from reduced air charge in the cylinder with increased back pressure. We check Mr. Dicksee's figure of a loss of one bmep for each psi increase in back pressure at the higher piston speeds. At lower piston speed the loss becomes as high as 1.5 bmep per psi increase in back pressure.

CONSIDERATIONS

The Authors

J. A. NEWTON (M '41) is chief engineer of the Valve Division, Thompson Products, Inc. He obtained his bachelor's degree in mechanical engineering from the University of Illinois and his master's in metallurgy from Ohio State University. After two years as a student engineer with Chrysler, Mr. Newton joined Thompson Products in 1941 as a service engineer.

N. HOERTZ (M '38) is chief engineer of the Service Division, Thompson Products, Inc. He received his degree in mechanical engineering from Case Institute of Technology, and has been with Thompson Products since 1933. Mr. Hoertz is a past-chairman of the SAE Cleveland Section.

THE purpose of this paper, as the subject indicates, is to cover valve problems on behalf of the vehicle operator and maintenance man. The subjects to be discussed are:

1. Wear points in the valve train. How they should be checked, and the wear limits that can be tolerated.
2. Suggestions on the types of valves and valve seat inserts to be chosen for the various applications.
3. Valve rotator mechanisms, their service requirements and possible benefits from their use.
4. Valve lash adjustment and its effects on expected valve life.

Engine Disassembly and Inspection

The first operations necessary in any engine rebuild are disassembly, cleaning, and inspection of parts. At this point, we are mainly interested in the inspection of the units in the valve train. Two common wear points are the rocker arms and tappets (Fig. 1). A depression in the tappet screw or in the rocker arm face, if not corrected, will give erroneous tappet clearance readings. The actual clearance will be on the plus side and can contribute to valve breakage. The first signs of tappet-screw or rocker-arm wear can be noticed on the valve stem tip. The edges of the chamfer, which are normally sharp, will be round. The corrective measure to be taken is to regrind the worn surface to make it conform to its original contour,

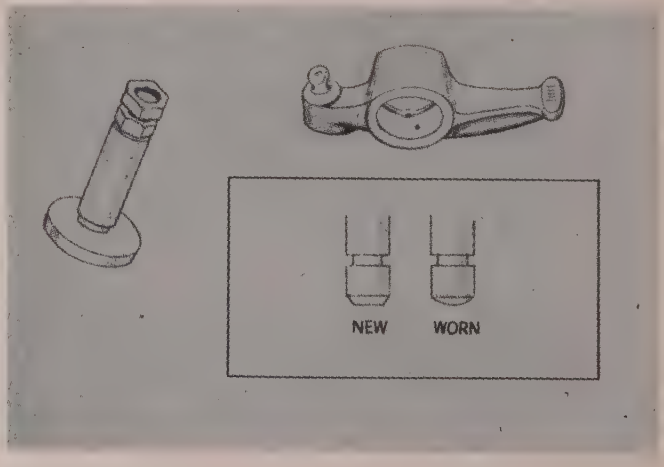


Fig. 1 – Tappets and rocker arms are two common wear points

FOR COMMERCIAL VALVE MAINTENANCE

J. A. Newton and N. Hoertz, Thompson Products, Inc.

This paper was presented at the SAE Annual Meeting, Detroit, Jan. 14, 1952.

thus providing for the proper feeler-gage reading. In cases of extreme wear, replacement of the tappet screw or rocker arm may be necessary. While observing the wear on the tappet adjusting screw, it is also good practice to note the conditions of rocker arm bushing and the tappet face. Extremely worn bushings or scuffed tappets should be replaced.

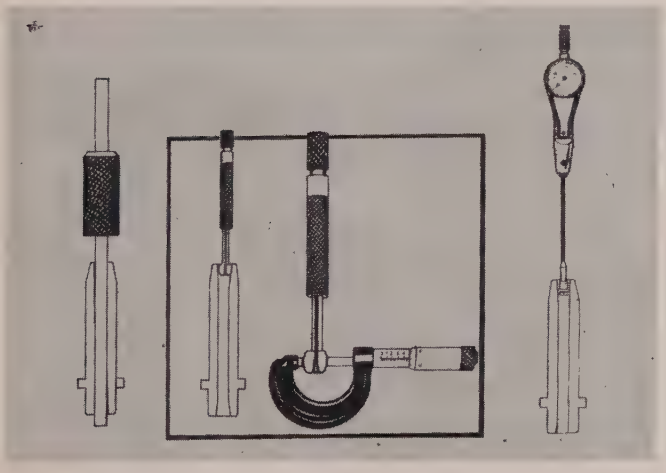


Fig. 2 – Valve guide wear can be accurately measured with gages

The next item to be checked is the valve guide. (See Fig. 2.) Since guide wear is usually in the form of bellmouth on the ends or directional wear, which makes an oval out of the inside diameter, it cannot be accurately measured with a plug gage. A more accurate device is the small hole gage. Both the tapers and the amount of oval can be easily taken from the guide and transferred to a standard micrometer. Some gage manufacturers also have available a direct-indicating dial gage. These, of course, are more expensive, but do offer a convenience.

Guide Wear May Be Serious

Guide wear should be taken seriously because it affects the overall valve temperature. (See Fig. 3.) As the guide clearance is increased, there is less contact between stem and guide, and likewise less heat conductivity, giving a temperature rise in both the valve stem and head. The higher clearance also allows movement of the valve on the valve seat insert, causing wear and gas leakage,

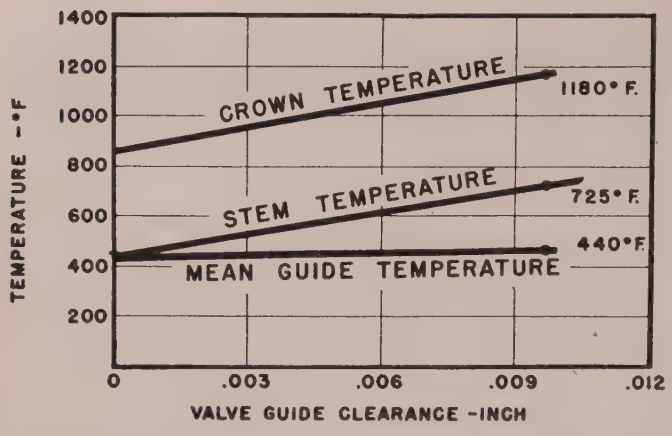


Fig. 3 – Effect of valve guide clearance on valve temperature

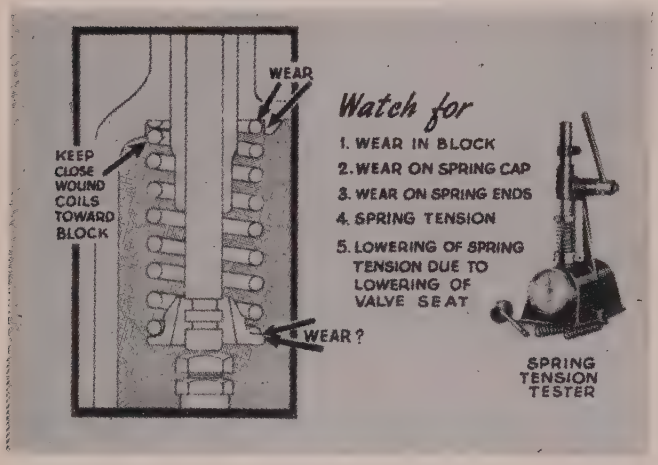


Fig. 4 – Valve springs should be checked for five wear points

factors that contribute to valve burning, loss of compression, and power.

After guides, the next consideration is the valve spring. (See Fig. 4.) There are actually 5 points to be observed:

1. Wear on the block usually comes from high-speed operations, where the springs rotate and produce grooves.

2. The same wear may or may not appear on the spring caps. Occasionally, spring caps have been found worn almost to a knife edge.

3. The same condition causing (1) and (2) may wear the spring ends; either one or both ends may have these wear indications.

4. The spring tension itself should be checked with an accurate scale. The high operating-stress level of valve springs, coupled with their high temperature of operation in engines, accounts for pressure loss over long periods of time. Tension loss exceeding 10% of the specified required load should indicate replacement.

5. Another factor which should be noted is the pressure loss, due not to the spring itself, but to the lowering of the valve on the seat caused by grind-in. (See Fig. 5.) As the valve seat is ground deeper, as shown in B, C, and D, the spring operating height is progressively longer, and the spring operating loads are lower. When the seat is lowered as much as 1/16 in. or more, it is desirable to re-establish the correct spring operating height by the installation of a spacer washer. Of course, a new valve seat insert should be considered under conditions of this kind. The seriousness of this can be more clearly portrayed by showing the exact loss in pressure caused by each increment of spring length increase. (See Fig. 6.) Each 1/32-in. of grind-in accounts for a 5% loss in valve open spring load with new springs, and much more if

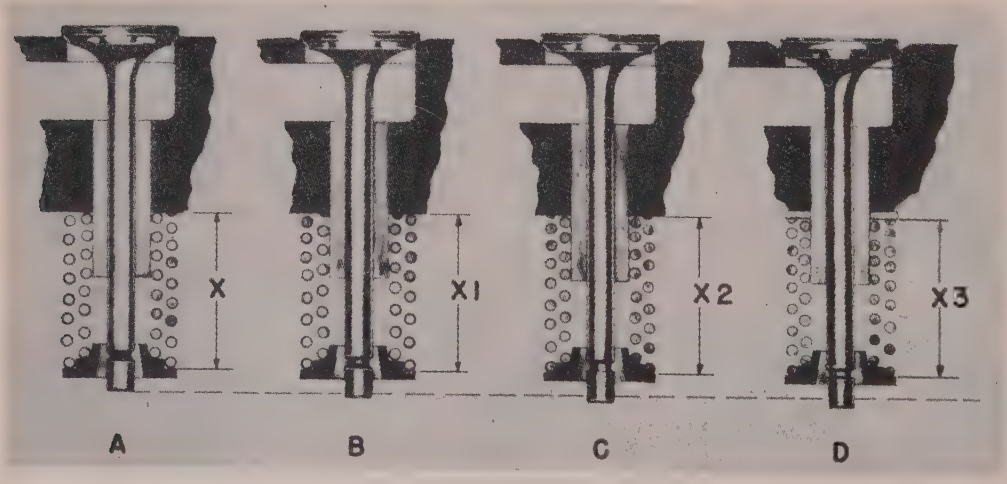


Fig. 5 – As valve seat is ground deeper (shown in B, C, and D) spring operating height becomes longer, and loads become lower

THE first step toward longer valve life as described in this paper, is a careful disassembly and inspection of the engine. Good judgment and care should then be used in correcting or replacing those parts that are scaled or damaged through previous usage.

When the engine is disassembled and inspected after a run, it may be found that some of the worn parts are still within wear limits that can be tolerated. The valve stem tip will show the first sign of tappet-screw or rocker-arm wear, which, if serious, will lead to erroneous tappet clearance readings. Regrinding may accomplish the correction. Extremely worn bushings or scuffed tappets should be replaced.

Valve guide wear should be carefully checked because it affects the overall valve temperature, the increase in clearance leading to less heat conductivity. The next consideration is the valve spring, for which the authors suggest the checking of five wear points.

It is pointed out that extreme care in the reconditioning of valve gear can be rendered useless by careless lash setting. Insufficient lash drastically reduces valve life, whereas excessive valve lash generally leads to broken valves.

The authors describe an effective means of rotation now available which does not require continuous maintenance to ensure its proper functioning.

the old springs, which have taken a set, are reused. The loss of tension greatly affects the valve motion. At high speeds, false valve motion results in high stresses which are a contributing factor in breakage.

Valve Seat Insert Choice

If a new insert is required in an engine, the following suggestions should be noted. (See Fig. 7.) If there is no choice except an exact duplicate of the original with the outside diameter increased, it, of course, should be used. In the event there is a choice between a deep and shallow seat, the one with the greatest depth should always be chosen. The deeper seat has been proved to stay tighter and perform longer. In the event there is a choice between a hard iron seat and a hard-faced seat, such as stellite, the following fact should be considered. (See Fig. 8.) Under test, the hard-faced seat has an endurance life of 4 to 5 times that of hard iron. There is also a difference in cost to be considered.

These facts, with the previous history on the operations of valve seat life, should give the information needed to make the choice. After the choice of insert is made, the next important item is the proper amount of interference fit. (See Fig. 9.) It should be pointed out that the cavity in the cast iron block should be accurately machined for size, by all means very smooth, with no tool marks. Tool marks will rapidly shear off, causing a reduction in interference fit. The bottom of the cavity should also be smooth and square to provide the largest possible contact with the seat itself. The greater the contact area, the greater the heat conductivity, and a consequent lower seat-operating temperature will result.

Even with extreme care and accuracy, there are some disturbing factors affecting the valve seat insert (See Fig. 10.) The unevenness of the water jacket area, scale formations in the water jacket, and the positions of the exhaust port in relation to the insert, as well as installation technique, all have the effect of shifting the seating surface under operating conditions. With an indeterminate amount of shift inherent in the engine it is hardly necessary to re-emphasize the necessity of accurate grinding of the seat for concentricity to the guide inside diameter.

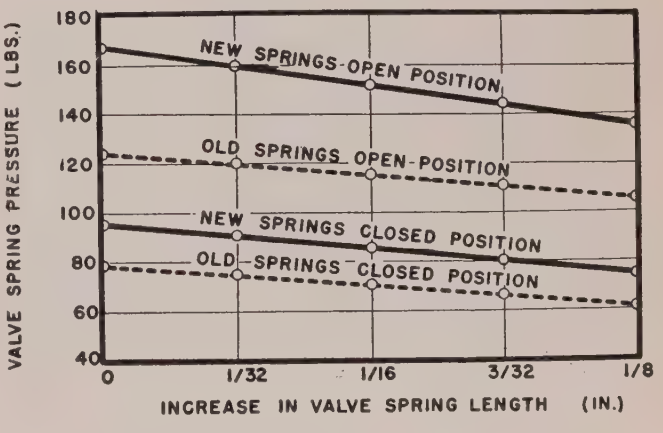


Fig. 6 – Each increment of spring length increase leads to loss of pressure with both new and old springs

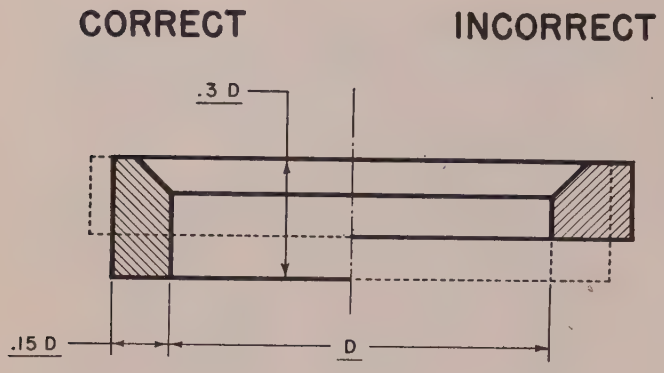


Fig. 7 – If possible, a deep, hard-faced seat should be selected

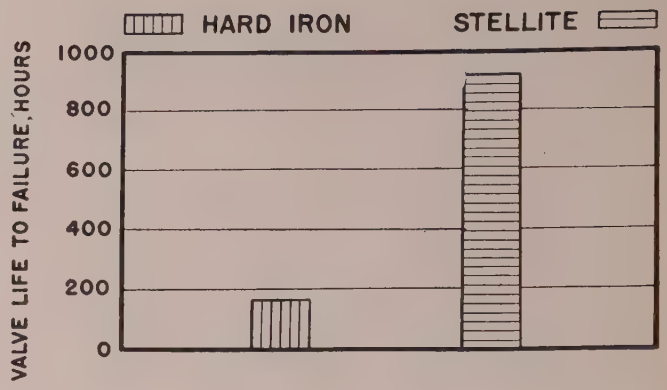


Fig. 8 – Hard-faced material has considerably more endurance than hard iron

OUTSIDE DIAMETER	INSERT DEPTH	INTERFERENCE FIT
0 - 1	0 - 1/4	.001 - .003
1 - 2	1/4 - 3/8	.002 - .004
2 - 3	3/8 - 9/16	.003 - .005
3 - 4	9/16 - 11/16	.004 - .006

Fig. 9 – Recommended interference fits for hard cast or wrought inserts, with cast iron ports

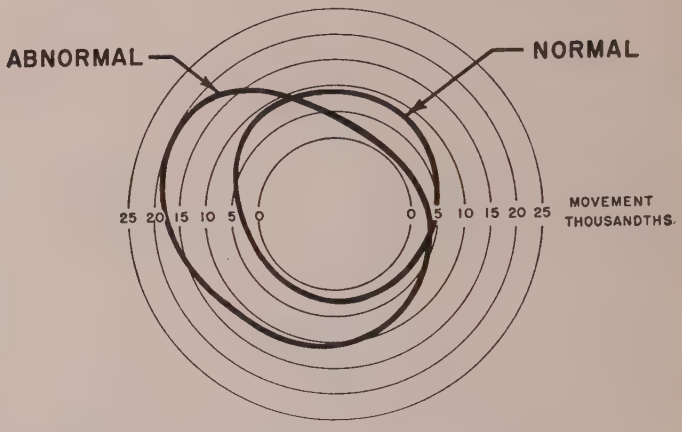


Fig. 10 – Insert seating surface shift necessitates accurate grinding of seat

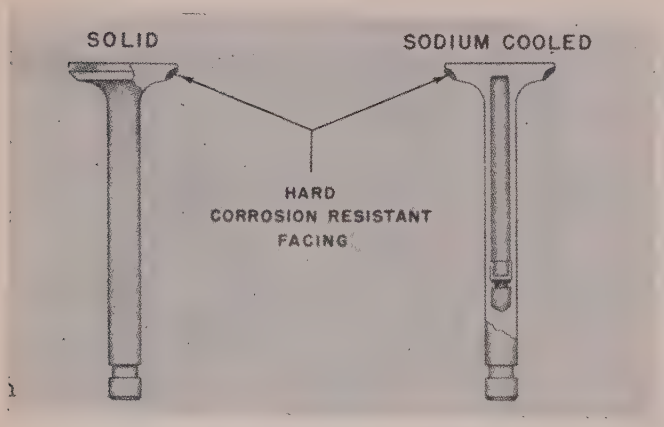


Fig. 11 – Two heavy-duty valves in common usage

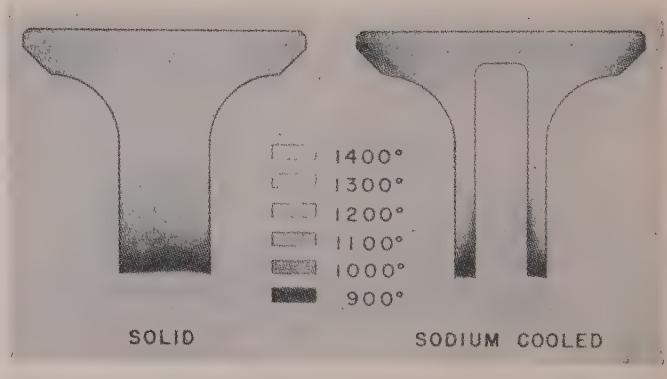


Fig. 12 – Valve temperature comparisons showing normal reduction through sodium cooling

Valve Choice

The next important consideration is the selection of the proper valve. Heavy-duty valves in common usage are of two types: (Fig. 11) (a) Solid with hard facing. (b) Sodium-cooled with hard facing. Other types having the dome either partially or totally coated with a corrosion resistant material (nichrome) have been specified by original-equipment users, but experience has indicated that for the same amount of expense sodium cooling provides the greater benefits. (See Fig. 12.) Sodium cooling normally reduces the valve operating temperature about 200 F. This valve temperature reduction usually provides conditions for increased valve life. Whenever sodium-cooled valves are available as a service item, the extended valve life gained through their use usually justifies their increased cost.

Valve Rotation

Valve rotation (Fig. 13) is being accepted generally by original-equipment manufacturers and gaining wide favor in the replacement market. The premium paid for equipping an engine with rotation is justified by the increase in trouble-free valve operation. While rotation generally extends valve life 2 to 5 times, frequently valve life extension of 10 times and more is obtained. Valve rotation extends valve life for the following reasons:

a. Valve face and seat are constantly wiped clean – no buildup of deposits to flake off and start valve guttering.

b. Stem deposits, which cause valves to stick in their guides and to be held open, are prevented.

c. Effects of distortion due to thermal and/or mechanical factors are minimized.

d. Local hot spots that cause burning are prevented.

e. The valve tip is prevented from grooving due to rocker arm scrubbing.

f. More uniform stem lubrication is provided, and acts to prevent scuffing.

Basic Types of Rotators

There are three basic types of rotators available today. All three are made available as original-equipment items and two are made available through service outlets for additional applications. (See Fig. 14.) The positive-type rotator, trade name Rotocap (Fig. 15), is the most widely distributed system of valve rotation. It imparts positive rotation to the valve during each valve cycle. The Rotocaps in use may fit either the tip-end or the guide-end of the valve spring. In making the original design, that location is selected which is the most satisfactory from a valve-gear design standpoint. There are no maintenance procedures required for the Rotocap during its operation; it

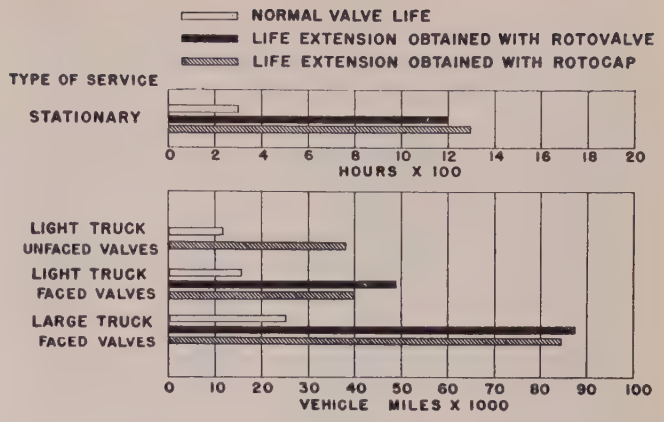


Fig. 13 – Valve life improvement obtained by rotation

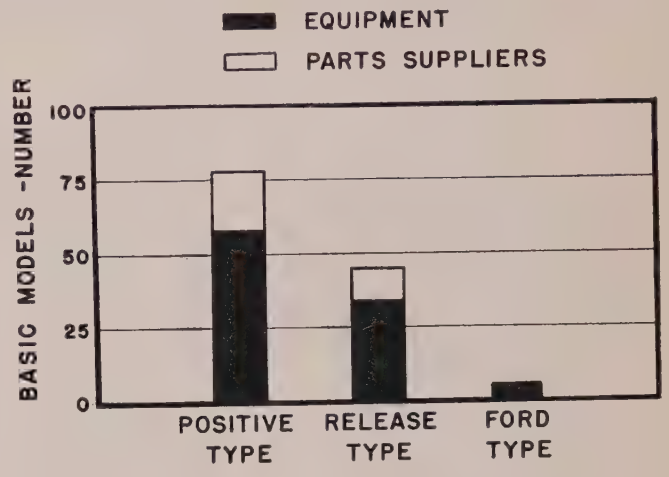


Fig. 14 – Availability of three basic types of rotators

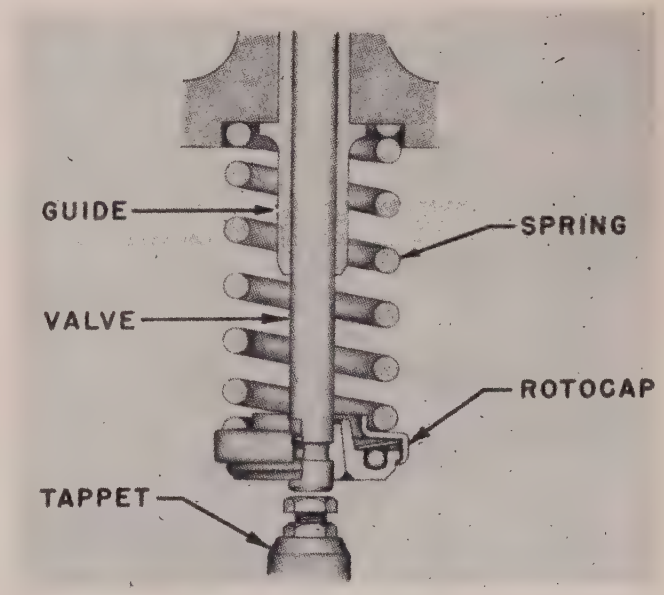


Fig. 15 – Positive rotator imparts positive rotation to valve during each valve cycle

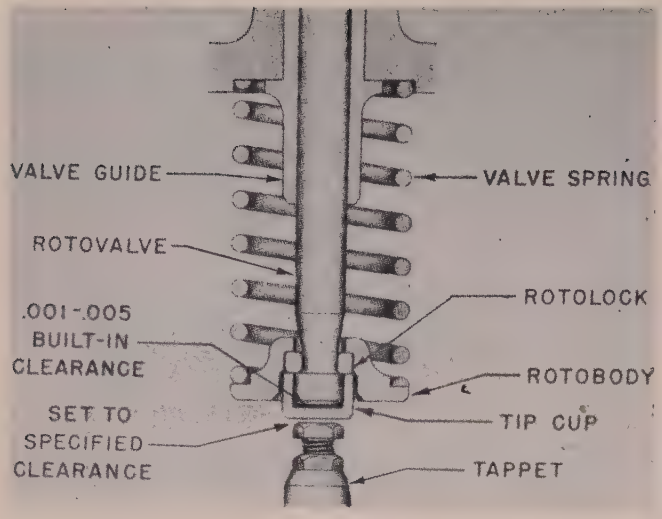


Fig. 16 – Release rotator depends upon engine vibration to impart nonpositive rotary movement to valve

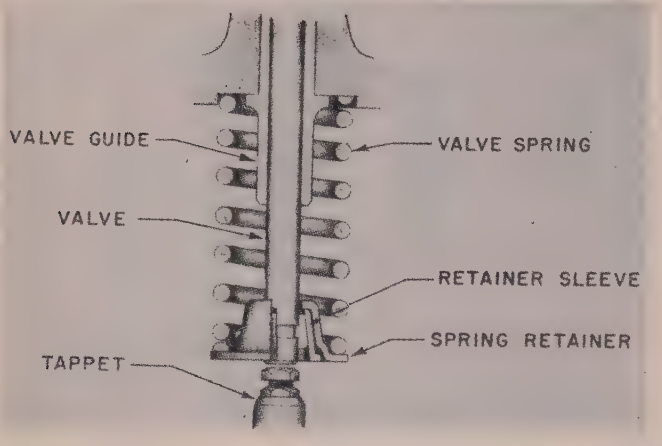


Fig. 17 – New Ford rotator depends upon a combination which allows valve to creep around

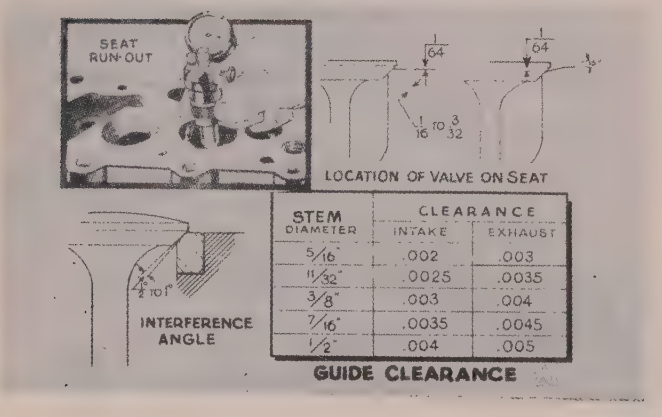


Fig. 18 – After worn parts are replaced the valve is fitted to the engine

will continue to function satisfactorily under all normal conditions for 100,000 miles or more. However, when valves are replaced or reground at high mileages, Rotocaps should be replaced, since there is no visual inspection that can be made to determine their functioning characteristics – rotation of the unit can be checked *only* in the engine. Whether or not the Rotocap rotates when held in the hand is no indication of its functioning characteristics in the engine.

The release-type rotator, trade names Rotovalve, Slo-roto, and Free Valve (Fig. 16), is the other type of rotator commonly used. It depends principally on engine vibrations to impart a nonpositive rotary movement to the valve. There are certain maintenance and rework procedures necessary to the release-type rotator to insure its continued operation.

A rotator was recently introduced by the Ford Motor Co. (See Fig. 17.) Its operation is dependent on a low-friction bearing area in conjunction with a low-frequency, flexible-valve-spring design. This combination allows the valve to creep around. The service record on this type of rotator is still somewhat limited.

Reducing Deposit Problem

After the engine has been carefully checked, and worn parts culled out and replaced by new parts, the engine is ready for assembly (See Fig. 18.) The valve guide, after installation, is reamed to give nominal clearance. The valve seating surface is ground to a 0.002-in. maximum runout limit and is cut to the proper width of 1/16 to 3/32 of an in., allowing 1/64-in. overhang of the valve outside diameter. An interference angle is accepted practice. With rotation, the interference angle is omitted and the seating surface should be 0.090 to 0.100 of an in. wide wherever possible.

Rotation has practically eliminated the need for the time-proved fixes to minimize sticking. (See Fig. 19.) Yet, in some installations the guide counterbore and the valve-stem carbon relief are used separately and in conjunction with rotation to cut the stem deposit problem to a minimum.

Valve Lash

Premium valves are available for replacement purposes. These, in many cases, are made from different material than the original valve. (See Fig. 20.) Differences of as much as 50% in the coefficient of expansion may occur in the different materials, and the change in length occurring during the valve heating under load must be accounted for by different lash settings. Engine manufacturers in their instruction manuals generally indicate a proper hot or cold valve setting. Unless otherwise instructed by the manufacturer, lash adjustment should be made on a thoroughly

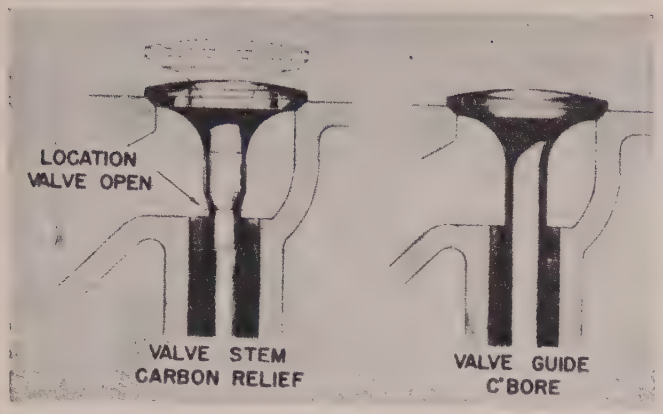


Fig. 19 – Two methods used in conjunction with rotation to prevent sticking

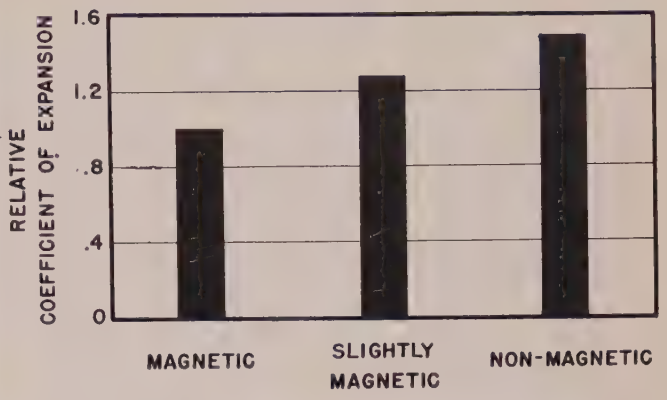


Fig. 20 – Expansion rates of valve steels depend upon magnetic properties

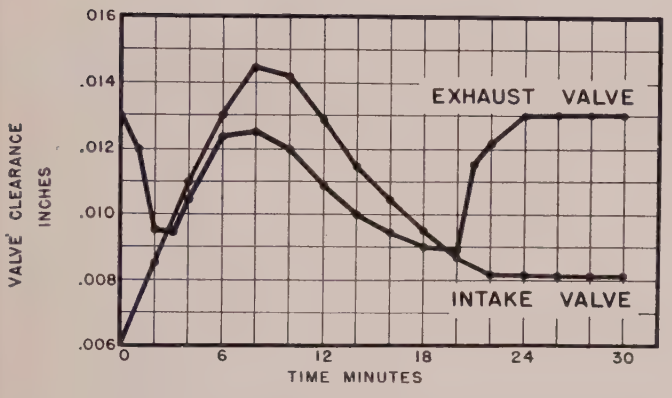


Fig. 21 – Effect of warm-up on lash setting with a 6-cyl, overhead-valve engine

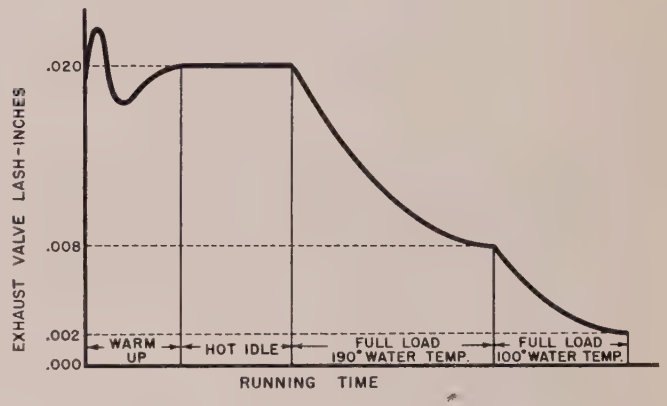


Fig. 22 – Effect of operating conditions on exhaust valve lash

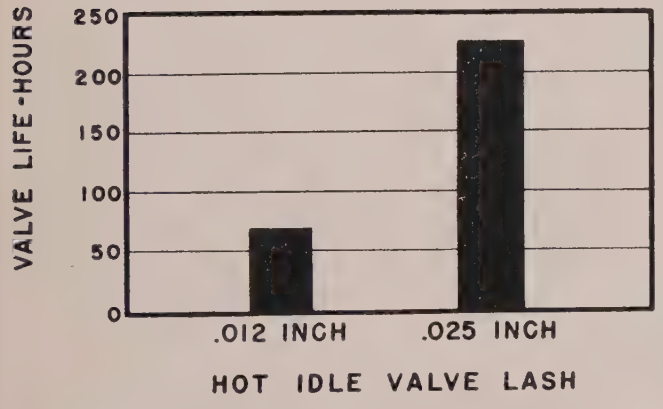


Fig. 23 – Effect of insufficent valve lash on valve life

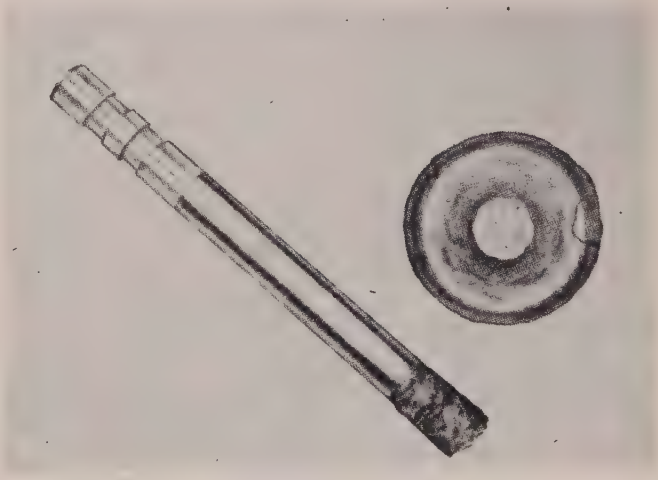


Fig. 24 – Broken exhaust valve due to excessive operating clearance

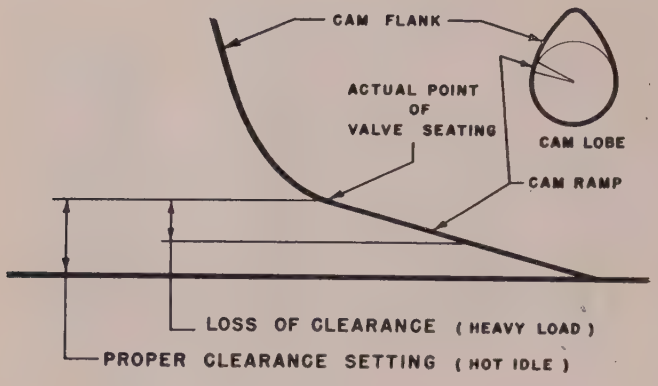


Fig. 25 – Valve seating with proper tappet clearance

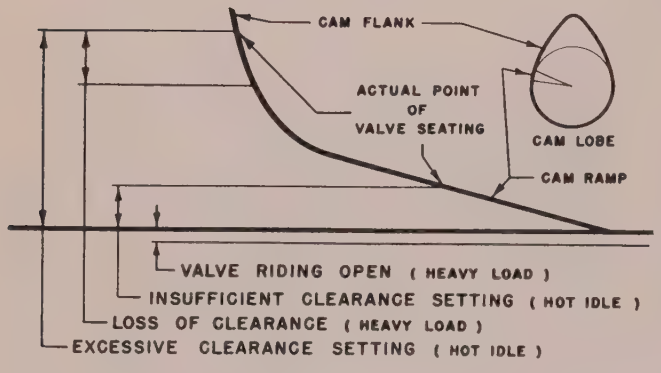


Fig. 26 – Effects of tappet clearance on valve seating

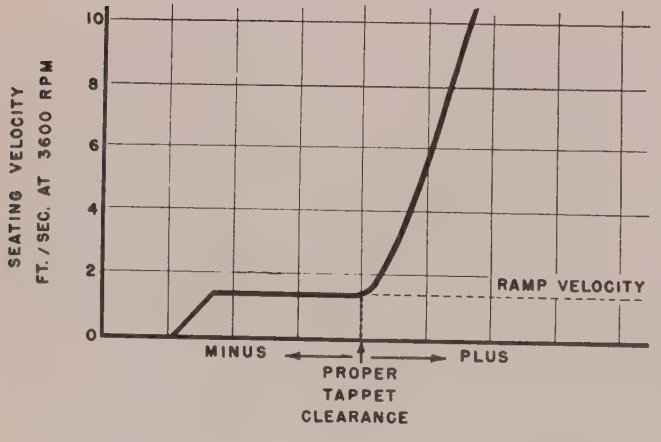


Fig. 27 – Valve seating velocity remains constant with properly established lash setting

warmed engine (See Fig. 21.) There is a possible wide fluctuation in the valve lash readings between a cold and a thoroughly warmed engine. The conditions affecting these lash readings include temperature of the water, oil, block, head, and valve gear. In the example shown, the agreement of lash setting of the exhaust valve, hot and cold, was purely coincidental and is not indicative of a general condition.

Other engines may show a change in valve lash of 0.005 in. or more between a cold and thoroughly warmed engine. Setting the valve lash correctly is extremely important. Operating lash will change with load from its hot idle condition. (See Fig. 22.) Manufacturers specify lash based on studies made of operating clearances while various load conditions are applied to the engine. This work is all done on the engine in its early stages of development. Operation without thermostats can, in cold weather, be responsible for decreased operating clearances under full load.

Results of Excessive Clearance

Insufficient operating clearances, whether due to improper lash or lax engine maintenance condition, reduce valve life to as little as 25% of normal. (See Fig. 23.) Operators realize fully the result of not enough lash, but few realize the result of excessive clearance, which is broken exhaust valves. (See Fig. 24.)

To understand valve breakage caused by excessive lash, the actual seating of the valve must be studied. (See Fig. 25.) A valve having the correct lash setting at hot idle will seat as the tappet enters the ramp portion of the cam lobe. The ramp of the cam is that part of the lobe which allows for quiet idle operation, and easy seating of the valve with a means for accommodating the fluctuating valve lengths as they occur during operation. With clearances set less than the prescribed minimum, the valve is riding open during full-load operation, allowing the blowby of hot exhaust gases to precipitate premature burning. On the other hand, clearance beyond the prescribed maximum allows the valve to seat abruptly and noisily while the tappet is on the cam flank. (See Fig. 26.) The magnitude of the impact taking place while the valve seats is expressed in terms of seating velocity. (See Fig. 27.) The valve seating velocity remains constant with the properly established lash setting. Excessive lash settings rapidly increase the impact due to seating. The stress developed in the valve is in direct proportion to the seating velocity. (See Fig. 28.)

Burned and Broken Valves

Excessive clearances commonly found in the field often result in stresses reaching a magnitude of 3 or more times the normal. This (Fig. 29) predicts

failure. As designed, the valve stress due to seating is sufficiently low with ample safety factor to assure freedom from breakage.

To summarize, the most important factor in good valve maintenance is correct lash. Proper settings may not be responsible for desired valve life, but improper lash can and is the cause of several undesirable conditions, the most important of which are burned and broken valves. In other words, tappet noise, rough engine idle, burned and broken valves, poor performance, and poor economy, all result from improper valve lash settings.

Valve Rotator Maintenance

With the engine in service, the only valve gear item left to consider, except for the routine lash settings, is rotator maintenance. Neither the Rotocap nor the Ford rotator require any maintenance. The release-type rotator requires periodic checking of the clearance built into the valve tip cup. To make this check in an overhead valve engine, Automotive Service Tools, Inc. have produced a gage. (See Fig. 30.) The built-in clearance is to be maintained at 0.001 to 0.005 in. Wear is uniform up to 0.006 in., after which time the rate of wear rapidly increases and is liable to cause valve breakage, in the same way as excessive valve lash. This gage is used as follows:

1. Remove tip cup from the exhaust valve end.
2. With the end of the valve stem and the plunger pin clean, the gage is placed in the end of the exhaust valve stem with the gage set at the O mark. The plunger pin is held firmly against the valve tip and the clamp screw tightened.
3. The valve tip cup is placed against the plunger pin, and the clearance or lack of clearance is transferred to the spindle barrel.
4. Readings to the right of O indicate positive clearance, and readings to the left of O indicate negative clearance.
5. To check the tip cup clearance with the valve removed from the engine, the same relative position of the parts is maintained and the readings made as outlined above.

Two Gages Suggested

If the valve mechanism is removed from the engine, either the gage made by Automotive Service Tools or the one produced by Kent-Moore can be used. (See Fig. 31.) This gage can be used as follows:

1. Mount the tool on a bench vise and set the micrometer head at O.
2. With all the slack in the retainer and locks eliminated, the valve with the parts in place is pulled up against the micrometer head by means of the valve spring retainer.
3. Allow the retainer and valve locks to rest on the V-block. This permits the valve tip cup to rest

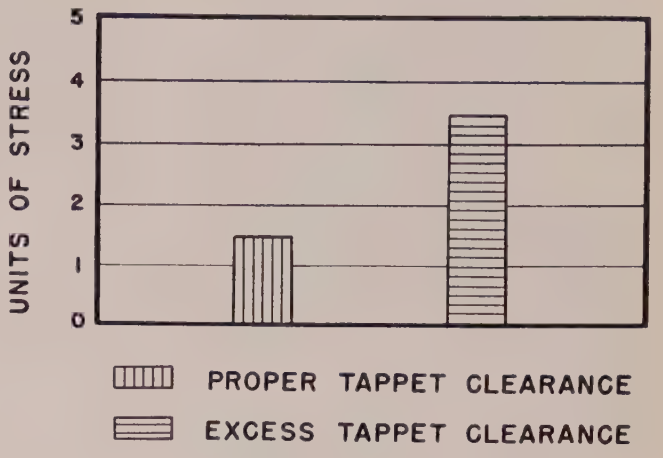


Fig. 28 – Effect of tappet clearance on impact stress

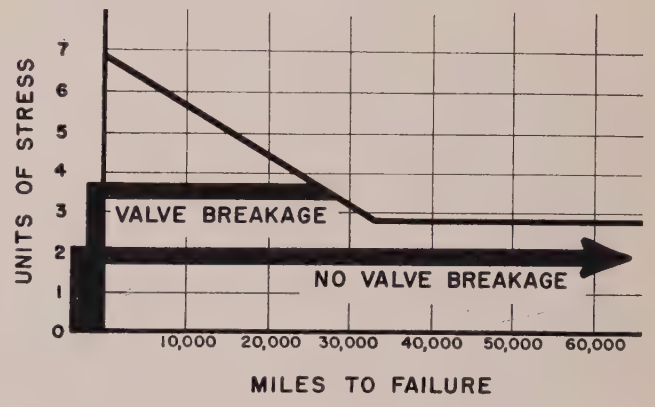


Fig. 29 – Excess clearance cause of valve breakage

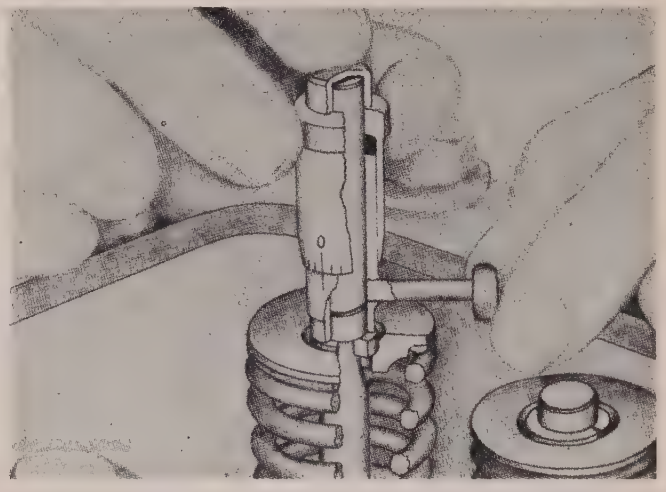
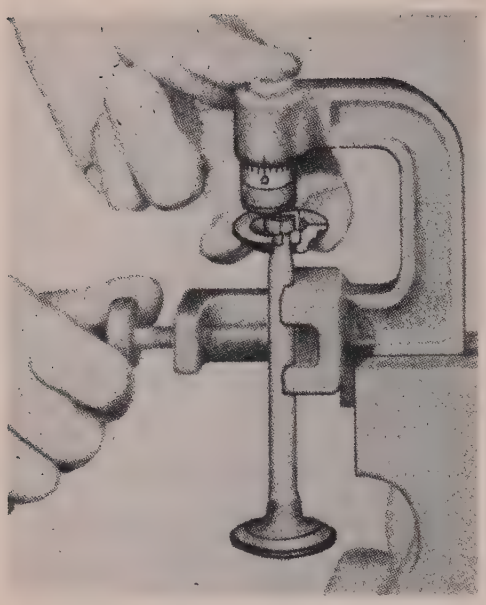


Fig. 30 – Checking clearance of release-type rotator with specially developed gage



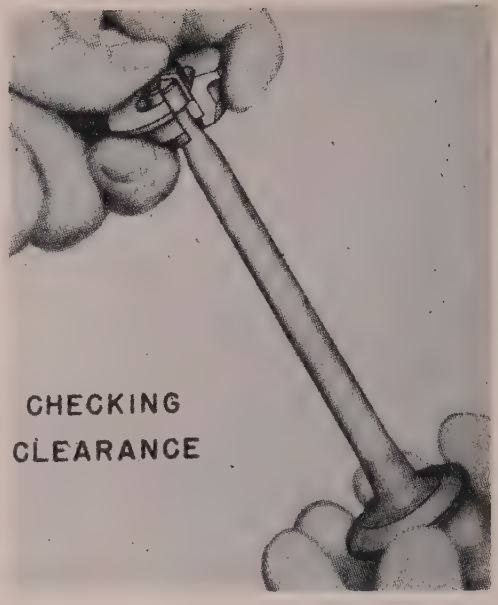


Fig. 31 (left) – Checking clearance of release-type rotator with a second type of gage designed for this purpose

Fig. 32 (right) – Although precision measurements are preferred, clearance of release-type rotators may be checked by shaking to determine looseness of valve

on the valve tip, exposing the clearance between the cup and the micrometer head.

4. Rotate the micrometer head to the left until it contacts the valve tip cup. The clearance is read in thousandths directly from the micrometer scale.

Another means of checking valve-tip clearance is the plastigage method. A plastic strip is inserted between the cup and the valve tip. With the locks held firmly against the shoulder on the valve, the cup is pressed into place. The width of the ribbon is then checked with the gage supplied. Still another simple but not very accurate method is to hold the parts as required and shake for looseness of the valve. (See Fig. 32.) Looseness of the valve indicates valve-tip cup clearance, but does not give the amount. A precision measurement is greatly preferred.

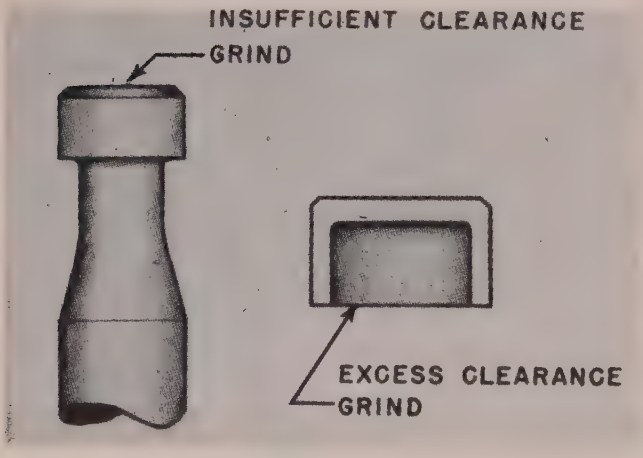


Fig. 33 – Adjustments may be made by proper grinding to increase or reduce clearance

Reducing or Increasing Clearances

In any case, if adjustment is required, grind the open end of the tip cup to *reduce* the clearance. Grind the valve tip to *increase* the clearance. (See Fig. 34.) Flat and square grinding of the valve tip or cup is recommended for proper adjustment and best results. With the release-type rotator, several items are worthy of mention.

1. Valve locks and tip cups, once fitted, should be kept as a group after the engine has been placed in service.

2. The valve locks and shoulder of the valve tip may show wear, which is not harmful as long as the tip cup clearance is maintained. When reinstalling, be sure that locks are set with wear in same position to avoid cocking the spring retainer.

3. Rectangular section locks can be inverted to utilize the unworn side of the locks. If this is done, the valve tip cup clearance must be rechecked.

In summary, if you follow with good judgment the principles advocated herein, the valve gear of the engine you are rebuilding may not produce a million-miler, but the chances are it will give much better valve life than one which has been rebuilt ignoring the fundamentals that have been stressed here.

Acknowledgment

The authors wish to thank E. A. Owsiak for his assistance in the preparation of all of the figures which have been used in this paper.

Investigation of Jet Engine Compressor Stall and Other Transient Characteristics

W. J. Kunz, Jr., Bendix Products Division, Bendix Aviation Corp.

This paper was presented at the SAE National Aeronautic Meeting, New York City, April 23, 1952.

THE inherent design performance of the modern jet engine can only be realized if suitable fuel control mechanisms are available. Thus far, more progress has been made in developing the basic engine structure than in defining the problems associated with controlling the engine power. The characteristics of the early centrifugal compressor-type engines lent themselves so readily to adjustable fuel controls that the importance of specific information on the engine control characteristics was underestimated and considered more or less a secondary problem.

The current development trend is toward larger and more efficient engines which generally employ an axial-flow compressor. Engines of this type are rapidly approaching the design objectives set up, but they appear to have one undesirable trait in common, that of compressor stall. Stall can be broadly described as a drastic reduction in the quantity of compressor air delivery which suddenly

THIS paper describes the equipment and specific technique used for rapidly determining the shape and location of the compressor stall area. Complete fuel feed analysis for optimum performance operation at sea level can be accomplished in 4 hr. Information is presented on the tests conducted to date, and the general trends observed regarding compressor stall.

The instrumentation used consists of a direct writing rectilinear plotting board and a universal electronic fuel control. The fuel control permits the selection of any desired acceleration fuel schedule and thus provides a medium for an exact examination of engine characteristics.

When selecting an acceleration fuel schedule, consideration must be given to the shift in the stall area caused by throttle manipulation and past engine history. Illustrations are presented to show the variation between ideal and practical fuel schedules.

The effect of ambient temperature on compressor stall is discussed. Theories on steady-state fuel flows have been substantiated, but some question exists as to the validity of computation under transient conditions. Comparison is made between the calculated and measured fuel flows for two modern axial-flow-type engines.

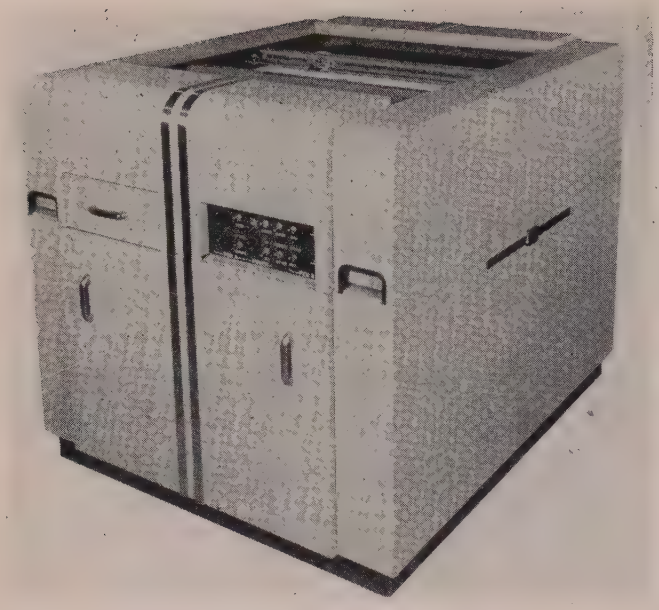


Fig. 1 – Photograph of a pen plotting board

occurs when the back pressure exceeds some critical value. (Analysis of compressor stall is an extremely complex study. This simple explanation is to acquaint the reader with effect of total stall which is necessary for background and paper continuity.) Increasing the back pressure by burning fuel or by throttling the compressor beyond the critical point causes compressor blade cavitation, and consequently very low air-pumping efficiency. The existence of a stall area in the transient operating range of an engine usually lengthens the time required to accelerate the engine from idle to maximum speed, and a condition of dangerously high burner temperatures will be encountered, if the fuel flow is not reduced the instant stall occurs.

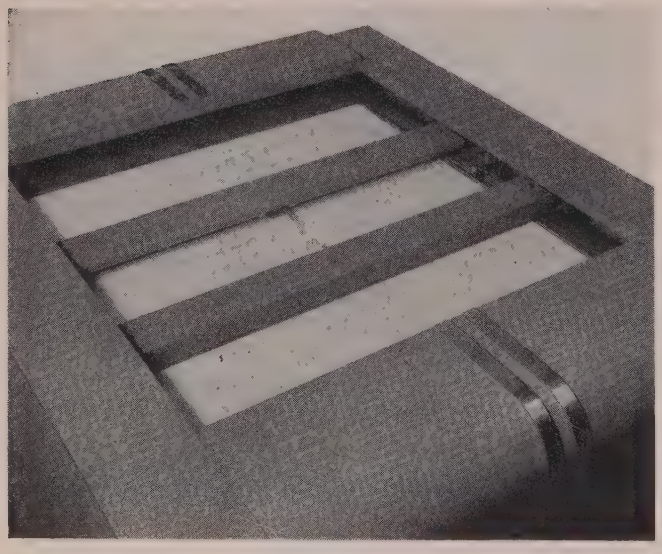


Fig. 2 – Another view of pen plotting board shown in Fig. 1

As a general rule, the fuel control is expected to incorporate the necessary mechanisms to permit safe accelerations without entering or causing stall. An automatic stall detector would be most desirable, but the results obtained to date have not been entirely satisfactory and much more development is required. Therefore, some method of scheduling is usually incorporated in hydraulic or electronic controls to lower the acceleration fuel flow near the stall area. The problems of contouring the fuel control schedule is in itself difficult, and the situation is made more complex by the lack of information concerning the location, shape, and changes of the stall area.

Most assuredly, an ideal engine control cannot be made a reality until accurate information on the engine characteristics is made available to the control designer. Until recently designs were predicated almost entirely on classical or computed data, such as parameter plots or compressor maps. Information of this type is usually derived from individual compressor or turbine tests and theory, with little regard for the overall integrated engine assembly and efficiency effects.

The complexity of the stall phenomena, together with the lack of design data, has made the control manufacturer's position difficult. Engineers were hard pressed to provide even a workable control for a complex engine, no less a unit which can operate the engine at maximum performance. It was felt that the importance of the problem justified organization of a competent and separate instrumentation group, which was placed on an equal status with the balance of the engineering organization. Although the group performs a service function, it nevertheless has the responsibility of allocating funds, securing equipment and outlining measurement and test procedures to be followed. As a matter of interest, the operating efficiency of this organizational arrangement is rather good, and it has many desirable aspects compared to instrumentation service at the technician level.

This group has recently made what seems to be an important contribution by developing equipment which for the first time rapidly and accurately determines the real location of the compressor stall area, and also records other pertinent information on engine fuel requirements under transient operating conditions. This special equipment is proving valuable today, and it is believed will be indispensable for testing future engines involving complex functions such as variable nozzles and afterburners. This paper is not intended to describe electrical circuit details, but rather to outline the overall equipment, operating techniques, and test results obtained. In all probability, information on the electrical circuits will be presented at a later date in such a manner as to be useful to those familiar with the art.

The types of instruments initially procured for carrying out the measurement program were rather universal in application and conventional in nature. Included were such items as recording string galvanometers, photographic camera oscilloscopes, direct writing equipment, and most commercially available pickups. The technical results obtained were satisfactory and very high speed phenomena could be observed; but the details associated with setup, calibration, and data reduction left much to be desired. It was also found necessary to replot most of the information in order to obtain the desired relationship between variables.

When a new experimental engine is first placed in operation, the time permitted for control evaluation purposes is usually small since preference is given to obtaining information on the mechanical and efficiency characteristics of the engine. This fact, coupled with the delays incurred by the measurement system, severely limited the number of tests that could be performed. This experience indicated that engineering surveys could not be carried out on a practical basis when the results of one test had to be developed and plotted before being able to make readjustments and succeeding runs.

It was obvious that the techniques and equipment used were not compatible with the test time available, and that more efficient use of operating time was required.

As an ideal solution to the problem was not readily evident, Bendix decided to sponsor a 2-phase development program which appeared to have several major advantages. The program adopted required, first, the development of a very accurate, medium fast, direct-writing recorder system which would present data directly on X and Y coordinates. Second, the design of a truly universal fuel control that could operate the majority of jet and turboprop engines under development.

Plotting Board System of Recording

Most of the electronic recording devices available are of a type which present one or more channels of information with respect to a time base. They usually have a small scale deflection, and some of them require a photographic process to develop film after recording. To serve the needs it was desirable to record two independent variables simultaneously, and therefore a special instrument was required. It was known that navigational position plotters have been in use for many years, and although slow in response time, they represented the type of machine needed. Normally, these equipments are used to show the instantaneous position of a ship in terms of latitude and longitude.

Initially, consideration was given to designing a high-speed position plotter; however, a commercial unit became available which had most of the qualities deemed necessary for this type of work. Therefore, several plotting boards were purchased and

The Author

W. J. KUNZ, JR. is chief engineer of electronics, Fuel Metering Engineering, Bendix Products Division, Bendix Aviation Corp. He is in charge of the development work on electronic fuel controls and instrumentation applicable to gas turbine engines. Mr. Kunz received his electrical engineering degree from Tri-State College.

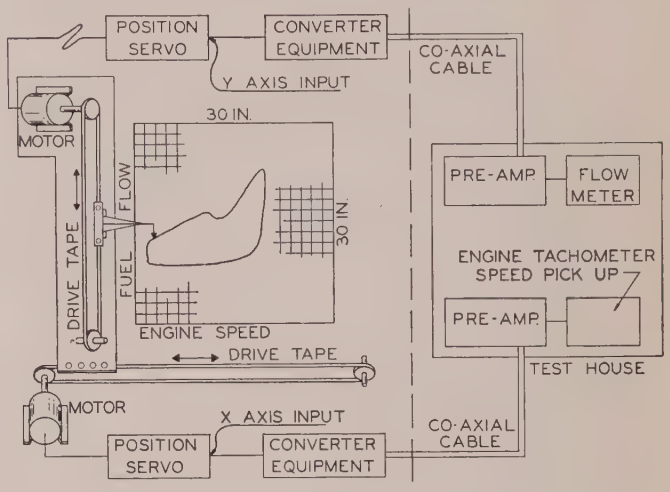


Fig. 3 – Block diagram of plotting board system for measuring engine speed and fuel flow

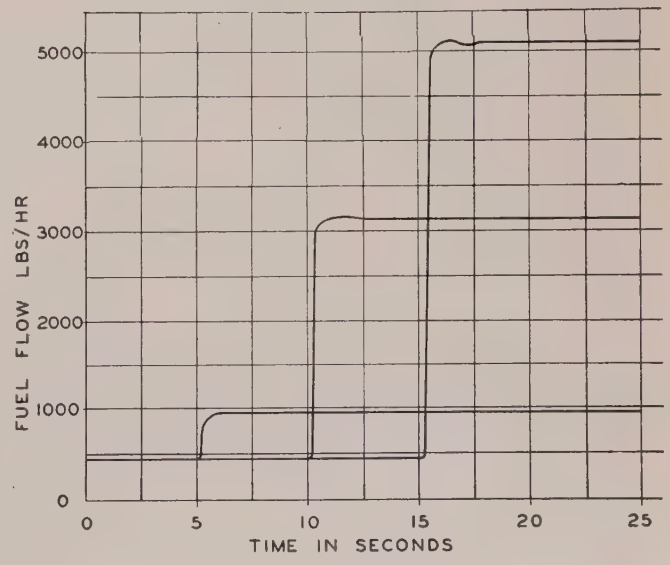


Fig. 4 – Plotting board response to step input changes of fuel flow

Fig. 5 – Photograph of universal test control

placed in operation at the South Bend and customer plants. Each unit is capable of drawing a graph 30 x 30 in. in size, which is large enough to permit accurate interpretation of the graphic information. The plotting board proper, when operated within its specification limits, has dynamic and static inaccuracies of less than 0.1%. Figs. 1 and 2 are photographic views of one of the improved design plotting boards currently being used. This particular unit has two pens for presenting data on $X-Y$ and $X'-Y'$ axes simultaneously.

For explanatory purposes, a single pen board basically consists of two precision electrical servomechanisms which direct the X and Y travel of the writing pen. The pen position on either axis is proportional to the magnitude of the input voltage being applied to that axis. As the plotter is only capable of recognizing d-c input voltages, special

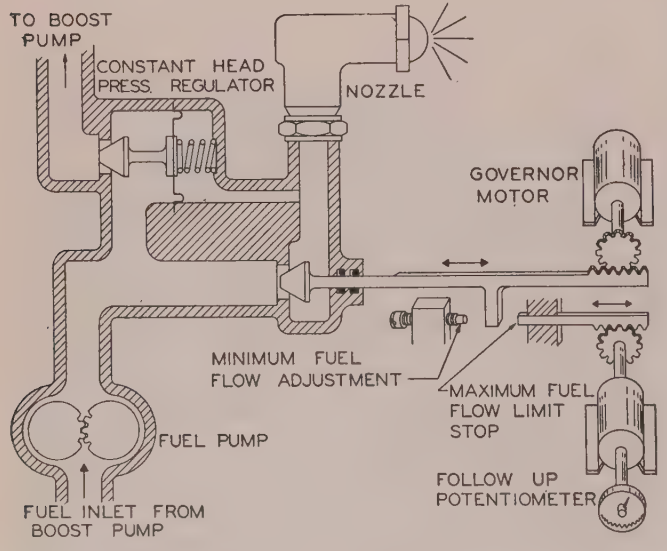


Fig. 6 – Schematic drawing illustrating operating principle of universal test control valve mechanism

Table 1 – Additional Recorded Data

Measurement	Approximate System Inaccuracy, Including Plotting Board
Engine speed	¼%
Fuel flow	½%
Engine temperature	10 F (With respect to thermocouple emf)
Gas and fluid pressure	2% usually limited by the pickup
Mechanical motion of levers	Usually highly accurate – depends on type of pickup
Time	1/10%
Voltage (a-c)	½ – 1½%

intermediate conversion equipment was required to permit operation from various types of available pickups.

Exacting engineering of the conversion equipment circuits was necessary to maintain a high order of accuracy in the entire recording system. Fig. 3 shows a block diagram of the plotting-board system as set up for measuring engine speed and fuel flow. In addition to the measurements shown in this block diagram, other equipment has been designed to permit picking up and recording the data listed in Table 1.

The equipment is proving valuable for engineering analysis in conjunction with hydraulic and electric fuel controls, flow benches, engines, and miscellaneous work. In some cases, the recording is done at distances of 700 ft from the test position. The use of radio telemetering techniqus would, of course, permit airborne test programs to be recorded on the ground with immediate interpretation of results. Plans for a complete air-ground instrumentation system are currently being considered.

The determination of the characteristic response time of the recorder, conversion equipment, and pickups is a rather involved study. Plotting board response speed cannot be defined as a simple time constant relation because of the limiting slew speed of the pen drive motor. For example, on the Y axis the pen can accelerate at a rate of 350 in. per sec per sec until a slew speed of 30 in. per sec is reached. A detail study of these characteristics is important and should be made when considering the particular parameter to be recorded.

Fuel flow measurement is usually considered a difficult problem, and will be used as an illustration of recording characteristics. Fig. 4 shows the system response to step input changes of fuel flow.

The author would like to summarize this section and emphasize the following points:

1. The plotting board system of recording is a most efficient means of recording data on two variables.

2. Engineers consider this recorder a valuable tool as it permits immediate visualization of information, and on the same chart, can show the effects of various control adjustments.

3. The equipment holds calibration for relatively

long periods of time, (4 to 8 hr) and does not require pampering by operating personnel.

4. This equipment is *not* suitable for high response rate transient studies, but has a definite place in industry if used within its known and definable limitations.

Universal Fuel Control

The design of a universally applicable fuel control has been the subject of debate for many years. The control to be described is, of course, in the embryo stage; nevertheless, it provides an interesting preview of the operation that may be available with advanced controls embodying extreme flexibility.

The philosophy underlying the design of this test control was to develop a unit which was capable of operating any unknown engine to the maximum performance ratings, without requiring machine-shop or design adjustments. Furthermore, the system selected should be positive in operation and permit repeatable performance from run to run. Once an engine is operated satisfactorily throughout the steady-state and transient ranges, it is then a simple matter to determine the characteristics and interrelations of many other parameters available for control purposes. Therefore, the type of control selected for the initial engine operation is only a means to an end, and does not necessarily constitute the most ideal final control for that particular engine.

The control system chosen utilizes an all-speed isochronous type governor for steady-state operation of the engine, and a system of speed-scheduled fuel flow for acceleration control. The electronic

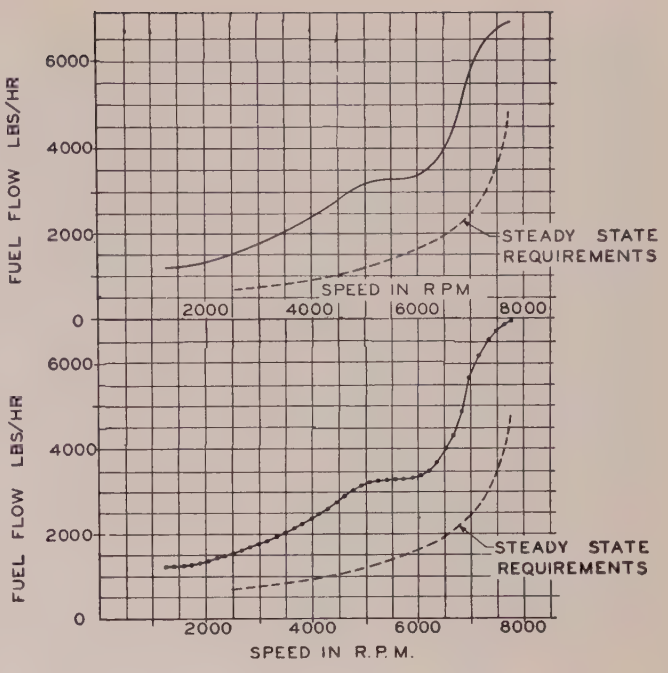


Fig. 7 – Upper: illustration of an ideal type of acceleration curve. Lower: step or increment method of curve approximation

medium was selected for system operation because of its inherent flexibility, accuracy, and the minimization of special shaped engine mounted components. The only engine connections required for the operation of this control are those of a tachometer speed pickup, and the fuel lines to the nozzle assembly. Fig. 5 is a photograph of the completed test control.

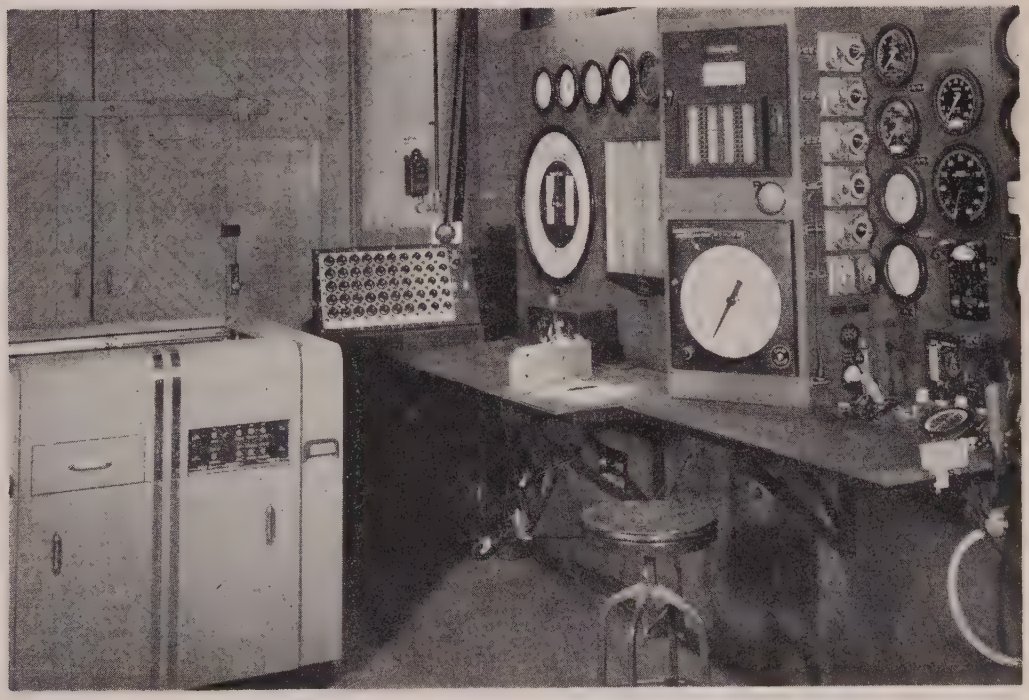


Fig. 8 – Photograph of integrated equipment used for experimental engine testing

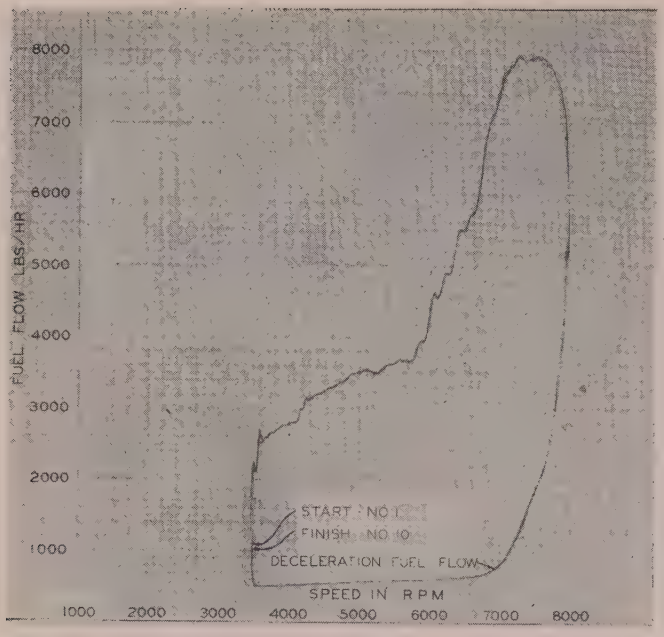


Fig. 9 – Photograph of 10 repeat accelerations as recorded on plotting board

Fuel Valve and Governor

One of the most important elements of the control is the main fuel valve assembly. This unit consists of an electric motor-driven valve which operates freely between two adjustable mechanical limit stops. The fuel pressure across the valve is maintained at a constant value, consequently, the engine fuel flow is directly proportional to the metering valve area. One of the limit stops is used for controlling the minimum fuel flow to the engine and is a screw-driver type adjustment. The second limit stop controls the maximum permissible valve opening, and is positioned by an electric servomotor. Fig. 6 illustrates the operating principle of the valve mechanism.

To govern the engine speed during steady-state conditions, it is only necessary to apply a suitable electrical control signal to the fuel valve motor. The governor circuit used in this instance is precise in nature and measures instantaneous engine speed through the use of an engine-mounted a-c tachometer generator. The frequency of the tachometer a-c voltage is measured and is used to produce a d-c voltage which is directly proportional to engine speed. This d-c speed voltage is then compared to a throttle-selected reference voltage which represents the desired engine operating speed. The difference between the throttle and speed voltages is fed to the input of a servoamplifier which in turn operates the motor connected to the fuel valve. Suitable stability networks are incorporated to provide optimum dynamic control of the system loop.

Whenever the engine is below the selected speed, the fuel valve operates in an opening direction until it contacts the acceleration limit stop. Thus, the instantaneous fuel supplied to the engine during accelerations is controlled by positioning the limit stop. It is interesting to note that with this system there is complete isolation between the governor and acceleration elements.

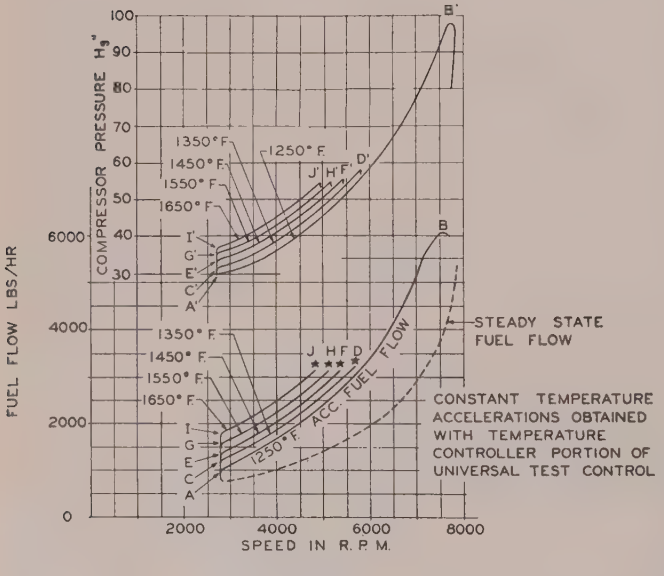


Fig. 10 – Typical curve information obtained when conducting compressor stall investigation (low rpm boundary limits)

Acceleration Control

To achieve maximum acceleration performance from an engine, it is necessary to inject the exact and proper amount of fuel at each instantaneous value of engine speed. The fuel values required vary with engine types and operating conditions, therefore, the flexibility of adjustment of the acceleration fuel feed determines to a large extent how universally the control can be applied.

The control of acceleration fuel in a smooth continuous manner is most ideal however, it is practically impossible to achieve such scheduling and still permit accurate curve adjustment. Components such as tuned circuits, resistor capacitor filters, and so forth, are not flexible enough to provide exacting control. A suitable solution to the problem consisted of dividing the speed range of the engine into a number of finite increments, and then using precision potentiometer controls at each of the individual increments. In the present universal control, the engine speed range has been divided into fifty increments, or in other words, fuel feed is adjustable from zero to a maximum value every 2% of engine speed. Fig. 7A illustrates an ideal curve, and 7B the step or increment method of approximating the same curve.

The control of the individual steps is accomplished with two positional electronic servos. The first servo rotates a 50-contact switch shaft to an angular position corresponding to the instantane-

ous value of engine speed. Thus, for a given engine speed a known contact is selected. Each contact is connected to an electrical potentiometer, the adjustment of which will determine the desired area opening of the fuel valve. The second servo then positions the valve limit stop to the value required by the above potentiometer.

Initial Engine Tests

The initial engine tests of the fuel control proved that it was capable of extremely flexible and precise performance. The definition effects of the fuel flow increments was not objectionable and repeatability was practically perfect. The potentiometer dial calibrations can be relied upon to approximately ± 3% of the desired value of fuel flow.

One major shortcoming which became evident when running new engines, was that 25 to 30 accelerations were necessary before an ideal maximum engine performance setting could be obtained. The majority of the accelerations were required to obtain the exact fuel flow settings at low engine speed where stall was no particular problem. The excessive number of accelerations required was not considered desirable in view of the initial objective of keeping engine test time to a minimum.

The overall operation of the control was materially improved by adding a temperature controller unit to the fuel valve, and temporarily disconnecting the scheduling mechanism. When in operation, the temperature control supplies the correct amount of fuel to accelerate the engine at whatever tailpipe or burner temperature value is selected by the operator. By reviewing the fuel flow values obtained with the temperature unit, an initial setting for the schedule control can be determined. The temperature unit also permits the rapid determination of the surge or compressor stall region of the engine. The technique used will become evident in later explanations.

It should be noted that the temperature acceleration portion of the control is used as an expedient only, and does not enter into the exact determination of the final fuel flow schedule.

Integral Equipment Setup and Accuracy

The operational characteristics of a new engine are determined with the aid of both the plotting board and the universal test control. The control is connected to the engine and the maximum and minimum speed adjustments are made. Usually tailpipe thermocouples are connected to the temperature controller, and the plotting board is arranged to record the fuel flow versus speed and the compressor discharge pressure versus speed relationships. The measurement of compressor discharge pressure is used to indicate the point at which compressor stall occurs. Additional supplemental data is also recorded but only on the final run, when the engine is operating at the maximum performance rating. Equipment used for these measurements is dependent on the particular parameters selected, and can be of the oscillographic type as the information recorded is not immediately necessary and has no direct bearing on the test control performance or adjustments. Fig. 8 is a photograph of the integrated equipment used for experimental engine testing.

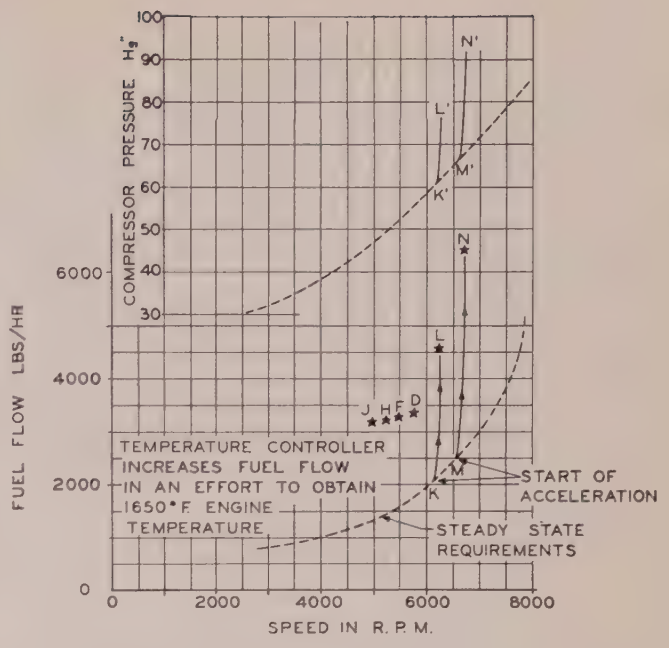


Fig. 11 – Method of determining high rpm limits of compressor stall

The combined errors of the overall system, including the fuel control, pickups, and plotting board recorder, usually average less than ± 1½%. Fig. 9 is a photograph of ten repeat accelerations as recorded on the plotting board.

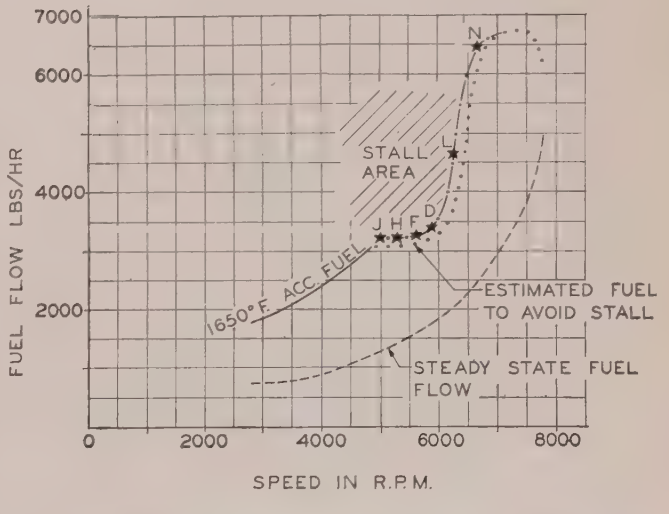


Fig. 12 – Composite information on stall area prior to determining final engine acceleration schedule

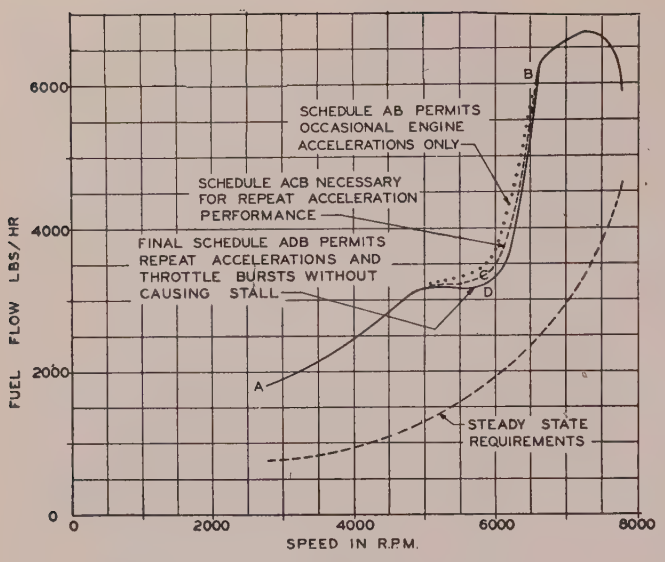


Fig. 13 – Illustration of fuel flow progression required to define a practical engine acceleration schedule

Engine Stall Determination

The procedure outlined is currently in use and requires approximately seven accelerations to define the shape and location of the compressor stall area. The tests are divided into two parts, the first being the determination of entering or low rpm points of the stall area; and second, defining the high rpm limits or trailing boundaries. With the equipment setup as previously described, the following tests and adjustments are made:

(1) The engine is brought up to operational idle speed. The scheduling portion of the universal test control is removed from the circuit and the temperature controlling amplifier is connected to the main fuel valve. The temperature controller is adjusted to some low value just above the required to run temperature of the engine, for example 1250 F.

(2) The engine is then accelerated from idle to maximum speed. One pen on the plotting board records the fuel being supplied to the engine to maintain the 1250 deg selected temperature value. The second pen plots the compressor discharge pressure with respect to engine speed. Typical curve information obtained in this test run is shown in Fig. 10, curve *AB*. Usually on a low temperature acceleration such as 1250 deg no stall is encountered.

(3) The engine is returned to idle speed and the temperature setting value of the controller is increased to 1350 F. The engine is again accelerated to maximum speed and recordings are made on the plotting board. If stall is encountered, the exact point of entry can be noted by the sharp decrease in compressor pressure as illustrated in Fig. 10, curve *CD*. (Stall could be defined by sharp reduction in compressor pressure in all engines tested to date. However, if this relation does not apply universally, some other parameter could be selected as an indication of stall.) The exact fuel flow value and speed at which stall occurred can also be determined by referring to the plotting board graphic information. The fuel flow value thus derived is used as one point for defining the complete shape of the stall area.

(4) Accelerations are repeated similar to (3) above, except that the temperature setting of the controller is raised in 100 deg increments until the engine is operating at its maximum safe temperature limit. The fuel flow values required to cause compressor stall are noted, and will form a pattern as shown in Fig. 10, lines *EF, GH,* and *IJ*. Five accelerations usually suffice for determining this part of the stall area.

To complete the second phase of the stall investigation, it is necessary to define the high rpm limits of the area. This is usually accomplished as follows:

(1) The temperature amplifier is set to the maximum value permissible for engine operation, for example 1650 F. The engine speed is adjusted by means of the throttle to a speed which is slightly higher than the previously determined speed information on stall, and the engine is then accelerated from that speed point to maximum. During acceleration, the fuel being fed to the engine increases rapidly in an effort to bring the engine up to the temperature demanded by the temperature controller. Stall may be encountered at some point on the fuel flow rise curve, as illustrated in Fig. 11, curve *KL*.

(2) The same procedure as above is repeated with one or two more accelerations and each time the speed from which acceleration starts is increased. This procedure will net a series of stall points as illustrated in Fig. 11, lines *KL* and *MN*.

Two accelerations are normally used to determine

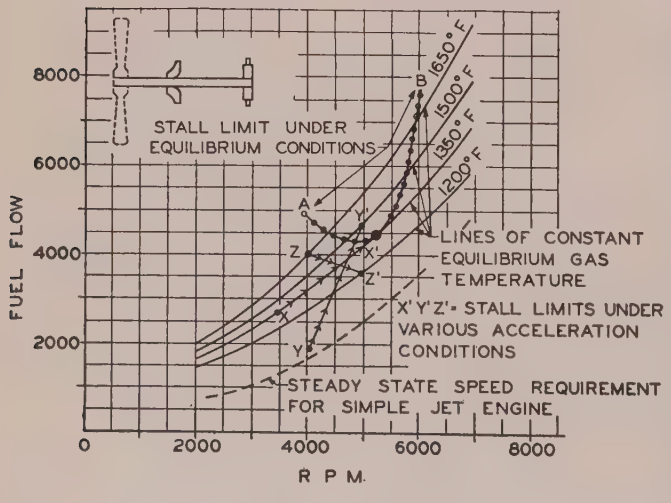


Fig. 14 – Graph showing influence of acceleration method on compressor stall

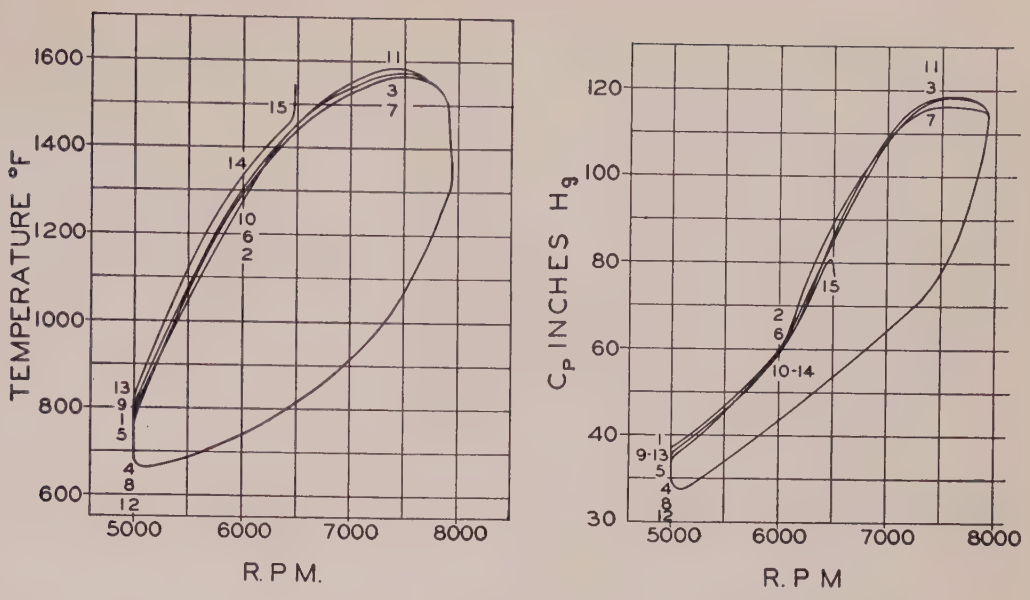


Fig. 15 – Effect of burst accelerations on engine tailpipe temperature and compressor pressure values

the high rpm boundary limits because the stall effects in this region are extremely violent and the engine could sustain structural damage. The seven engine accelerations described provide a reasonably good definition of the entire stall area shape and location, plus approximate fuel flow information for operating the engine at maximum temperature. (See Fig. 12.)

The stall area now defined is, of course, valid only for the environmental conditions under which the test engine was operated. It is well known that changes in ambient temperature and other engine effects will cause the stall area to shift position. Fortunately, these tests can be carried out in a short period of time, and therefore it is a simple and practical matter to conduct other test programs with the engine operating under various ambient conditions. Our tests also show that when utilizing temperature control during acceleration, the selected value of engine temperature is not exceeded even when stall is encountered. In some cases, the engine may accelerate right through the stall area; in others, the engine will operate in a steady-state manner at the constant selected temperature value. Whether or not an engine will continue to accelerate when operated in the stall region is dependent entirely on the shape of the area and the particular engine characteristics.

Determining Engine Acceleration Requirements

The next tests in the program are for the purpose of determining the complete acceleration fuel requirements for the engine. The temperature controller is now removed from the circuit and the fuel flow is metered by the scheduling portion of the universal control. The fuel flow values obtained during the stall investigation are used as preliminary information for adjusting the various dials on the fuel control. A schedule is selected that permits accelerating the engine at maximum temperature except at the stall area. The quantity of fuel in this region is usually reduced 3% below the values which are known to cause stall.

After the schedule has been set up, the engine is accelerated from idle to maximum speed. The fuel flow versus speed information is again recorded on the plotting board, and the dial adjustments are trimmed in the event any irregularity is noted. Usually two accelerations are required before a smooth contour is obtained. The engine under test is now being operated at its maximum transient performance, as limited by the permissible exhaust gas temperature and maximum fuel to skirt the compressor stall area. To all outward appearances, the final and ideal schedule has been secured. Unfortunately, this schedule will not permit practical operation of the engine in flight service. (Fig. 13, line *AB*.)

During our operational tests with ideal schedules, we first observed that a change in fuel flow requirements was necessary to prevent stall under different throttle manipulations. It was found that if repeat accelerations were made in close succession, the engine would stall after the second or third acceleration. The fuel schedule can be modified to permit satisfactory operation if it is contoured as shown in Fig. 13, line *ACB*. In practice, we select a schedule which will permit safe operation with 10 accelerations. For this evaluation the engine is accelerated to maximum speed, decelerated to idle and returned to maximum speed as quickly as the engine will follow, without permitting any steady-state operation.

Further investigation of practical engine operation with the latest repeat acceleration schedule revealed that there was still a third stall effect which could be caused by burst accelerations. With

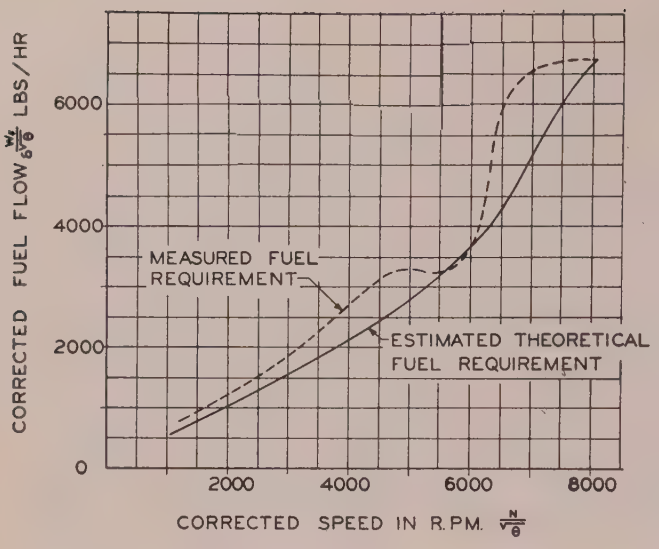


Fig. 16 – Illustration showing variations between estimated and measured acceleration fuel requirements for a modern jet engine

the schedule as determined above, an engine could be made to stall if the acceleration is started from some midspeed point rather than idle. This condition is similar to what a pilot might experience if he approached an airport with the throttle partly retarded and then required full power to comply with a "wave-off", or to miss some object.

The fuel schedule must be further revised and usually requires leaning out as shown in Fig. 13, line *ADB*. This last schedule is considered final and will permit safe engine operation regardless of the method of throttle manipulation.

The test program is now completed and all pertinent information is available. Numerous engine tests have shown that the entire stall and fuel flow schedule determination can be accomplished in a period of two to four hours.

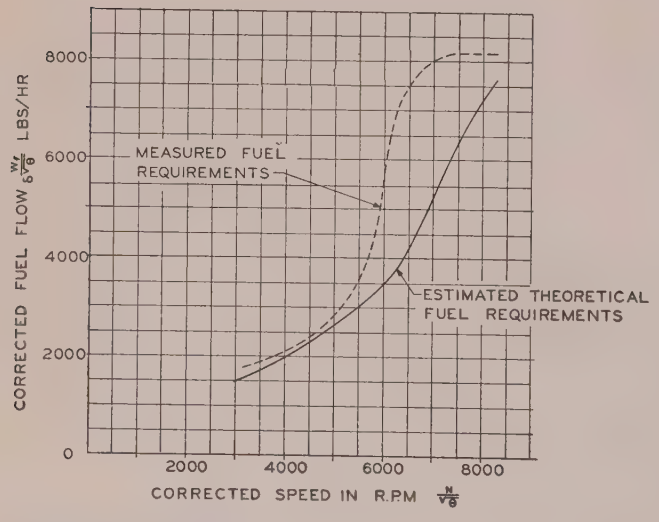


Fig. 17 – Another variation between estimated and measured acceleration fuel requirement for modern jet engine

Compressor Stall

The previous test results have shown that throttle manipulation influences the location of the stall area, and that compensation of the acceleration fuel feed is necessary to achieve satisfactory engine performance. Mr. Frank Mock of Bendix Products Division, has studied the observed phenomena and he offers the hypothesis that the fuel flows required to produce stall can logically be expected to change, if different acceleration methods are used. Compressor stall appears to be influenced by the residual heat of the engine, and may be associated with the expansion effects of the various internal metal parts.

To illustrate the conditions that might exist, assume that we have a jet engine with an extended power shaft on which is mounted a brake or propeller. One could then maintain a fixed equilibrium condition of rpm, fuel feed, and normal turbine gas temperature. (This latter is defined as the temperature which would result directly from fuel combustion.) A chart, as shown in Fig. 14 could then be plotted which would carry lines of normal turbine gas temperature plotted on the axes of fuel feed and rpm. Assume that the engine is operating at a constant inlet air temperature condition. The individual stall points could then be determined for equilibrium gas temperature conditions, as shown by the circles on line $A-B$. This curve would be derived by holding the speed constant at each point by means of the propeller or braking device, and slowly increasing the fuel until stall is encountered.

If the brake is now removed from the engine and a constant temperature acceleration is made along the line X to X^1, it then seems logical that the engine would enter stall at the previously determined equilibrium point X^1. If the time required to accelerate from X to X^1 is of sufficient magnitude, then the entire engine would have reached a relatively constant equilibrium temperature and the stall point at X^1 should be identical with that obtained with the propeller brake. However, if the acceleration is made in a different manner, for example from a point of high temperature, Z to a lower temperature Z^1, then one might expect to find a lower value of fuel flow necessary to prevent stalling the engine at point Z^1. Because of thermal inerita, the actual temperature of the metal parts would be greater than the instantaneous gas temperature at Z^1. This explanation is substantiated by the repeat acceleration tests described earlier. It will be recalled that a lower schedule was necessary to circumvent stall.

Conversely, it might be expected that accelerations from a low temperature region, Y, to a higher temperature point, Y^1, would permit feeding more fuel to the engine without causing stall than would

be apparent from studying the equilibrium line $A - D$. However, this has not proved to be the case on burst accelerations made when the engine was in a decelerating condition. Possibly the added fuel required to stop the deceleration and cause acceleration resulted in sufficient heating to nullify the high fuel flow values indicated on line $Y - Y^1$.

Although more tests are required on the stall area change phenomena, it is nevertheless clear that there are differences between equilibrium and transient fuel flow values which cause compressor stall. There appears to be a direct relation between the past history of engine acceleration, the temperature of metal parts and the ease with which stall may be induced.

The effect of burst accelerations on engine tailpipe temperature and compressor pressure values are shown in Fig. 15. Both of these parameters reflect to some extent the past engine history and as such, should prove useful for control purposes.

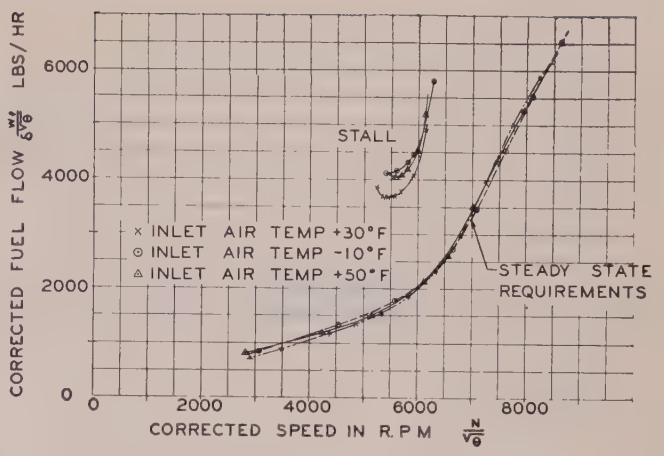


Fig. 18 – Parameter plot of compressor stall under varying ambient temperature conditions (stall defined by temperature acceleration method)

Estimated and Measured Fuel Requirements

The number of engines tested to date has been limited, and consequently very little information is available to make an accurate comparison between estimated and measured acceleration fuel requirements. Figs. 16 and 17 illustrate the degree of variation on two modern engines for which data was available. The estimated curves in each case were received early in the engine development program. The final measurements were made later when engines became available, and thus reflect some fuel changes caused by engine modification or improvements. The fuel controls initially procured are expected to accommodate schedule changes by adjustment or simple modification. Inspection of the curve discrepancies existing, clearly illustrate why the control manufacturer may experience difficulty in providing such a wide degree of flexibility in his product.

Stall Area Shift and Ambient Temperature Change

The discrepancies existing between theoretical and actual engine characteristics stimulated investigation of the stall area shift attributable changes in ambient temperatures. To obtain this information, a test program was arranged where a modern engine was first operated at an ambient temperature of 30 F. Measurements were made on the stall area, steady-state, and the acceleration fuel flow values, as shown on the parameter plot in Figs. 18 and 19. Unfortunately, damage to the engine occurred during the latter part of this test and it was necessary to replace the combustion liners. As near as could be determined, the replacement liners, although different in design, did not affect the overall static engine performance and the test program was continued. The engine was then operated at ambient temperature of –10 F and 50 F. The same measurements were taken and the data plotted again on the parameter curve.

The steady-state fuel flow values are quite consistent with, and appear to substantiate, the existing theory of a common parameter plot line. The acceleration and stall values, however, vary quite widely.

The instrumentation group has observed similar shifts in stall characteristics as an unofficial part of other engine measurement programs. As we are at a loss to explain the discrepancies existing, at this time, we can only offer the following possibilities for consideration:

1. Severe changes in compressor stall can be caused by replacing an engine element such as combustion liners. (This would have serious practical effects on attempting to build mass production en-

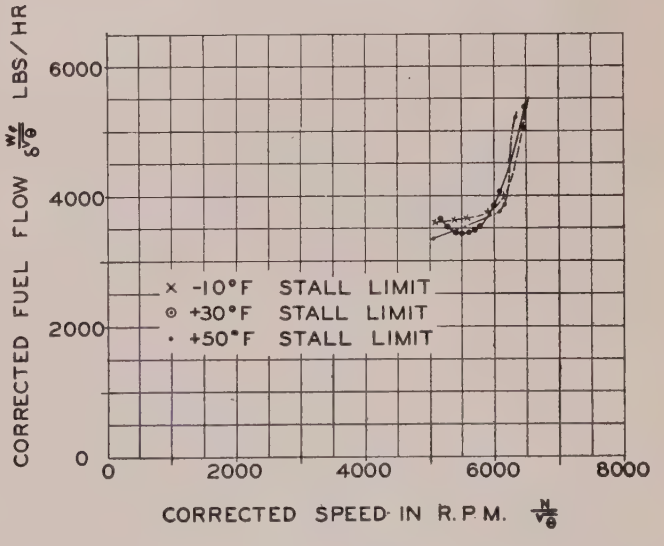


Fig. 19 – Parameter plot of compressor stall (stall defined by operating fuel flow schedules)

gines and fuel controls which do not require matching.)

2. The fundamental relations which describe compressor stall are not valid when the compressor is used as part of an integrated engine assembly, and that a point of inflection in the stall characteristics occurs under ambient temperature change, possibly influenced by engine efficiency changes.

3. That our instrumentation and measurements, which have heretofore been satisfactory, were inaccurate in this case. We were successful, however, in checking the theoretical steady-state values almost exactly.

The implication that engine stall characteristics do not follow theory, or that stall is affected by component part changes of the engine, should be further investigated. In all probability, information of major interest would be obtained if various engines, and also engines of the same model, could be tested under different temperatures. In the event that engine efficiencies are responsible for stall changes, additional information could be secured through the use of an airborne universal test control.

Conclusions

The equipment described has been in existence only a short time, and the test programs were limited by the availability of engines and pressing production problems. It is nevertheless feasible to conclude the following:

1. The location of the compressor stall area, as shown on a plot of fuel flow versus rpm, is influenced by the manner in which the engine is accelerated and also by the past history of engine operation. More than ever there is need for some form of automatic stall detector.

2. It is impractical to operate a jet engine with fuel flows scheduled very close to the stall area unless suitable compensation is made for the shift in the stall area location caused by throttle manipulation.

3. The compressor pressure and tailpipe temperature appear to reflect past engine operating history, and should prove useful for control purposes. Automatic temperature control of the fuel feed will prevent damage to the engine attributable to heat if compressor stall is encountered.

4. Theory, as represented by the basic parameter plot, appears substantially sound with respect to steady-state fuel flow values and changes in ambient temperature.

5. Inconclusive information is available on the changes in location of compressor stall at various inlet air temperatures. A questioning of the validity of our existing theories on integrated engine stall is indicated.

6. When compressor stall is encountered on the high rpm side, the reaction is most violent, and under certain conditions can be combined with rich blowout.

7. The universal fuel control and plotting board equipments provide an efficient method of determining new engine fuel flow characteristics. The equipment is also proving useful for solving difficult production fuel control and engine problems.

8. Engines tested to date displayed a definite stall notch which was clearly definable by the equipment and technique outlined.

9. This equipment can be used for determining other engine characteristics, such as starting, deceleration fuel feed, and effects of bleedoff and variable nozzle operation on acceleration fuel requirements.

ORAL DISCUSSION

Mr. Madrackus, General Electric Co.: How do you measure fuel flow?

Reply: We have used three methods:

1. Wobble-plate meter (no longer used).
2. Magnetic pickup.
3. Photometric pickup.

The pickups are commercially available. Their time response is on the order of 0.05 sec; they are heavily damped.

G. W. Newton, ARO: Your instrumentation certainly lends itself to fast checking of results. Arnold Laboratory is going to similar equipment.

What kind of pressure transducers do you use?

Reply: We have used strain-gage pickups and a variable-reluctance pickup, both commercially available.

Mr. Newton: How about their transient response?

Reply: We do not consider their response fast enough for explosion-type rates of pressure change. The response is in the neighborhood of 3-5-8 cps. It is satisfactory in view of engine speed and acceleration characteristics. Both the rotor and the fuel flow have inherent inertia.

D. J. Nolan, Allison Division, GMC: The reason for the need of the data gained is this: the military want a faster acceleration from idle to take-off on the ground. The aim is 20 sec.

You can dump a lot of fuel in at the beginning of the acceleration, back off at the surge knee, and then increase again afterwards to fit the whole sequence into 20 sec. But this is difficult to do mechanically in the control mechanism.

There are, the audience might like to know, transducers that can read the initial slight fluctuations that presage surge even before the surge effect is noticeable on compressor temperature and pressure.

Reply: Yes, there are stall-detecting transducers. Our equipment differs in that it aims to define the boundaries of stall on the pressure ratio versus fuel flow chart.

Frank Mock, Bendix Aviation Corp.: It is true that at sea level you can dump in almost any amount of fuel, provided you are not at the stall knee. But if you do it at 20,000 ft and around 1600 F tailpipe temperature, don't you get combustion surge or rich blow-out?

Mr. Nolan: You're right about the altitude conditions. But acceleration is already fast enough there. The need is for shorter accelerations at sea level.

Reply: Let me emphasize that it's a big help to the control manufacturers to know the engine's stall characteristics early in the design of the control.

D. Q. Marshal, General Electric Co.: How far along are you in making this equipment airborne or telemetering the information?

Reply: Telemetering is coming along. It could be made airborne, in fact, it is already reasonably light in weight.

PLASTIC PROTOTYPES
Revolutionize Preparation for Manufacture

Wallace A. Stanley, Ford Motor Co.

This paper was presented at the SAE Summer Meeting, Atlantic City, June 3, 1952.

THERE has taken place recently a mild revolution in the medium by which manufacturing engineers have been able to work more quickly and more closely with product designers.

This new medium is the *plastic prototype* – made for the sole purpose of analyzing manufacturing problems in advance, at the design stage. A prototype for those not fully acquainted with the term, is an original, or first model, or pattern from which products are copied.

Designing engineers should always discuss problems encountered by manufacturing to assure that each succeeding product design eliminates or reduces as many of the foreseeable manufacturing difficulties as possible.

Product design, normally the responsibility of

ONE plastic model is worth 40 lb of blueprints and 40 hr of the explaining that goes along with the prints, according to engineers who have worked with the new plastic "toys" which can serve as perfectly scaled miniatures for virtually every phase of automobile design and manufacture.

The extensive benefits occasioned by this revolution in methods may be summarized thus:

1. Shortens design development time by providing a third-dimension evolvement of structure and form.

2. Used in advance discussions for compromising engineering and manufacturing problems, showing construction so clearly that troublesome problems are foreseen; thus enabling clear-cut, reliable decisions with a minimized chance of encountering major revisions.

3. Show in one minute what could not be found on prints for hours, saving time over any other methods, while generating valuable counter-proposals reducing costs, operations, man-hours, and so forth.

4. Parts can be made in plastic faster than drawings with detailed dimensions.

5. Duplicates can be obtained in a matter of minutes.

6. Used for processing, reducing manufacturing operations; in standardization.

7. Gives entire tooling programs a start months ahead of ordinary procedures, saving thousands of hours of tool and fixture design, and build time.

The Author

WALLACE A. STANLEY is manager of the Forward Products Department, Ford Motor Co. He was doing consulting work on body construction for Ford, when he joined their Manufacturing Engineering Staff.

engineering departments for function, appearance, performance and several other factors, can be resolved in many optional designs, all perfectly acceptable or desirable from the engineer's viewpoint.

But from the manufacturing viewpoint, the choice usually narrows down to one, after the determination has been made of cost, manufacturing processes involved, availability of machinery or facilities, labor requirements, and others. It is entirely possible, for example, that in producing stampings the metal drawing might be acceptable, while other factors such as material handling difficulties could prove too costly and even impractical.

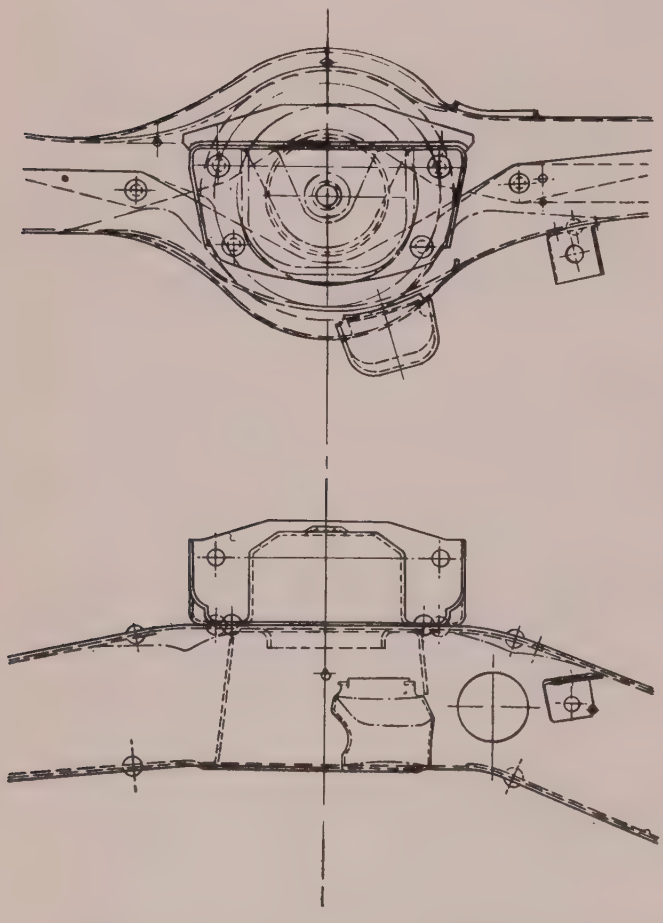


Fig. 1 – Typical chassis assembly drawing showing 7 parts. This is frame construction detail of 1952 Lincoln around the area of the front wheel suspension

Manufacturing Considerations

And so, as product design variations are being considered, there should take place, at the earliest possible stage, some agreements of design acceptance by representatives of the various manufacturing activities, based on their review of each product design in relation to cost, tools, labor, facilities, material handling, processes, and the many other necessary considerations affecting acceptance for manufacturing.

It is only logical that manufacturing considerations be incorporated in the design, because manufacturing problems parallel product engineering problems. Manufacturing know-how and experience should be integrated with the development of a product design, not following it, to assure that the product is designed for manufacture.

Analyzing Designs

This principle is logical and desirable, and far-sighted engineers want their product designs to be free of manufacturing problems. If discussions involving manufacturing problems are not brought up for resolution in the initial design stage, or close to it, they are certain to be brought up later on in the tooling or production phase, where engineering changes will be requested to ease the manufacturing problems encountered.

The usual way in which key manufacturing engineers analyze product designs for constructive evaluation is by looking at layouts, drafts, sketches, descriptions, and clay forms. Most of the manufacturing personnel reviewing these designs do not immediately visualize the construction, nor do they all see the problems in the same light.

It requires a considerable amount of time even for the most experienced manufacturing executives to analyze drafts and prints in sufficient detail to foresee the future problems at this point, especially enough to assure their elimination with any particular design. Most experienced personnel simply do not have all the hours necessary to devote their time to analyzing prints and drafts in this great detail. (See Fig. 1.)

Seeing Designs Quickly

Liaison between designers and manufacturing engineers is usually accompanied by stacks of prints, the engineers' language, with dimensioned details, over which each man must spend many hours before he will clearly see the design sufficiently to analyze each part for potential manufacturing problems. (See Fig. 2.)

Perspective illustrations have made it possible to

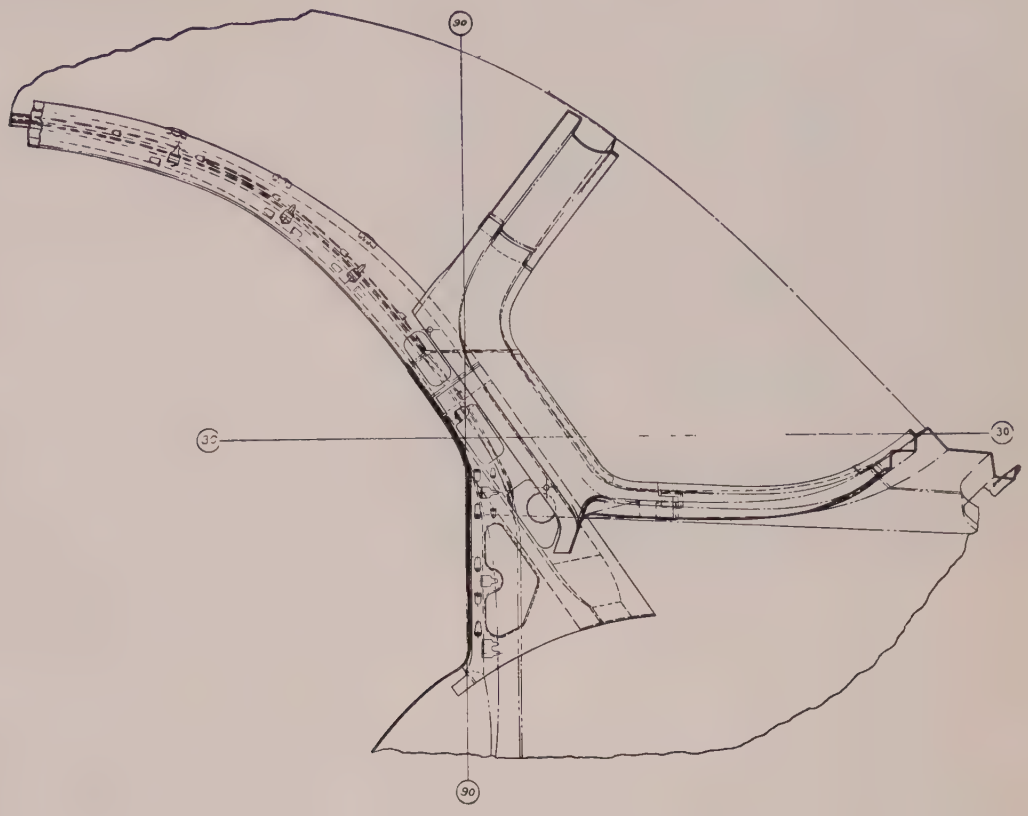


Fig. 2 – Typical body assembly drawing showing 10 parts of 1952 Ford. This area is around rear window opening and lock pillar

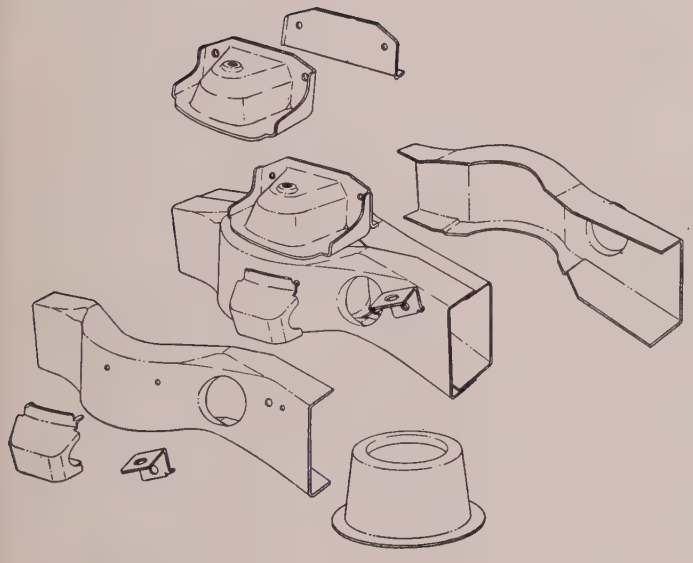


Fig. 3 – Perspective drawing of frame assembly shown before in Fig. 1. Notice how clearly each part can be seen

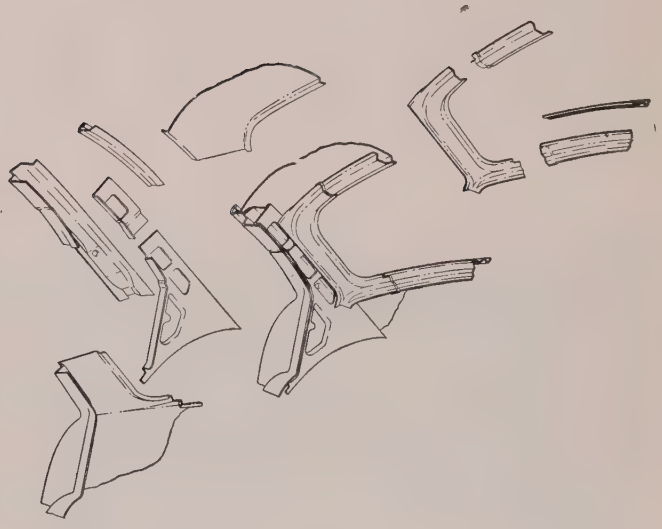


Fig. 4 – Perspective drawings of body assembly shown previously in Fig. 2. Notice how each part stands out

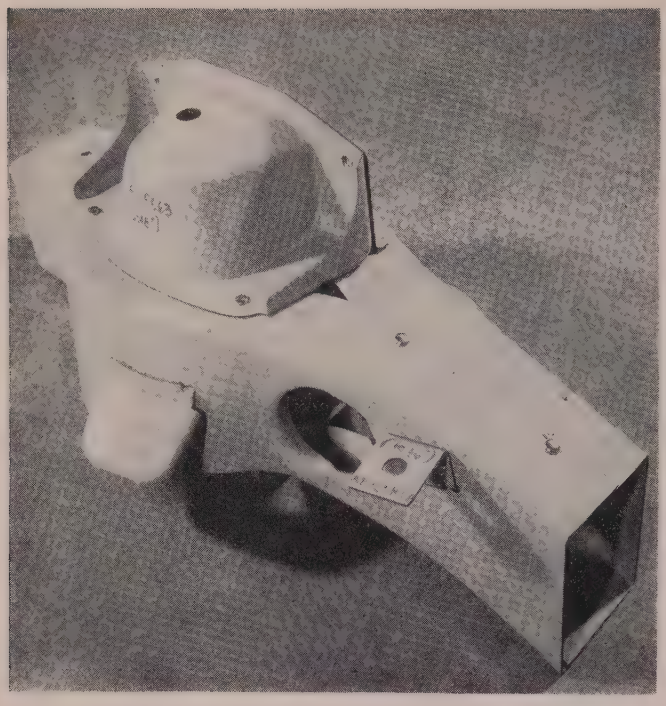


Fig. 5 – Plastic prototype study model of chassis frame construction of Fig. 1. All parts are made to exact gages specified. Note how clearly all problems are seen

discuss and review the design of the product with much more clearness than drafts and layouts. They have long been recognized as having tremendous advantages for discussions of proposed designs, processing entire manufacturing operations, and so forth. (See Figs. 3 and 4.)

Actual Models Clear Up Problems

But there is a further step that has tremendous advantages – the plastic scale prototype. Men can discuss the same product designs, but with real models of the actual parts, they can take them apart and reassemble them during their discussions. They are not only able to visualize the design instantly, but can immediately spot their specific foreseeable manufacturing problems involved. (See Figs. 5 and 6.)

Stamping experts can foretell the exact difficulties in the stampings involved. Welding men can state whether the design is good, or whether it will lend itself to accessibility of welding tools, or whether the design is good or poor practice for cost, strength, speed, and other essential considerations. Material-handling men can handle these actual parts and visualize the necessary shipping and handling equipment for them. (See Fig. 7.)

Saves Time and Expense

The advantage of having plastic models benefits experts from all other manufacturing phases, because they are able to visualize the tools and fixtures that will be required, estimate very accurately the cost of producing such parts, and see that the assembly of all parts have proper accessibility and desirable manufacturing aids such as location holes machining setup surfaces, and so forth. (See Fig. 8.)

To obtain parts in steel for studying manufacturing problems takes too much time and expense. For example, in obtaining a partial assembly model of a complicated jointure in steel on one of our future model vehicles, it took 3 months to get the steel parts handmade, and at a cost exceeding $4000. Even though the model repaid its cost in settling a difficult problem delaying decisions affecting both manufacturing and engineering, waiting for 3 months resulted in holding back other aspects of the design, and losing 3 months in engineering time.

Introduce Plastic Prototypes

It occurred to Ford engineers that since models of actual parts and assemblies were so effective, there was no sense in waiting 3 months to obtain a handmade model in steel when they could obtain the exact same models of formed sheet plastic parts within a couple of days. Not only would plastic models give them the same answers, but they could be made quickly, and for one-twenty-fifth the cost of metal parts. For example, the model assembly that was made of metal for more than $4000, was also made of plastic parts for $150, and in two days instead of the three months required for the steel parts.

D. J. Davis, director of manufacturing engineering and D. S. Harder, vice-president in charge of manufacturing, immediately foresaw the advantages of using plastic to make prototype parts quickly for manufacturing studies. They authorized the establishment of a department on a pilot operation specifically for the purpose of making proto-

Fig. 6 – Plastic prototype study models of partial body assembly of Fig. 2. Note how every detail is separate for individual study and accessibility for assembly. Roof is transparent so structure can be immediately analyzed

type parts of both cast and sheet plastics. (See Fig. 9.)

Their objectives were to have plastic study models made of questionable parts and assemblies, to assure that designs which were accepted for manufacturing could be made by the most modern methods, at the lowest possible costs, and that all parts, subassemblies and complete assemblies could be properly assembled and tooled. Problems involving stampings, forgings, castings, machined and other forms of parts, were to be evaluated and resolved in the design stage by these plastic models, to make certain that each design would be desirable for manufacturing processes.

Personnel Carefully Chosen

The personnel of this department was selected on an individual basis to obtain men with creative spirits, constructive shortcuts, and teamwork cooperation. They were thoroughly indoctrinated with the idea that models were always needed immediately and, therefore, perfect finish or appearance was not as critical as speed. They must turn work out in hours or days that would not ordinarily be done for weeks or months by conventional methods. Also, it was necessary to consider models as expendable, because after painstaking work in developing them, models would be discarded on a moment's notice for a better design. Requests to alter and rework models were not to be interpreted as an annoyance, but as additional proof that plastic models were serving their purpose in helping to improve designs for better manufacturing.

Early Costs Reduced by Millions

Plastic prototypes utilized early enough in the design and first stages of preparation for manufacture, make it possible to save millions of dollars annually by getting programs under way in less time, reducing engineering man-hours, reducing engineering changes, reducing tooling costs, increasing participation or contribution of manufacturing suggestions, and effecting more cooperative teamwork between manufacturing and engineering from the beginning of the design to the finished product. The first 54 problems which were developed into plastic models during the first few months of the departments existence used only 7 skilled craftsmen, and saved an estimated $500,000.

The use of models for development of both product designs and manufacturing processes far in advance where engineers can remedy undesirable conditions is not new, but heretofore obtaining models has been slow and expensive because they were made of metal.

Opens New Field

The use of plastic models, however, opens up a whole new field as a development medium to work out design problems harassing both engineering and manufacturing. Many people think plastic models are toys, or are good only for exhibits of one sort or another. The use of plastic for prototype models did not occur to many persons because the actual parts were going to be made of steel.

Fig. 7 – Close-up of rear window corner construction showing every part. Model can be taken apart to study sequence of assembly

Making plastic prototype parts after the form is once made cannot be compared to making parts of metal by the conventional casting, forging, hand-hammering, welding, cutting, patching, and working over hammer forms. Further, when parts are handmade of metal, each duplicate part requires nearly the same amount of time as the first part. However, in making plastic prototypes after a form

Fig. 8 – Two typical plastic study models of complicated jointure on experimental vehicle. Model to left is rear door opening consisting of rear quarter panel, lower back panel, and eight parts of floor construction of a truck body. Model to right is a front door opening at lower lock pillar showing quarter panel, lock pillar, rocker panel, floor pan, and five gussets

Fig. 9 – View of a corner of half of the plastic prototype model shop. Note plastic study models ready for delivery

is once made, any number of duplicates can be made readily in a matter of minutes and with little handwork except for trimming. (See Fig. 10.)

After the plastic has been prepared and placed in or over the mold, the parts can usually be formed in a single operation, whereas its counterpart of steel in dies might have to be made by many drawing operations followed by cam forming, restriking, or flanging. Back drafts and difficult drawing areas are of minor concern because the plastic technician can compensate for these difficulties either in making the mold or selecting the proper forming techniques. (See Fig. 10.)

Faster Than Drafting

Our department rapidly developed timesaving devices and shortcuts until they could make prototype parts in plastic from layouts faster than they could be drawn by a draftsman as details with dimensions!

Plastic prototype study parts are light and easy to handle, are very tough, resistant to severe bumps, shatterproof, waterproof, and stainproof, can be carried easily to meetings and conferences where they will not scratch tables or cut a person in handling. One man can carry a complete automobile side assembly with little effort. Since they can be quickly duplicated, copies can be sent for study to groups remote from the original engineering meetings, and copies can be used for showing the steps in the evolution of the design.

Resolution of Problems

Preliminary preparations for manufacturing starting with tool estimates, have been customarily made from plaster, clay, or wood models, special mockups and drafts usually prepared by styling. These estimates are usually based on the general exterior appearance and the association of previous cost history for the major parts, plus combined experience and know-how to determine the tooling differences between new and old designs. (See Figs. 11 and 12.)

It is only logical for manufacturing review to go further than just the outer appearance, and show the structural design of the major parts and subassemblies as they are made up into the vehicles,

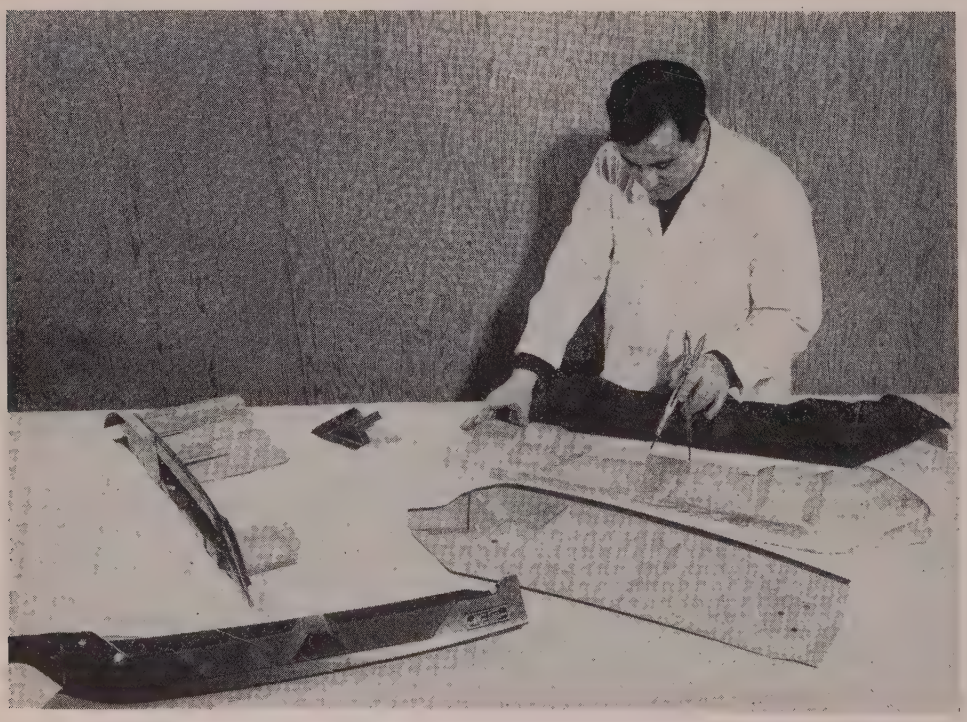


Fig. 10 – Proposed design of 3 pieces (on left) presented an extremely difficult complex stamping job. Manufacturing engineers made counterproposal of 1-piece design which was a simple press operation. Form and parts can be made and delivered in about 7 hrs

because complete bodies (in white) can be made quickly in plastic prototypes. Then much more accurate cost analysis can be assured. By fashioning plastic models directly from layouts, manufacturing problems can be considered parallel with engineering product design, instead of after the models, tools, dies, and fixtures have been completed, or even parts made.

Rapid Contribution of Ideas

Wise old Confucius once said: "One picture is worth a thousand words." Manufacturing engineers say "one plastic model is worth 40 lb of blueprints and 40 hr of explaining that goes with the prints." Whenever plastic study models are used in discussions, floods of new ideas are contributed by both engineering and manufacturing personnel, who would not ordinarily have contributed these ideas until they had seen metal parts later on in toolng or production. (See Fig. 13.)

Plastic prototype study models, as a means of joint development between manufacturing and engineering, insure incorporation into product design of the best processes in:

1. Designing.
2. Machining.
3. Casting.
4. Coring.
5. Material handling.
6. Welding.
7. Forging.
8. Tooling.
9. Repairing.
10. Checking.
11. Assembly.
12. Metal and other finishing.
13. Purchasing.
14. Labor standards.
15. Cost comparisons.
16. And many others.

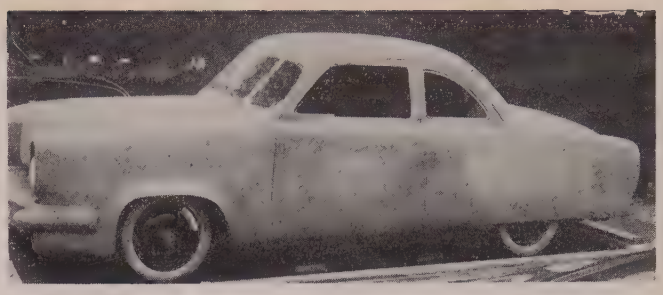


Fig. 11 – Typical clay and wood styling model with only outer appearance complete is base for determining preliminary tool estimates and manufacturing operations

In the early design stages it is also important to determine which designs to eliminate that would present unacceptable manufacturing problems. It would be folly to design a product which would have an excessive cost or could not be manufactured with either present or obtainable facilities and equipment, or even with vendors' facilities and equipment.

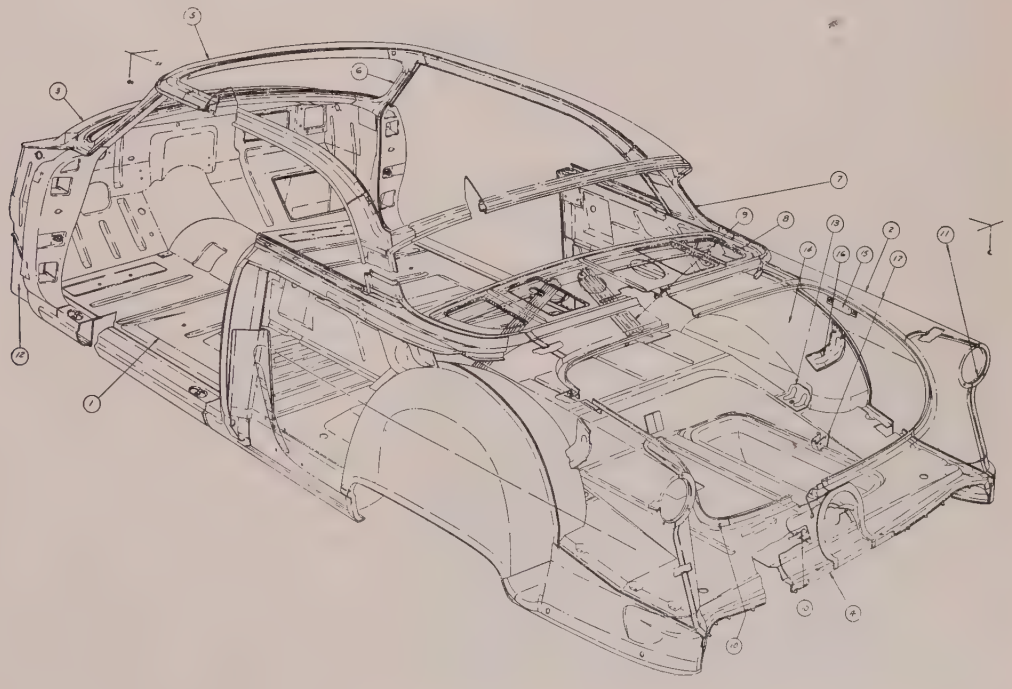


Fig. 12 – Prototype development model showing all major structural parts is also used for determining preliminary tool estimates and manufacturing operations

Fig. 13 – Body development engineers, weight analyst, and manufacturing engineer discussing plastic prototype study model for construction details

Fig. 14 – Plastic study model of difficult area on an experimental commercial vehicle which is not acceptable to manufacturing because of difficulty of assembly, complex stampings, and high shipping costs. The number 6, just below modelmaker's hand, indicates that this was the sixth proposal

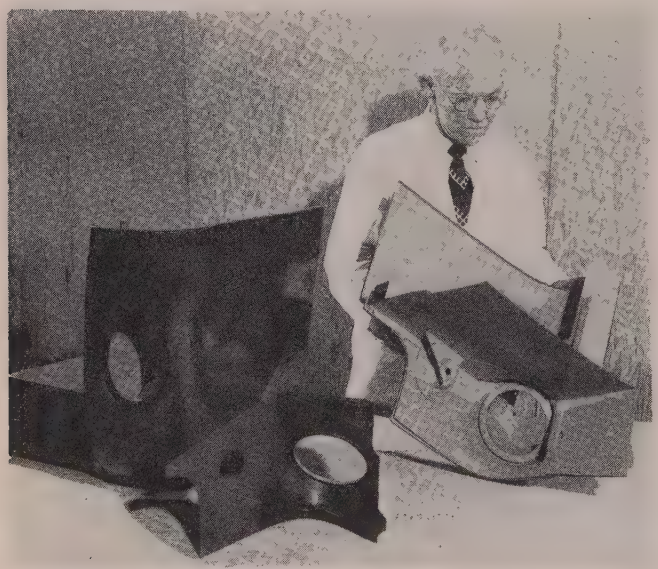


Fig. 15 – Plastic model of same area as Fig. 14, except that this proposal is completely acceptable to manufacturing

Manufacturing Changes Anticipated

Proceeding Without Manufacturing Concurrence – Far too often, engineering must proceed with designs and even detailing without receiving constructive criticism from manufacturing personnel in the early stages. Then, after detailed parts and assemblies are released to manufacturing, they will be studied for consideration of stamping, assembly, material handling, welding, and other manufacturing operations.

Manufacturing always requests a considerable number of changes, or at least to compromise the design, to allow for some of the manufacturing problems. Unfortunately, changes requested at this late date cannot be easily granted because the general design has progressed too far to accept any major revision without upsetting the entire program timing schedule.

Specific Decisions Quickly Reached

Counterproposals Made Quickly with Plastic – Manufacturing problems can be discussed so quickly and clearly by using plastic study models, that specific dependable decisions are quickly reached which ordinarily might have required up to several months. With plastic prototypes to discuss, engineering and manufacturing personnel are both immediately enlightened with the problems of each other so they can iron out designs satisfactorily to both interests. (See Fig. 14.)

Counterproposals from manufacturing personnel are often necessary to alter the design to conform to certain manufacturing practices, to take advantage of the tools on hand, and for other reasons. It is desirable that many designs and counterproposals in plastic be discussed. As many as 15 design proposals for a particularly nasty problem can be narrowed by elimination to 3 or 4 of the most advantageous designs, and by further elimination to the best of 15 product designs. (See Fig. 15.)

Accomplishments in One Day

An example of the strategic advantages of reworking plastic for counterproposals is demonstrated by an incident where a plastic study assembly of 10 parts was being used in a discussion. At 11 o'clock several beneficial changes were suggested, so the 10 plastic parts were returned to the plastic model shop, where they were promptly reworked, reassembled and resubmitted by 3:30 of the same day. Within 10 min after examining the reworked plastic models, the committees accepted the proposal. If this were to have been done in steel, it might have required days (if not weeks) of diligent handwork to have brought about similar alterations.

Plastic Prototypes Help to Guide Design – Since plastic prototype models can be made much more quickly than detailed parts can be drawn or dimen-

sioned, it is a logical step for keen product designers to take advantage of plastic study models as a medium of development from rough layouts and drafts without detailing. Parts can be fashioned so quickly (and changed so easily) than an evolution into 3-dimension product designing can be readily achieved to replace the flat, laborious thinking-on-paper currently required. (See Fig. 16.)

This, of course, requires a creative personnel group comprised of actual development engineers who work as plastic technicians. By fashioning plastic parts, instead of making detailed prints, more immediate solutions are found to design problems, and constructive criticism can be obtained from manufacturing engineers with these models.

Parts Can Be Stresscoated

Plastic parts can be stresscoated and subjected to torsional and beam loads to detect stress patterns from deflections. Stresscoated plastic parts have displayed identical stress characteristics with steel, provided they are limited to the same degree of deflection that would be encountered with steel parts. Weak areas and fracture points can be detected early enough to guide the engineer in altering his design to obtain maximum stress values.

Further, after utilizing plastic development models to achieve the accepted advance design, these accepted plastic models can be turned over to detailers to complete the engineering drawings. This reduces drafting time considerably, because each part is completely and clearly visualized. It has, of course, not been practical to do this with large clay models of accepted designs. (See Fig. 17.)

Plastic has been used for some time in a limited scale in styling work to represent outer parts of a car for appearance. Often such parts have been plated or painted so cleverly that they were mistaken for the real parts. Styling uses plastic parts for not only original designs, but also for "face lifts" and other changes in cars, trucks, tractors, buses, and other vehicles. (See Fig. 18.)

Models Sent to All Plants

Plastic Models Easily Duplicated and Sent to Many Places Simultaneously – Once the design is determined, and manufacturing engineering (processing and tooling) can begin, plastic prototype duplicates can be easily made and sent to specific activities which process operations, design tools and machines, or prepare facilities. Thousands of hours of tool and fixture design and construction time can be saved by using these plastic duplicates to quote and work from. Tooling can be done with little fear of major design changes which by this time should be negligible, because those expected or usual changes always requested later on in the tooling program will already have been worked out to the satisfaction of everyone.

Plastic parts and assemblies of complete bodies

Fig. 16 – Modelmaker checking ¼-scale model of partial jointure of hinge pillar, instrument panel, cowl side, cowl top panel, dash, and other parts

Fig. 17 – Modelmaker trimming scale plastic models of metal stampings. Note fender study models at left made off of ¼-scale die models

Fig. 18 – A styling mockup of 1952 passenger car showing plastic parts simulating real parts

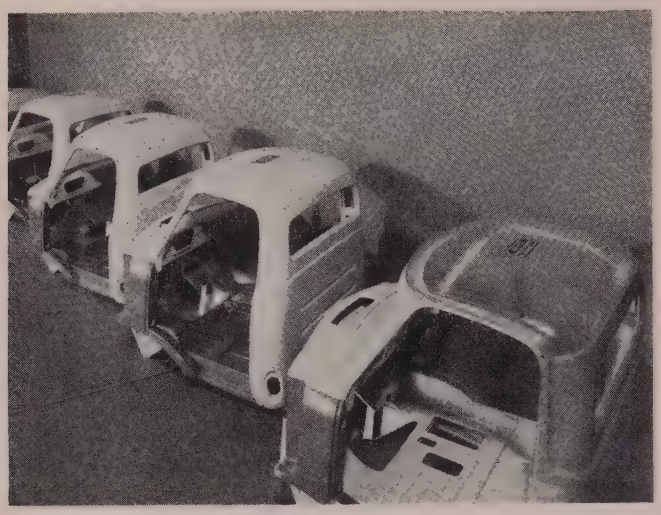


Fig. 19 – Duplicates of plastic prototypes of completely assembled experimental truck cabs ready for delivery to different groups in engineering and manufacturing, where they will be studied for various problems of manufacturing, assembly, shipping, and so forth

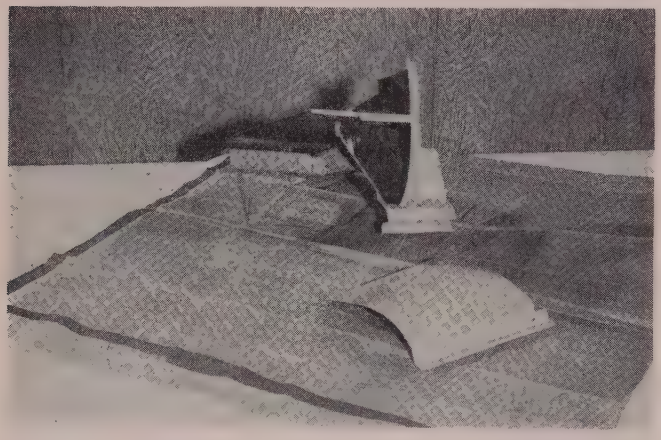


Fig. 20 – Partly assembled plastic scale model of a complicated jointure furnished to tool design for preparation for manufacture. Note loose panel in foreground which has been detached from assembly. Also note tremendous difference between seeing the actual jointure and the assembly print on which plastic model is resting

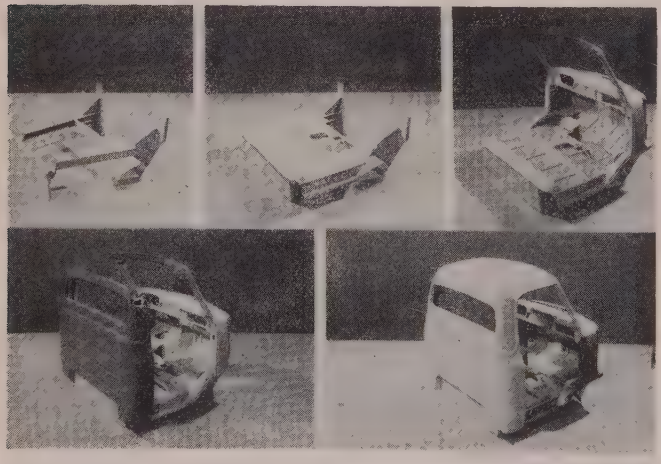


Fig. 21 – Plastic prototype study models used to experiment with different sequences of assembly on an experimental truck cab. Many combinations of sequences of assembly can be determined by plastic subassembly

may be made for shipment to plants in other parts of the country for advance planning and preparation for their manufacturing. Parts would not ordinarily be sent to these areas for at least 9 months or a year later, after they became available in steel from production dies. Ford sends plastic models of not only troublesome jointures but complete bodies to one of its manufacturing and assembly plants in South America, where the stampings and tools will be made locally. The value of having plastic prototype parts for that country, already short of skilled tool designers, cannot even be calculated in savings of time, errors, and manpower in preparing for manufacture at that plant. (See Fig. 19.)

Schedule Advanced One Year

Plastic Models Effective Aid for Processing – For manufacturing engineers to have prototypes of actual parts with which to work 9 months to a year in advance of actual production parts is, to say the least, the most beneficial new aid they have had in years. They always hoped to have parts to accompany their design prints through some of the most critical manufacturing phases: plant planning, sourcing, purchasing, and the very comprehensive tooling up. (See Fig. 20.)

The value of using plastic prototype parts for processing is almost incalculable, because all those items usually questionable from prints are as clear as crystal in plastic. A processing engineer can visualize all the operations and their sequences far more quickly and comprehensively. More imaginative and clever manufacturing operations result from having actual parts to work from. (See Fig. 21.)

Wide Interchangeability of Parts

In many instances plastic models showed how several operations could be combined with others or dropped altogether. In other instances they showed the way to more standardization, either with other parts or with machines and tools available from other parts production. Much wider interchangeability of parts was found possible by taking plastic models apart for exchange on other assemblies. (See Fig. 22.)

One of the most important benefits from plastic models was that they showed the way to balancing the operations in time cycles and manpower requirements. An example of this was that full-size plastic sections showed the exact requirements for welding gun accessibility; which guns should be used; what location and spot pattern should be followed; which guns could be used on these and other parts in different areas on the bodies; how much time would be required to do each operation; how the welding guns would have to be tilted and balanced; and what standardization could be achieved on gun styles and electrodes. All the advance planning and processing was safely done a year in ad-

vance without worry about slipping up on some overlooked detail.

How Plastic Parts Help Tooling

Many tool engineers have joked that they would have "given their right arm" to have had prototype parts to use in the tooling and processing phases to avoid errors and wasted time in planning for manufacture. Manufacturing engineers can do a much more efficient, trouble-free tooling job if they have advance parts, as can be seen in the vast number of engineering changes they request *after* they receive the real parts to use their tooling on.

Finding out troubles in parts at the tooling or preproduction stage is too late to effect many changes to improve costs, tools, methods, and machinery. Thousands of design-hours have already been expended; die models have been made; machine tools with long delivery have been ordered; facilities and procurring arrangements have been settled; and hosts of other tooling commitments have been made, all of which would make it costly to put through changes this late to help manufacturing.

In the tooling-up phase, plastic prototypes serve not only as substitutes for the real parts to familiarize the tooling people with every aspect of the product design, but they actually make it possible to shorten tremendously the amount of preliminary work required. They give a green light to proceeding at least with accomplishing the roughing-out of tools.

Exact Draw Tip Established

For example, in designing and building dies, having plastic parts will permit the die expert to physically tip the parts to establish the exact draw tip for balancing out the draw of the metal. He can see where to place the binder rings and how many die operations will be required. The die designer has already tipped the plastic part in final position so he can proceed with his layout very quickly and confidently. He can see where to place inserts, where to establish the main casting limits, and can release his patterns out for casting and rough machining long before he has his final die surfaces detailed. (See Fig. 23.)

Where needed, his plastic parts can be formed or reworked to represent the parts in any progressive stage for the corresponding dies. The blank design, cam operations, piercing, restriking, rolling, and flanging, or any of the scores of other die operations can be worked out in plastic. In addition he can also check the plastic models for thinning out of metal for any particular area of deep draw or stretch.

Parts of reinforced plastic can, where required, be made so perfect and accurate that they can be used for designing and constructing checking fixtures to them. For this use, however, it is customary for bracing to be used on such parts while they were still on (or in) the master forms. Checking fixtures

Fig. 22 – Modelmaker assembling experimental cabs. Note sub assemblies as well as complete assemblies for study by production engineers

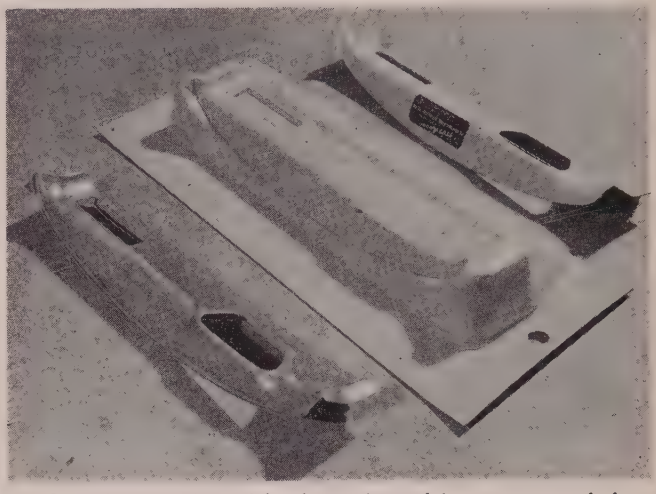


Fig. 23 – Plastic study models of experimental instrument panel showing two stages of plastic forming. In center is the completely drawn instrument panel on gridded sheet plastic which indicates exact flow of material. At right is a plastic instrument panel trimmed; at left is a metallized plastic instrument panel trimmed

Fig. 24 – Plastic scale models of 1952 Ford passenger-car frames made from 18 hardwood dies shown at left. These models are used to determine manufacturing operations as well as material handling. Complete set of hardwood dies requires 40 hr to make; plastic duplicates can be formed in minutes, and trimmed and assembled in 6 hr

Fig. 25 – Plastic prototype shipping assemblies – ⅛ scale – made for material-handling executives to place in a scale railroad car to determine shipping space accommodations, arrangement, equipment required, and so forth. Shown in foreground are hood assembly, center and rear floor pan, lower back panel, complete side assembly, and frame

themselves can be made of special plastics with glass-smooth surfaces, true dimensional stability, and the toughness of heat-treated steels. (See Fig. 24.)

Dies and forms of plastic are also made to produce low-volume stampings of steel and aluminum, and master die models are made of thermosetting plastics which are far more dimensionally accurate than mahoganies, over a long period. Further, they are not only cheaper, more quickly made, and lighter to use than master mahoganies, but duplicates are very easily made to serve several tool activities simultaneously.

Fig. 26 – Typical plastic duplicate of forging for manufacturing study. Modelmaker is suspending model in mold for duplication in plastic for study in connection with various manufacturing activities. Solid plastic models are easily made in above manner for forgings and castings

Ideal for Jigs and Fixtures

Plastic prototype parts and assemblies are a natural means of helping the jig and fixture men, because each part can be dismantled from the assembly for study of positioning, sequence of assembly, locating, clamping, operating time, checking and all the other operations and motions needed to complete the parts and assemblies in the jigs or fixtures. (See Fig. 25.)

Material handling engineers have been able to prepare their special parts handling equipment months ahead of time by using plastic parts. They can position their assemblies, check parts for palletization, determine railroad car dunnage, special racks, boxing or crating needs, and develop hosts of other preparatory material-handling devices.

Similarly, machining experts obtain tremendous benefits from having prototype plastic parts as a substitute for real castings and forgings, such as wheel spindles, suspensions, motor blocks and heads, bearings, transmissions, axles, crankshafts, housings, and scores of other machined parts. They can see the parts in all their aspects to enable them to visualize every single machining operation, the positioning of the parts, inspections, various stations, types of machines required, degree of interchangeability with other machining operations, cycling of machines, and hosts of other considerations. (See Fig. 26.)

Different Types of Plastics Used

Plastic is available in many forms for making prototypes. The most easily used form is in sheets, in thicknesses of from less than 0.010 in. to over ¼ in. These sheets are available in transparencies and in colors from a thin, milk-white to jet-black, through pastels and shades of colors (blue, red, green, ivory, brown, and so forth) which are quite advantageously used for contrast between the different parts, and to follow a color pattern to indicate interchangeability and other points. The sheet plastics are readily formed over (and into) very low-cost molds of the individual parts. Plastics can be molded, sawed, sheared, drilled, machined, sanded, sewed, cemented, polished, patched, reworked, metal-coated, painted, plated, and treated in many additional ways in accordance with each requirement.

Plastic parts and forms are also made from the liquid state, to which catalysts are added to solidify them into the degree of hardness wanted. Polyesters and phenolic resins are reinforced with Fiberglas to make parts which are even stronger than

steel, are dimensionally stable, and so accurate that they can be used in building checking fixtures.

Liquid Plastics

Liquid plastics with catalysts and accelerators are used to a very great advantage in making master molds and dies over which not only plastic parts, but also steel, aluminum and other metal parts can be made. (See Fig. 27.)

These plastic dies and forms are not only extremely accurate but are tough and stable for a period of years. Liquid plastics are also used in casting and molding parts to represent prototypes of castings, forgings, die castings, extrusions, and several other forms of materials.

There have been few limits on which prototype parts can be made of plastic, because they can be cast, machined, cored, formed, extruded, molded, and fabricated from any combinations of these methods. They can be reinforced; cast or formed with metal inserts at specific points; threaded or tapped; pigmented to any color combinations; bolted, cemented or riveted, braced or contained by steel or other materials; and treated to other requirements.

Flexible Plastics

It was found very time-saving to even make seals, moldings, insulations and other rubbery or flexible shaped parts for tryouts on real car doors and windows. The type of plastics used for this purpose are the vinyl plastisols and plastigels, available in liquid or paste form, which are fused into the degree of rubber-like toughness desired. Ordinarily it requires five weeks or more to obtain molds and regular cured rubber parts, but with plastic prototyping it can be done in only a few hours. Further, if there are any alterations to be made (and there usually are), it formerly required another 5 or 6 weeks until the engineers can obtain new molds before they test the revised moldings out for water and air leaks.

Other Manufacturing Activities Aided Indirectly – Manufacturing illustrations of parts and assemblies in perspective are usually made on a master perspective grid, which takes much plotting time. To save thousands of illustrating man-hours, finished plastic models can be photographed in car position (or other views more advantageous) and printed directly on cloth or vellum with additional data added, before making prints for distribution.

The purchasing department can request specific plastic models to achieve more bonafide quotations and commitments than can be obtained from the regular blueprints alone. Many hidden costs are added to the price of tools and machines to allow for oversights or intangibles which always occur when people deal strictly from prints. Further, negotiations can be made more confidently when both

Fig. 27 – Various steps showing how to make experimental hood parts. Clay modeler is making clay form; at his left is rubber mold; in the foreground, from left to right, are plastic form and reproduction plastic prototype scale parts

Fig. 28 – Typical cast plastic duplicates of standardized tools. Parts shown are for a standard portable welding gun with interchangeable yokes, adaptors, extensions, cylinders, handles, cable terminals, and welding tips

Fig. 29 – Final trimming and finishing touches on plastic scale model of typical experimental truck cab

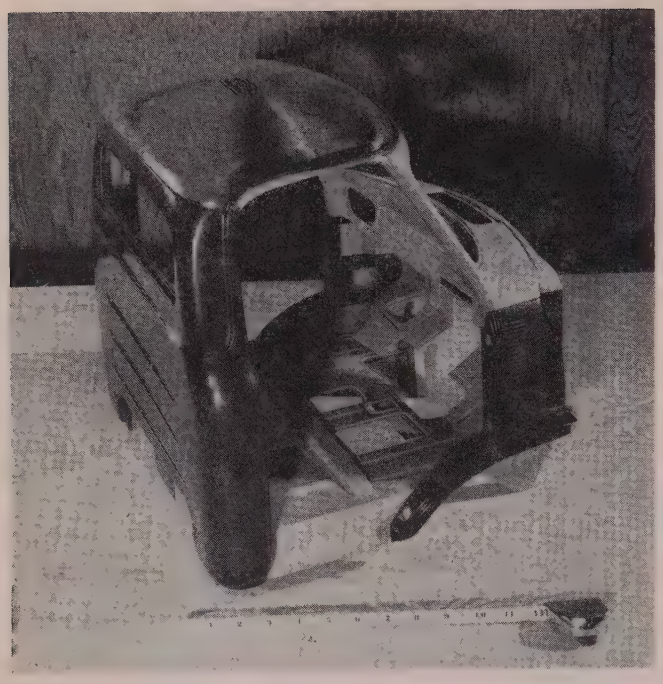


Fig. 30 – Plastic ¼-scale model of experimental Ford truck cab. Transparent roof is for purpose of showing construction details of model. Ruler in foreground shows model is only 1 ft long

Fig. 31 – Full-size plastic model of jointure of 1952 Ford passenger car showing rear quarter panel, upper back panel, rear backlight frame, drip molding, inner roof rail, outer roof rail, and roof rail to wheelhouse gusset. Roof is made of transparent plastic to give visibility to structure for accessibility at assembly

buyer and vendor are talking about the same thing, and both see the problems from using plastic parts.

The design of special machines is always helped greatly if there are actual-size parts available around which the machine can be constructed. Plastic prototype parts fill this requirement magnificently as much as a year in advance. In making working models of special machines or manufacturing processes themselves, plastic parts have paid off handsomely. As a few examples, we used them to develop a special hopper and automation unit for bearing caps which vendors said could not be done; it did not work with either the first or second model, but the third worked like a charm, and was loaded on a truck and delivered to a machine tool vendor for real construction. Similarly, we made special fixtures for separating sprues from cast valve clusters, and models of standardization for conveyor drives, and special jet-blade contour polishers, plus scores of other development fixtures or partial machines.

For working models of larger quantities of standardized tools we made cast plastic reproductions. For example, portable welding guns having interchangeable jaws, yokes, adaptors, extensions, cylinders, handles, cable terminals, welding tips, and so forth. (See Fig. 28.)

Full-Size or Scale?

Both full-size and scale prototypes are required. With ⅜-scale prototypes, greater speed can be attained in making the forms, because they require less material, less labor and the entire assemblies are easier to handle for over-all development work. (See Fig. 29.)

Three-eighth scale assemblies show up the same problems in design as full-size, and every part and assembly can be worked out so it will clearly serve its function and go together as a unit of an assembly. (See Fig. 30.)

Full-size models show true and actual conditions for studies of processing and tooling, accessibility, stamping problems, and other factors which are sometimes misleading on a smaller scale. In addition, full-scale plastic prototypes of bodies and front end sheet metal parts can be put on a chassis and actually driven. (See Fig. 31.)

Plastics to Aid both Engineering and Manufacing Barely Started – Modern industry is constantly searching for better materials, better tools, better facilities, and better ways of doing things. We are certain that as better ways of doing things are found, we will adopt them. The widening use of plastics in aiding our engineers to plan and prepare for manufacture is one of them. We are all limited only by our imaginations and hard work. In our honest opinion, this has been like finding a gold mine—having inexhaustable wealth in it ready to be utilized, but which reaches its greatest value only when it is worked.

ENGINE-TRANSMISSION RELATIONSHIP FOR HIGHER EFFICIENCY

D. F. Caris and R. A. Richardson,
Research Laboratories Division, GMC

This paper was presented at the SAE Summer Meeting, Atlantic City, June 6, 1952.

ENGINEERS have been concerned about engine-transmission relationships since the earliest days of the automobile industry, W. S. James[1] as well as several others have presented their interpretation of the requirements of an "ideal" transmission before this society in the past.

Even before the days of the automobile, men were familiar with a few of the general problems (Fig. 1).

Considerable engineering progress has been made since then.

[1] See *SAE Journal*, Vol. 54, August, 1946, pp. 50-52: "An Ideal Transmission," by W. S. James.

WHETHER to design an ideal transmission to overcome the deficiencies of the internal-combustion engine, or to reach a similar objective through engine improvement, is discussed in this paper.

The authors present a picture of the historical development of engine-transmission relationships from the earliest days of the industry down to current trends. They emphasize the increasing importance of fuel economy, and point the way toward an operational goal of 30 mpg of gasoline for the average car.

Other suggestions are offered for the reduction of engine friction which would eliminate high power losses, the reduction of wind resistance through the development of aerodynamic design, and the possibilities of the variable displacement engine eventuating in an ideal transmission which would be an extension of the crankshaft to the differential.

The Authors

D. F. CARIS (M '45) is head of Mechanical Engineering Department Number 4, Research Laboratories Division, General Motors Corp. He received his electrical engineering degree from the University of Michigan, and joined GMC in 1926. Mr. Caris has been a co-author of several papers on the subjects of high-compression engines, electrical engine indicators, and other related subjects.

R. A. RICHARDSON (M '31) is head of the Technical Data Department, Research Laboratories Division, General Motors Corp. He is a mechanical engineering graduate of the University of Minnesota. Mr. Richardson's department provides engineering service and correlates work of the laboratories with engineering departments of the General Motors manufacturing divisions and central staff operations.

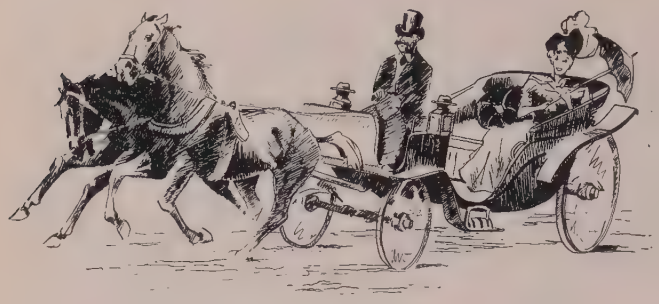


Fig. 1 – Even before the automobile, men were familiar with some engineering problems

As the present internal combustion engine-powered automobile was evolving in the 1890's, there was a continual effort to develop a transmission which would give high torque for starting, accelerating, and hill climbing and, in general, allow more flexibility in using the power from the engine. Proponents of the steam engine and electric drive pointed out the inherent advantages of these means of propulsion, since they had built-in transmissions with easy driver control. But even though the power-speed relationship of the gasoline engine still remains a problem, this powerplant proved to be lighter, more reliable, low in cost, and used a widely available fuel. Cold starting and winter operation problems were satisfactorily solved. For a mass production vehicle, the gasoline engine proved to be the best all-around powerplant in competition with other types of engines.

The gasoline engine does have some inherent characteristics, or should we say limitations, which have to be taken care of in the transmission. Automotive engineers have worked on this problem since the beginning of the industry when belt or chain drives were the accepted means of power transmission.

An interesting history of the search for better transmissions is revealed by a review of the mechanisms studied by our Research Laboratories in the past 30 years. A report written in 1925 listed nine types of transmissions which had been studied up to that time:

1. Sliding gear transmissions.
2. Overgear transmissions.
3. Synchronized transmissions.
4. Constant mesh transmissions.
5. Planetary transmissions.
6. Variable throw transmissions.
7. Inertia transmissions.
8. Hydraulic transmissions.
9. Electric transmissions.

Models of most of these types of transmissions had been built and tested either on the dynamometer or in a car. Fig. 2 shows an early hydraulic transmission designed by Manley, the engineer who worked with Langley on his first airplane. This was a piston pump and motor type, and was probably one of the first hydraulic transmissions on which serious work was done. It aimed at producing an infinitely variable drive.

Fig. 3 shows an 8-step gear transmission which is typical of many such devices proposed in the early days of the industry. Again, designers were groping toward an infinitely variable ratio change. This model was built in the early 1920's.

Fig. 4 shows an inertia-type transmission which might be called a mechanical torque converter. This was the Constantinesco transmission model, also built and tested in the early 1920's as another attempt to obtain infinite ratio change.

One of the early attempts to provide an infinitely variable transmission was the use of a friction drive (Fig. 5). As a matter of fact, the organizers of GMC purchased the Cartercar Co. soon after the organization of the corporation in 1908. One of the important reasons for this purchase was that the Cartercar used one of the first friction transmissions, and had been successful in producing them. These early pioneers recognized the importance of infinitely variable transmissions and thought that friction drives would soon be universally used.

A Cartercar catalog of 1906 said, "We manufacture only friction-driven automobiles, under Carter's patents, because after three years' severe service under every possible condition, it has proved the simplest, most powerful, and least expensive to maintain."

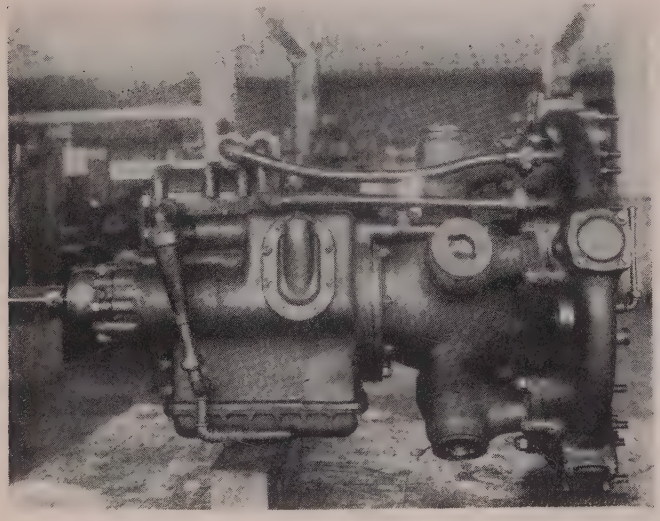


Fig. 2 – Early hydraulic transmission designed by Manley

The transmission consisted of a disc and sliding friction wheel. The catalog further said, "The friction wheel is slidably arranged on its shaft and may be moved across the face of the disc, its position determining the speed, which may be from absolute rest when in mid-position, full speed, or an intermediate speed at the will of the operator. Reverse motion causing the car to back, is effected by moving the friction wheel past the center of the disc. Absolute control is thus secured, both forward and back, by one lever. The friction thrown to the reverse also acts as the most convenient brake."

The many transmissions proposed in the past show that the idea of an infinitely variable type has been an attractive goal since the beginning of the industry. The Research Laboratories Division has learned much from an intensive project on a double toric friction transmission started in 1928 and continued through the early 1930's (Figs. 6, 7, and 8). Over 30 of these units were built and tested; 17 were installed in passenger cars, and road tested for over 300,000 miles. An additional 115,000 miles was run on dynamometer durability tests. The total was over 400,000 miles of testing.

This infinitely variable friction transmission was fully automatic and weighed about 250 lb. It had a maximum efficiency range from 87 to 91% at designed torque. It had a range of drive ratios of 8/1. Overall gains of 20% in fuel mileage were obtained on cross-country trips.

It failed to be adopted for a number of reasons, both economic and engineering. It came up for consideration during the depression of the early 1930's and before the actual incentive for automatic transmissions had begun to be felt. The difficulty of getting such a new and complicated mechanism into high production can be readily appreciated. For these and many other reasons, this transmission was not put into production, but it had a very important effect on the evolution of the automatic transmission of today.

The authors believe that it was the first attempt to design a fully automatic transmission after a thorough analysis had been made of engine characteristics. In early 1928, before a mechanism was chosen, computations based on dynamometer test data from a standard automobile engine were made to determine the performance and economy gains to be expected with an ideal infinitely variable transmission. Assumptions were that the engine would always be operated at the speed and load of minimum specific fuel consumption, and never above the maximum power point.

It was recognized that the best performance and economy of a car equipped with an infinitely variable transmission required an automatic method of control. Accordingly, one of the first automatic control systems based on an analysis of engine-transmission relationships was developed. The thorough and comprehensive development program which was started by the fundamental study of transmission requirements became the basis for much of the later transmission development.

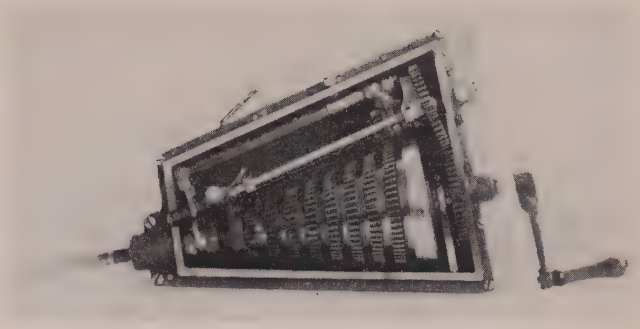


Fig. 3 – Eight-step gear transmission

Designers of present types of automatic transmissions had this wealth of background when they developed the Hydra-Matic and torque converter. Similar background information was no doubt available throughout the automobile industry. This is not to detract from the very excellent work which was done in producing these first production automatic transmissions, but only to show that the problem has existed for a long time. And just as there is much that remains to be done with engines, much can be done to correlate the engine-transmission characteristics.

What was learned from these extensive engine-transmission studies? Many people at the Research Laboratories learned for the first time what it felt like to drive a fully automatic transmission. It gave a baseline in performance, control, and economy which could only be obtained in an actual car. Fur-

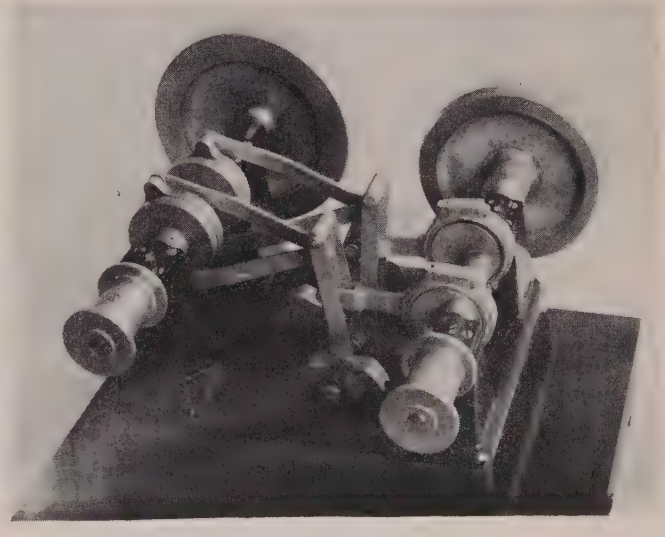


Fig. 4 – Mechanical torque converter

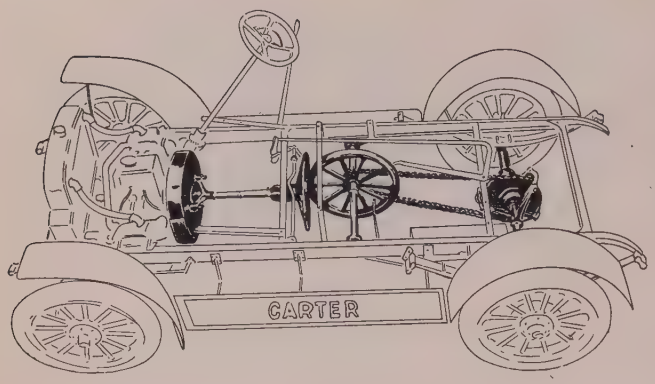


Fig. 5 – Cartercar used one of first friction transmissions

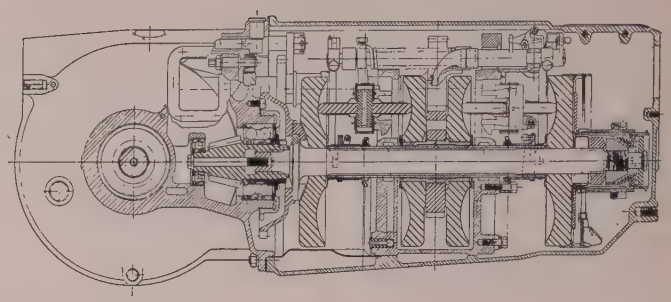


Fig. 6 – Section of infinitely variable friction transmission

thermore, the work pointed out five important contributions of an ideal transmission if such a satisfactory commercial device could be developed. These requirements of an ideal transmission are listed as follows:

1. Eliminate the clutch pedal.
2. Eliminate hand gearshifting.
3. Give added safety by making driving easier.
4. Automatically provide the ideal ratio for every driving requirement.
5. Take full advantage of the inherent economy of the engine while providing required performance in acceleration and hill-climbing ability.

Several present transmissions obtain a full measure of the first three points. Attempts to obtain full advantage of the last two in a practical production transmission have not yet been fully successful.

Since obtaining maximum economy depends as much on knowledge of engine characteristics as on transmission mechanisms, a review of the state of knowledge in engine development will be helpful in establishing desirable transmission characteristics.

The past 15 years, and particularly since 1946, has been a period of rapid development in both transmissions and spark-ignition engines. The Hydra-Matic type and then the torque converter were developed, followed by many combinations of the principles involved in both types. Higher compression engines have been developed, with still more to come.

Since this paper will present transmission requirements from the engine designer's viewpoint, a rapid review of the present status of engine development will be presented before discussing engine-transmission relationships. In the following discussion it will be noted that the Research engines are tested in standard General Motors cars. The results of this work reflect the engineering thinking of the Research Laboratories Division, which is independent of the production divisions.

High-Compression Engine Development

It has been shown before this society by C. F. Kettering[2] in 1947, and by C. L. McCuen[3] in 1951, that higher compression ratios are the key to higher efficiency. They have shown by a series of experiments on both single and multicylinder engines, that large gains in economy are possible with compression ratios of 12/1 or more.

There are, of course, many other factors besides compression ratio which can contribute to the overall efficiency of the automotive engine, such as mechanical efficiency, volumetric efficiency, carburetion, spark advance, and many others. It is assumed, however, that in modern engines these factors have been fairly well exploited, and therefore the key to further improvement in engine efficiency lies primarily in higher compression ratios.

There are two factors which permit compression ratios to be increased:

1. Chemical octane numbers (fuel quality).
2. Mechanical octane numbers (engine design).

Chemical octane numbers are put into the fuel at the refinery. Petroleum technologists have learned over the years how to rearrange the molecular structure of gasoline and improve its antiknock quality. The octane quality of gasoline has improved on the average about one octane number per year over the last 25 to 30 years. That this record of accomplishment has been achieved at no increase in cost to the customer is a tribute to the progressive efficiency of the petroleum industry.

How effectively the automotive industry has used

[2] See SAE *Quarterly* Transactions, Vol. 1, October, 1947, pp. 669-679: "More Efficient Utilization of Fuels," by C. F. Kettering.

[3] See SAE *Quarterly* Transactions, Vol. 6, April, 1952, pp. 290-303: "Economic Relationship of Engine-Fuel Research," by C. L. McCuen.

this improvement in fuel quality over the years is illustrated by the results of a group of Cadillac cars covering the period as far back as 1915. While many of these data were first presented by C. L. McCuen[3] before the API and Canadian SAE in 1951, the authors feel that the results bear repeating, since they sum up the gains made through engine development.

Fig. 9 shows these cars, equipped with modern test instruments, lined up on a desert road near the General Motors Phoenix proving ground. The car in front is a 4.25 compression ratio 1915 model, followed by a 6.25 compression ratio 1935, then a 7.5 compression ratio 1951 and, at the end of the line, the 12/1 compression ratio 19XX Cadillac. General Motors development work is aimed at this high ratio, and engines are expected to be ready when fuels of high enough octane quality are available. Since this is in the future, the 12/1 model was named "the 19XX." The 19XX also has a special transmission allowing a low rear axle ratio to be used. This gives a low ratio of engine rpm to vehicle speed (N/V), which results in high mpg values.

In comparing a group of cars such as this, it is necessary to evaluate two factors – performance and economy. Performance means the ability of a car to accelerate rapidly, to have good hill-climbing characteristics and to easily attain the maximum speed. General Motors, over the years, has emphasized balanced performance, never losing sight of the basic importance of economy in achieving the most efficient utilization of fuels. Engine efficiency may be defined as the ability to convert a gallon of fuel into the maximum miles of transportation.

First to be compared are the performance standards set for these various Cadillac cars over the years in terms of the relative distances traveled at wide open throttle in 20 sec from a common starting line. Fig. 10 shows that the 1915 car has traveled 900 ft; the 1935 model, 1400 ft; the 1951 model, 1675 ft; and the 19XX, 1700 ft from the starting line. The performance of the 19XX was deliberately designed to match the 1951 standard car. The entire gain in efficiency from the 12/1 engine and the resulting low engine rpm to car speed ratio was taken in economy.

In comparing economy, the four cars were lined up at a common starting line, with ten gallons of gasoline to each car. They were driven together at 50 mph until they ran out of fuel. The lower part of Fig. 10 shows the distances traveled from the starting point to where the fuel ran out. The 1915 car traveled 68 miles; the 1935, 115 miles; the 1951, 186 miles; and the 12/1 compression ratio 19XX traveled 255 miles before running out of gas. The improvement between the 1915 and 1951 was over 270%, made possible in large degree by the cooperative efforts of the automotive and petroleum industries over the past years. Now, if the 1915 is compared with the 19XX, the improvement is over 370%.

The engine designer can do much to raise compression ratios and improve efficiency by building mechanical octane numbers into the engine. Mechanical octane numbers are the sole responsibility of the engine designer. They represent the design features of the engine which permit it to operate at higher compression ratios on a given fuel octane number. There are many engine factors which might come under the heading of mechanical octane numbers, such as spark advance, carburetion, combustion-chamber shape, and to some extent, automatic transmissions.

As an illustration of mechanical octane numbers, an experimental 253 cu in. engine was built to operate at 7.7 compression ratio on regular gasoline. Fig. 11 compares performance and economy of two stock cars, one a standard car at 6.7 compression ratio and the other powered by the Research experimental engine. Twenty sec after a standing start at wide open throttle, the standard car has moved out a distance of 886 ft while the experimental car has traveled 1042 ft – a margin of 17.5% in performance.

A comparison of economy between the two cars is shown in the lower curves of the figure. Starting them off together, each with a supply of 10 gal of gasoline, and driving them together at a speed of 50 mph, the standard car traveled 175 miles and the Research car traveled 225 miles before running out of gas. This is more than 20% improvement in a car powered with an engine using current regular grade gasoline.

Viewed from a national perspective, a 20% improvement in fuel economy becomes impressive indeed. Since the yearly consumption of gasoline in the United States is about 40 billion gal, it would

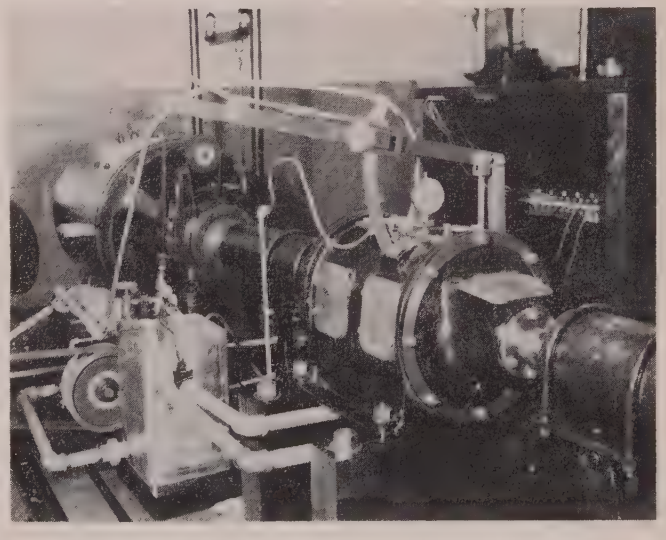


Fig. 7 – Infinitely variable transmission on dynamometer

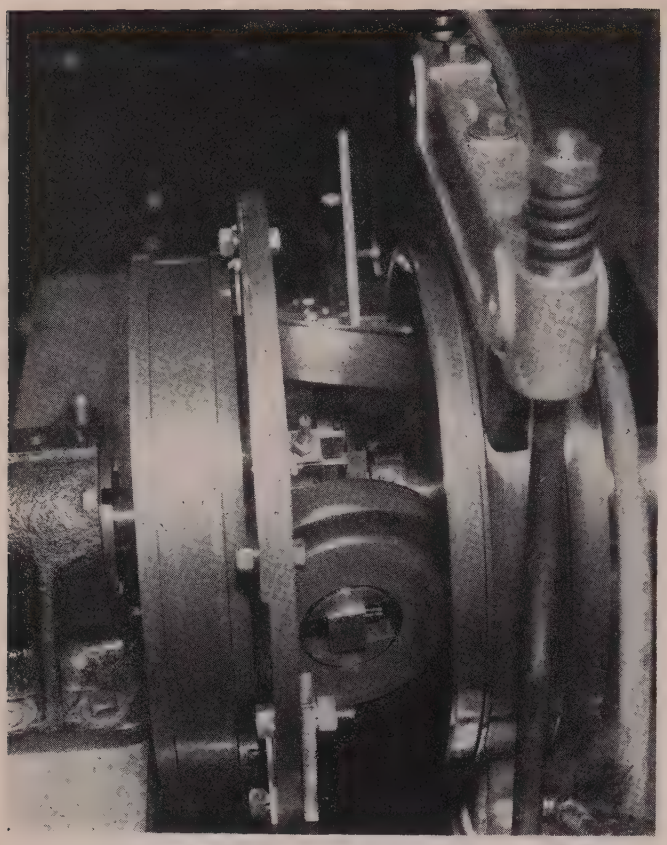


Fig. 8 – Torus race and roller assembly on dynamometer test

amount to a saving of 8 billion gal or, at 25 cents a gal, 2 billion dollars! Such is the incentive for improving the efficiencies of automotive engines by increasing compression ratios and by building in mechanical octane numbers. These two foregoing sets of tests summarize what has been learned in the past few years about increasing engine efficiency.

Fig. 9 – Cadillac test cars on desert near Phoenix

Design Problems of High-Compression Engines

The problems relative to the design of high-compression engines are not difficult once they are understood. A considerable number of very high-compression experimental engines, both sixes and eights, have been built over the past few years. The results of tests on the first of these engines, a 12.5/1 compression ratio, 7 bearing, line-6, were published in 1947.[2] It was demonstrated that when the structure as designed was rigid enough to carry the high-compression loads without excessive distortion, roughness was no longer an insurmountable problem.

This fact was further substantiated by the results of experience gained with the 287 cu in. V-8, high-compression fuel test engine which the Research Laboratories built and supplied to many oil companies for use in fuel development.

Before examining the engine-transmission relationship problem, it may help to sum up the basic facts on high-compression engines which have been learned by engine research in the past few years.

In general, it has been found that compression ratios above 8/1 require bearings on both sides of each crank throw in order to achieve the silky smooth operation of some recent engines. The entire engine structure must be more rigid than it usually has been in the past. Pistons, rods, and bearings must obviously be designed to carry the higher loads.

Higher compression ratios, above approximately 8/1, require overhead valves in order to maintain high volumetric efficiency. This accounts for the present trend toward overhead valves, which may be expected to continue. Finding room for overhead valves and spark plugs at the higher compression ratios sometimes presents a difficult design problem. Since the combustion volume is small at higher compression ratios, it becomes essential that the valves clear the bore. This can be accomplished best and still meet the requirements of good breathing by large bore, short stroke engines. The original Research high-compression engines were designed with this point in mind.

The desirability of using low stroke/bore ratios (S/B) in the design of high-compression engines gave rise to the question of the effect of S/B ratio on economy (Fig. 12). This factor was investigated in two engines of equal displacement and equal compression ratio, but with widely different S/B ratios. Engine A had an S/B ratio of 0.71 and engine B an S/B ratio of 1.33. The two engines were installed in cars of the same weight and body shape. The distributor spark-advance curve was adjusted on both engines to give the same loss from maximum power at comparable speeds.

Fig. 12 compares constant speed, level road economy data on cars A and B. In making comparisons of this type, fuel fishhooks are usually run at each speed using a variable jet carburetor, thus

eliminating the effects of carburetion. The data shown indicate that for all practical purposes the economy of one car is the same as that of the other.

Later, fixed jet carburetors were developed for cars *A* and *B*, keeping the relation between fixed and variable jet as nearly the same as practical in both cases, so that cross-country comparisons of tank mileage could be made. Fig. 13 shows the results of six 100-mile cross-country trips in which these two cars were driven together over the same route. It can be seen that both cars gave practically the same trip economy. The average for the 600 miles showed that car *B* with 1.33 *S/B* ratio gave 2% better economy than car *A* with 0.71 *S/B* ratio. Two per cent is regarded as within the limit of error of such measurements.

From the results of these and many other tests, it was concluded that the engine designer has quite a wide selection of stroke/bore ratios to work with without paying a penalty in fuel economy.

Ideal Engine-Transmission Relationship

What would a modern high-compression engine like in the way of a transmission to transfer its efforts to the road most efficiently? Before trying to answer this interesting question from an engine designer's viewpoint, one very important ground rule must be laid down. The authors will show what the engine wants the transmission to do, but will take no responsibility for designing a transmission to do it. The following analysis should stimulate closer cooperation between engine and transmission designers. It is hoped that the transmission engineers will set forth their requirements for an ideal engine in the discussion of this paper.

Before presenting the analysis of ideal engine-transmission relationships, the aim of this study should be noted. It is realized that much time and effort has been spent on transmissions, and the present excellent automatic transmissions are the result. However, just as there are many future gains in efficiency possible by increasing compression ratios of engines, there are many gains in economy possible by developments in transmissions to most effectively utilize the potentials of the engine. The aim of this paper is to show the large incentive to fit the transmission to the engine better and vice versa. It is realized that the problem is as broad as it is long. There are many engine variables and an equal number of transmission variables. The large gains in economy possible through the use of high-compression engines with new transmissions, give engineers a very important incentive to continue developments in both fields. In fact, the authors believe that a combination power-transmission system will be the next step in automotive development. The combined engine-transmission power package is the most fundamental development facing the industry today, and this paper will show the large gains in economy which it can make possible.

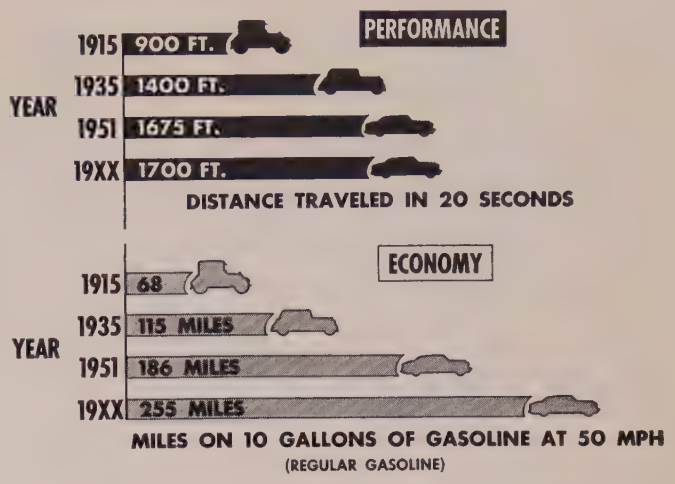


Fig. 10 – Performance and economy chart for four Cadillac test cars

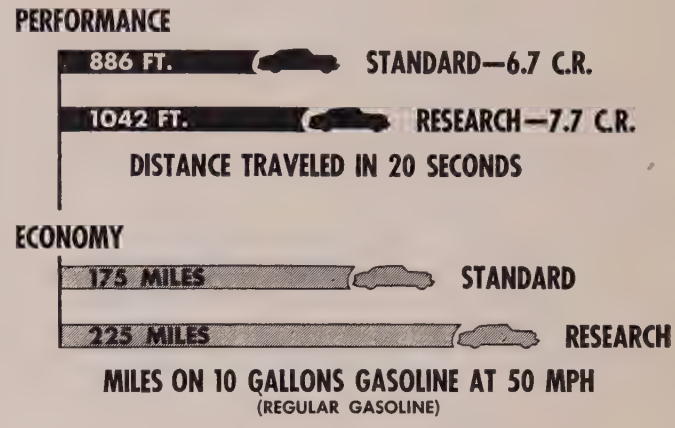


Fig. 11 – Example of mechanical octane numbers

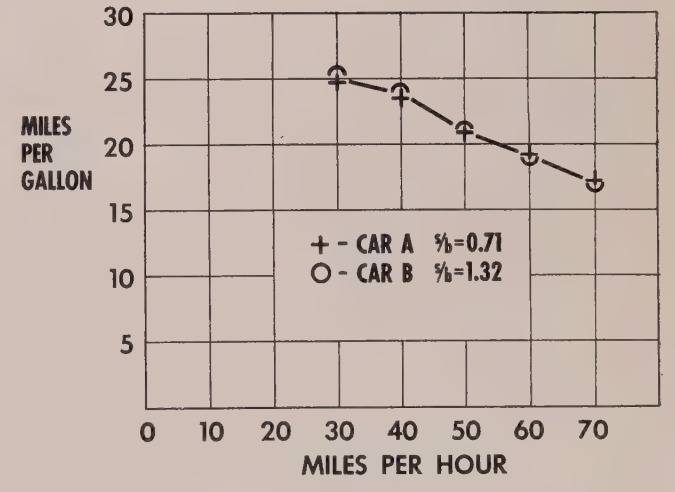


Fig. 12 – Effect of stroke/bore ratio on constant speed, level road economy

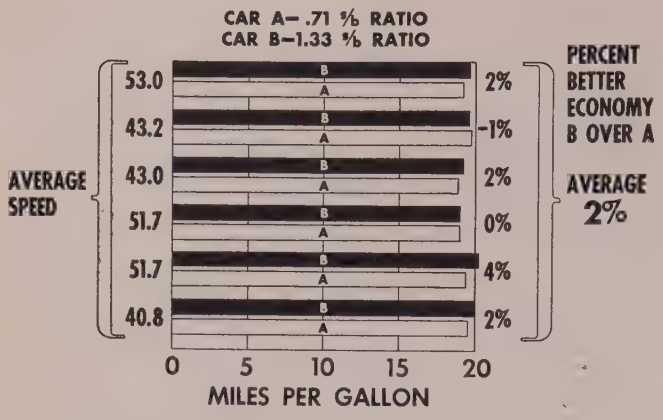


Fig. 13 – Effect of stroke/bore ratio on cross-country fuel economy

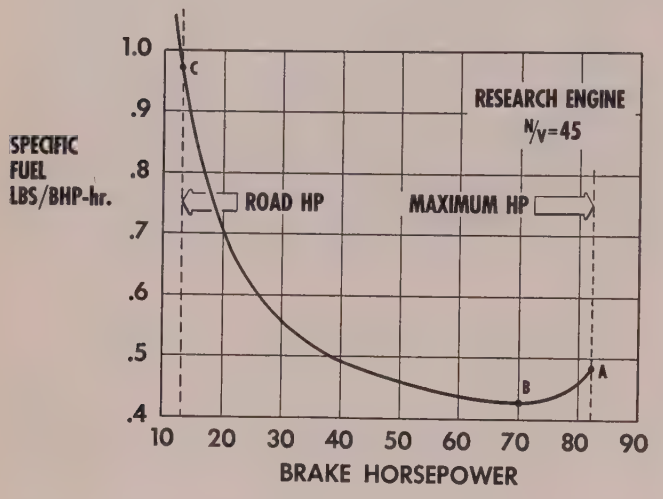


Fig. 14 – Typical cross-section at constant speed of 2000 rpm

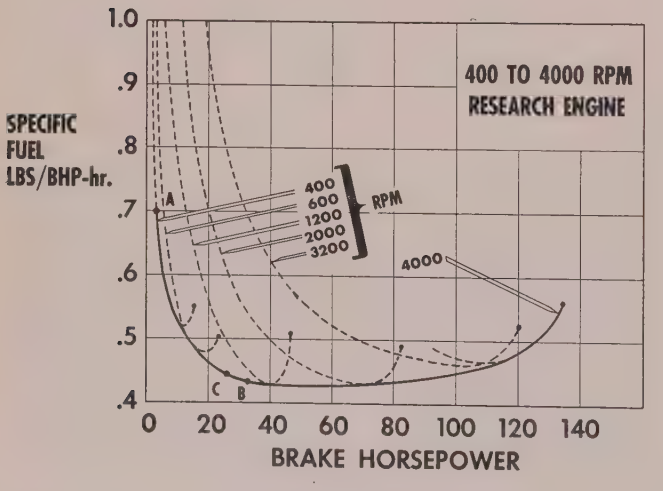


Fig. 15 – Individual cross-section curves under wide variety of operating conditions

To begin the analysis, consider a typical curve (Fig. 14), which shows the relationship between brake specific fuel and brake horsepower at the constant speed of 2000 rpm for the Research engine. This may be called a cross-section curve, since it shows how the specific fuel consumption varies in the cross section between road load and performance power. This curve is obtained by leaning the A/F mixture from point A, the mixture ratio at which the engine develops maximum power, to B where it develops best economy. The throttle is closed from wide open at B to the throttle opening required to develop road load horsepower at C. The specific fuel consumption at point B is obviously only a fraction of that at point C. In fact, at the minimum specific fuel consumption which occurs at 70 bhp, the specific fuel is only 0.43 lb per bhp per hr, in comparison with the road load point C, 13 bhp, where the specific fuel consumption is 0.97 lb per bhp per hr.

Superimposed on Fig. 15 is a whole family of individual cross-section curves. It shows the relationship between specific fuel consumption (lb per bhp per hr) and bhp under a wide variety of operating conditions for the 253 cu in. Research engine. One might look at Fig. 15 as a character sketch of an engine, because it discloses a great deal about how the engine will respond to almost any operating conditions.

When enough of the individual cross-section curves are included, the whole group can be enclosed by an enveloping curve. Under the operating conditions represented by any point on this curve, the engine will develop its best economy. It is apparent, therefore, that if the ideal transmission is to deliver best economy, it should permit the engine to operate on this curve at all times. In other words, the transmission controls should always adjust the engine setting to operate at the point of minimum specific fuel consumption.

In order to illustrate the method of arriving at the engine-transmission relationship for best economy, the following typical conditions will be discussed:

1. Minimum engine speed.
2. Typical point of minimum specific fuel consumption.
3. Typical performance calculation.
4. Engine limited to speed of maximum horsepower.

First, the power required to drive the car at various speeds is computed. For this the following formula is used:

$$\text{hp} = \frac{(KW + K_1AV^2)\ V}{375}$$

where:

$K = 0.012$
$K_1 = 0.00125$

W = Test weight (car weight + 600 lb test weight)
A = Projected frontal area, sq ft
V = Car velocity, mph
In this analysis a stock car having $W = 4000$ lb is used.

Fig. 16 shows such a road load power requirement curve for a standard stock car. It can be noted from the figure that it required 3.2 bhp to drive the vehicle at 20 mph on a level road. Referring again to Fig. 15, it can be seen that 3.2 bhp is developed at point A on the curve of best economy. At this point the engine uses fuel at the rate of 0.70 lb per bhp per hr, from which the mpg can be computed from the relation:

$$\text{mpg} = \frac{N \times C}{F \times P}$$

where:

N = Speed, mph
C = Lb per gal of gasoline
F = Specific fuel consumption, lb per bhp per hr
P = Hp

Arbitrarily, the minimum speed of the engines used in these studies was limited to 400 rpm, which accounts for the relatively high fuel consumption at point A. In choosing 400 rpm as the minimum speed, the authors are fully aware that it may be on the low side for present engines. However, remember that the objective of this discussion is to establish the ideal engine-transmission relationship for best economy. The engine has to be throttled considerably in order to develop 3.2 bhp at 400 rpm. An engine having less displacement would, of course, operate more efficiently at this road load. The effect of engine size will be discussed later in the paper.

The second typical condition illustrates the point of minimum specific fuel consumption. From the road load power curve, note that at 60 mph it requires 26 bhp to drive the car on a level road. Fig. 15 shows that at 26 bhp, point C, on the curve of best economy, the engine uses fuel at the rate of 0.445 lb per bhp per hr, as compared to 0.70 lb per bhp per hr at 20 mph where the 400 rpm minimum speed limitation made it necessary to close the throttle to meet road load horsepower requirements. At level road driving, the transmission would operate the engine on the solid, minimum specific fuel envelope curve.

The third typical condition which must be considered is the required performance, represented by acceleration and hill climb in the car, and by horsepower available for these factors to the engineer. In this example, operating conditions were selected to match the performance of the standard 4000-lb stock car (Fig. 17), which requires 30.2 bhp at 20 mph in addition to the 3.2 required for

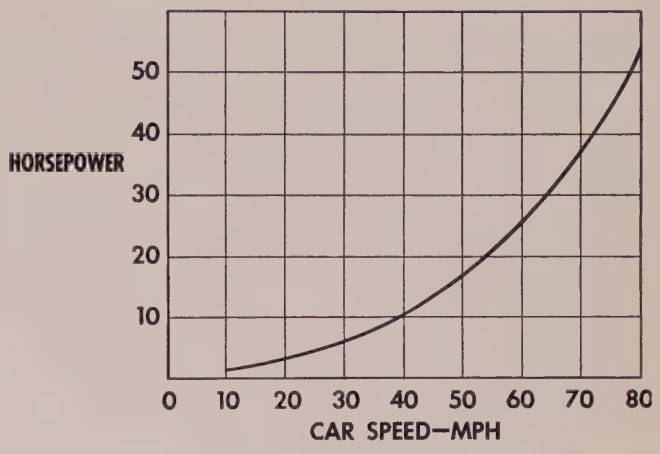


Fig. 16 – Road load horsepower requirement curve for 4000-lb stock car

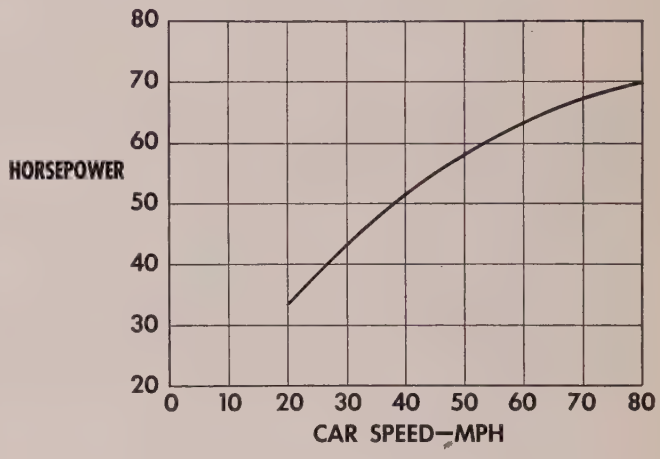


Fig. 17 – Power required at rear wheels to match stock car performance

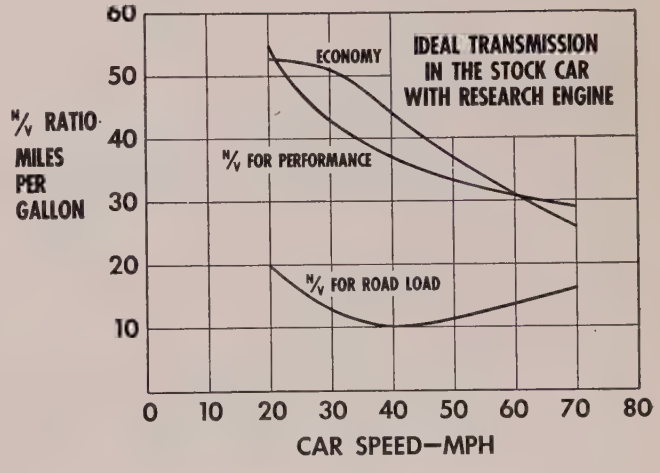


Fig. 18 – Level road, constant speed fuel economy with ideal engine-transmission relationship

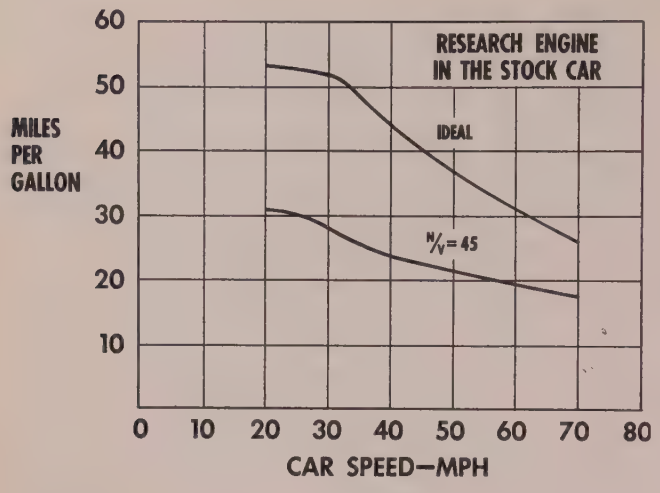


Fig. 19 – Constant speed, level road economy, showing ideal versus fixed ratio

road load, or a total bhp of 33.4. In order to develop 33.4 bhp and still keep on the best economy curve (Fig. 15), the transmission must obviously permit the engine to increase its speed to 1100 rpm (point B). It is seen that in this ideal engine-transmission relationship, the transmission should in effect act as a moderator to keep the engine always operating at its best efficiency to develop the required power. Once the desired performance speed is determined, the relation between engine rpm and vehicle speed (N/V) can be established. The transmission controls would pick the engine speed to develop the required horsepower at a point of minimum fuel consumption. This is again on the solid envelope curve.

Fig. 18 shows the level road, constant speed fuel economy over the speed range from 20 to 70 mph resulting from the above ideal engine-transmission relationship, assuming no losses between the engine and road. Also shown is the road load ratio of engine rpm to car speed (N/V) and the N/V to give the stock car performance over the speed range. For instance, at 40 mph the road load N/V is 10 and performance N/V is 37. It is recognized, of course, that much higher N/V ratios are required to start the car from rest.

Fuel economy figures such as those shown in Fig. 18 are difficult to evaluate unless they can be compared with a familiar yardstick. Fig. 19 compares level road, constant speed fuel economy with the two types of transmissions, the ideal and fixed N/V in the stock car equipped with the Research 253 cu in. engine. The curves were computed without considering loss in efficiency between engine and road in either case. The data are presented on a comparative basis, and it is felt that from the engine man's standpoint it is fair to ignore transmission losses in this study since they vary with the mechanism used to obtain the results. Moreover, since this study does not propose a transmission design, the actual losses are unknown. The figure shows that the use of the ideal transmission would make possible gains in constant speed fuel economy of from 70 to 48% over the speed range from 30 to 70 mph. This analysis presents what is inherently possible from the engine standpoint. How much of this is obtained will depend upon the transmission design; but even allowing for large losses, a large potential gain remains.

On the other hand, automobiles are not driven on level roads at constant speeds very much of the time. Fig. 20 is presented to show how operating conditions affect the gains in economy realized from the use of the ideal transmission. Two curves are shown in the lower half of Fig. 20 illustrating the relation between specific fuel consumption and brake horsepower for the experimental 253 cu in. engine in the stock car at 40 mph. The ideal transmission permits the engine to follow the best economy curve, whereas the standard transmission with fixed N/V ratio of 45 forces the engine to follow the other curve. The stock car requires 10.5 bhp to propel it at 40 mph on a level road. On the best economy curve at point A, 10.5 bhp can be developed at 400 rpm with fuel consumption of 0.518 lb per bhp per hr, while it requires 1.02 lbs per bhp per hr to develop the same 10.5 bhp at 1800 rpm (point B) on the fixed N/V curve. Now, point B is equal to 22.4 mpg and point A corresponds to 44 mpg as plotted on the upper set of curves. It is apparent that under these conditions it is possible

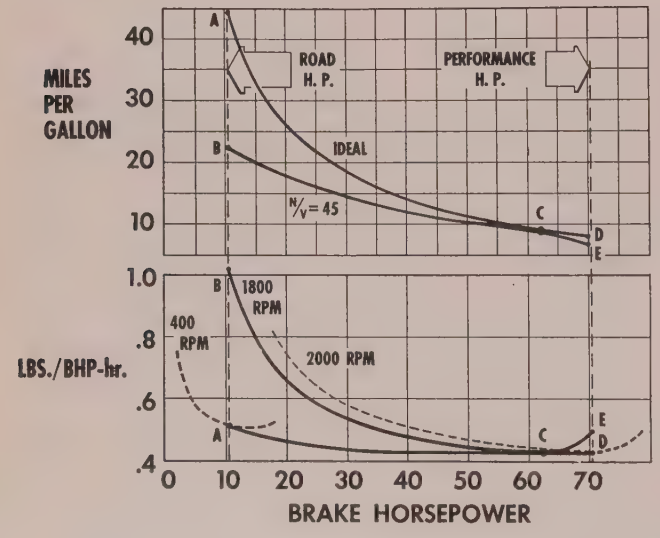


Fig. 20 – Cross-section at 40 mph with ideal engine-transmission relationship versus fixed engine speed – car speed N/V ratio

to almost double the miles per gal if an ideal transmission were possible.

The curves of Fig. 20 also show that, as the throttle is opened with the standard or fixed N/V transmission to develop more horsepower for performance, the fuel consumption is rapidly reduced until at C, with the throttle open, it exactly equals the fuel consumption with the ideal transmission. At E, corresponding to 70.5 bhp required for full performance at 40 mph, the standard transmission has required the engine to cut in the richer A/F power mixture, while the ideal transmission has permitted it to increase the speed to 2000 rpm and develop the 70.5 bhp without enrichment (point D).

This ability of the ideal transmission to let the engine operate in the lower speed range without enrichment, results in from 10 to 15% improvement in economy over present transmissions whenever the engine is operated at full throttle. At the higher car speeds it would, of course, be necessary to enrich the A/F mixture in order to obtain performance and top speed.

The ideal transmission will obviously limit the top speed of the engine to the point at which it develops maximum power. In some cases this would result in lower specific fuel consumption and slightly higher top speed.

Analysis of Fig. 20 discloses several important facts about the gains in economy with the ideal as compared to the standard transmission. These are the factors which will be important in the combination power-transmission system:

1. The greatest improvement in economy occurs at road load, around 98% in this case.
2. The improvement in economy rapidly decreases at greater loads.
3. Since the driver is continually operating between road and performance power, his overall gain in tank mileage will obviously be much less than the 98% he obtained under constant speed, level road conditions.
4. The effect of country and city types of driving on economy will be greater with the ideal transmission. However, use of the ideal transmission will always result in an improvement in economy under all conditions. This fact represents an important advantage, since a large percentage of fuel is used in city driving.
5. With the ideal transmission, road load economy is independent of performance, while with the standard transmission the greater the performance, in general, the lower the road load economy.
6. The higher the performance factor, the greater the economy gains resulting from the ideal engine-transmission relationship.

In the foregoing discussion, an experimental Research engine was used to illustrate the gains to be obtained by use of the ideal transmission. It might be interesting to apply the same engine-transmission relationship to a standard production car and engine. The curves in Fig. 21, represented

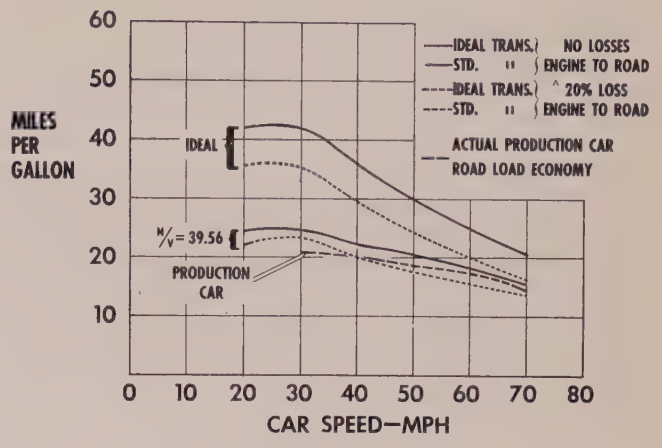


Fig. 21 – Constant speed, level road economy comparisons with production car and engines

by solid lines, show level road, constant speed economy of the standard car compared to the same car with the ideal transmission. Both curves were computed by assuming no losses between the engine and the road. It is apparent that improvements of 70 to 30% are possible in road load economy over the speed range from 30 to 70 mph with the ideal engine-transmission power package.

In computing the dotted curves shown by Fig. 21 a 20% loss in efficiency was arbitrarily assumed between engine and road. It can be seen that the actual values of mpg have been reduced in both

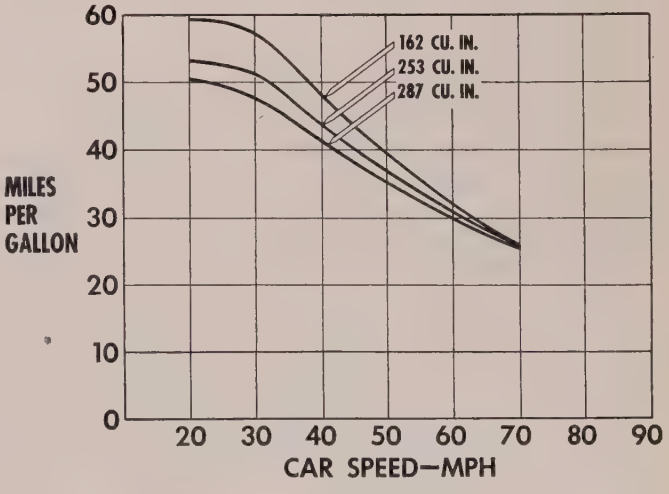


Fig. 22 – Effect of engine size on road load economy, with ideal engine-transmission relationship

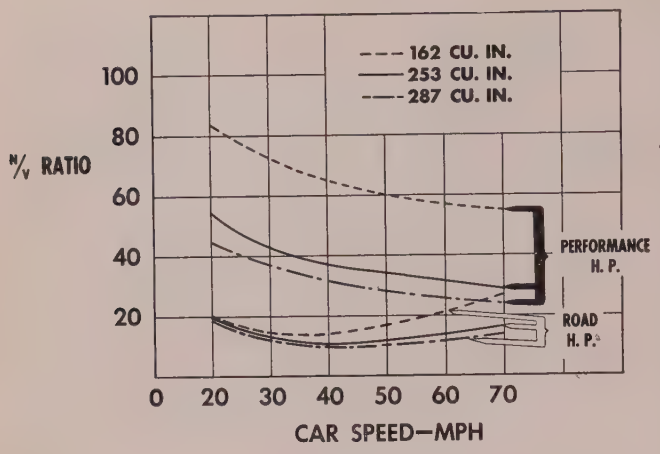


Fig. 23 – Effect of engine size on N/V ratio, showing best economy with stock car performance

cases; but even under these conditions, a 60 to 18% improvement in economy is shown.

Fig. 21 also shows the actual measured level road, constant speed economy of the standard car for comparison. Again, it must be emphasized that cars are not driven at constant speed on level roads and that the improvement in overall tank economy, while substantial, is not likely to be nearly as great with the ideal transmission as is indicated by these constant speed economy curves.

Effect of Engine Size on Road Load Economy

The effect of engine displacement on road load economy is shown by Fig. 22. Three engines having compression ratios of approximately 8/1 and the following displacements were used for this study:

1. 162 cu in.
2. 253 cu in.
3. 287 cu in.

The curves show a considerable advantage in level road fuel economy for the small engine. The reason the small engine has an advantage in the low-speed range is because internal-combustion engines are not very stable at very low rpm, and engine speed has been arbitrarily limited to 400 rpm for the investigation covered in this paper. It requires 6 bhp to propel the stock car at 30 mph on a level road. At 400 rpm the 162 cu in. engine, to develop 6 bhp, can operate almost at its best economy point while the 253 cu in. engine must operate at a much poorer load factor to develop the same horsepower. Obviously, the 287 cu in. engine must be throttled still further to develop the required 6 bhp at 400 rpm. It can be seen that the difference in economy between the three engine sizes diminishes gradually to 70 mph, at which point they are all about the same.

Before drawing any conclusions from Fig. 22, examine Fig. 23 and see what effect engine size may have on some of the critical transmission requirements. Fig. 23 shows the road load relation between engine and car speed (N/V) to obtain best economy. It also shows N/V ratios required to equal the stock car acceleration rates. It can be seen that the small engine requires a larger change in N/V ratio from road load to performance than the larger engines do. The small engine would also limit the top speed of this car.

In spite of the fact that the 162 cu in. engine gives appreciably better level road economy in combination with the ideal transmission in the low car speed range, this advantage appears to be more than offset by other limitations such as lower top speed, high N/V ratio changes, noise, and other factors involved with small high-speed engines. As is the case in most automotive applications, the selection of an ideal engine size for a particular car and transmission will represent a compromise based on experience with actual equipment.

Ideal Engine-Transmission Relationship

It has been shown in the foregoing analysis that very substantial improvements in economy can be accomplished by an ideal engine-transmission relationship. The authors believe that future gains in economy will come through development of the concept of a combined engine-transmission system in one package.

While it is difficult to assign firm values to the possible gains without the benefit of actual tests, it is believed to be entirely justifiable to predict a future overall saving of 25 to 35% in the nation's fuel bill by transmission developments. This is about half of the theoretical gains shown in the theoretical analysis made at road load.

It is realized, of course, that this improvement must be achieved without sacrificing the pleasing characteristics which make present automatic transmissions so popular. In fact, the so-called pleasability will be greatly improved with the ideal transmission in combination with engines up to modern standards of noise and smoothness. The power-transmission package would bring customers a new concept in motoring pleasure and economy.

One reason it is so difficult to foresee all the gains resulting from an ideal power-transmission system is that it so completely changes the nature of both engine and transmission.

The engine, for instance, would seldom operate much under 70 to 80% of its full power output. One of the car manufacturer's biggest headaches results from operation by some drivers at very light loads. One of the principal causes of spark-plug fouling is light-load operation. There is also a good possibility that elimination of light-load operation would reduce octane depreciation from deposit build-up because a good share of combustion-chamber deposits, particularly those which

are likely to cause preignition and wild ping, are accumulated during light-load running. Anything which will reduce combustion-chamber deposits will result in creating mechanical octane numbers or, in this instance, transmission octane numbers. The fact that, in the low-speed range where knock is critical, the engine would not be permitted to operate at full power, might contribute somewhat to its mechanical octane numbers.

It is unlikely that an engine operating near its full power output at all speeds would need a vacuum spark advance. The vacuum spark advance or its equivalent is employed on most engines because of the slow-burning light-load mixtures. Elimination of the vacuum spark advance would be a welcome simplification.

Engine valve-timing requirements would need to be carefully investigated under the new operating conditions of the ideal transmission to provide good high-speed breathing together with solid low-speed performance.

An ideal transmission might also simplify carburetion because of the near full-throttle operation. Operation at low vacuum might improve oil control and influence the deposit accumulation picture because less oil would get into the combustion chamber. In other words, the development of a commercial ideal transmission would open up a whole new field of engine development.

Up to the present time, automotive transmissions have generally been developed with the object of eliminating the clutch pedal and thereby making driving easier and more pleasant. The use of the ideal transmission-engine combination would add the incentive of much improved efficiency without sacrificing other desirable features in any way.

High-Compression—Ideal Transmission System

A summary of the gains in economy which are possible with a combination of high-compression engines and ideal transmissions, shows the incentive for further intensive work. This paper has shown how a gain of from 25 to 35% is easily possible with an ideal transmission. It has also been shown that large gains of 25 to 35% are possible with engines of 12/1 compression ratio. By obtaining the advantage of gains from both high-compression engine and ideal transmission developments through further research, a total saving of 45 to 60% could be made.

It seems entirely possible, therefore, to reduce gasoline consumption by half without a sacrifice in car size, performance, or roominess. To obtain a 50% increase in the present miles per gallon with normal driving is indeed an incentive for automotive engineers to take advantage of the potentials in the high-compression engine and the ideal transmission.

Progressive industry has always had a goal in the future, set by the research of today. This study presents such a goal as a challenge for future development.

When the goal is reached, motorists will go half again as far on a tank of gasoline. This will permit valuable oil resources to be used more effectively and more efficiently. If oil wells are considered sources of miles of transportation, each well will produce 50% more than the present mileage. Where 20 mpg in the family car is now considered, 30 will be obtained in the future. The savings, made up of the total of each motorist, will reach into billions of dollars per year.

Automotive engineers will have performed one of the basic jobs of engineering – to make the most efficient use of natural resources.

Acknowledgment

The authors wish to acknowledge the many helpful suggestions in the preparation of this paper and the contributions of their associates in the Research Laboratories as well as in the central office engineering groups and manufacturing divisions.

Additional References

1. "More Efficient Utilization of Fuels," by C. F. Kettering, SAE Transactions, Vol. 14, 1919, pp. 201-219.

2. "Fuels and Engines for Higher Power and Greater Efficiency," by C. F. Kettering, *SAE Journal* (Transactions), Vol. 53, June, 1945, pp. 352-357.

3. "Increasing Engine Thermal Efficiencies," by J. M. Campbell, D. F. Caris, and L. L. Withrow, SAE *Quarterly* Transactions, Vol. 3, April, 1949, pp. 341-352.

4. "Automatic and Semiautomatic Transmissions," by H. R. Greenler, *SAE Journal* (Transactions), Vol. 54, August, 1946, pp. 440-448.

5. "Hydraulic Transmission Performance," by A. H. Deimel, *SAE Journal,* Vol. 54, July, 1946, p. 38.

6. "Torque Converters," by R. J. Miller, *SAE Journal,* Vol. 54, August, 1946, p. 54.

7. "Automatic Transmission Control Systems," by O. K. Kelley and M. S. Rosenberger, SAE *Quarterly* Transactions, Vol. 1, October, 1947, pp. 559-565.

8. "Automatic Transmissions – Today – Tomorrow –," by A. H. Deimel, *SAE Journal,* Vol. 58, September, 1950, pp. 23-26.

9. "Motor Vehicle Powerplants and Transmission Line Units," by Robert Cass, *SAE Journal,* Vol. 52, August, 1944, pp. 36-37.

10. "Hydraulic Transmissions for Motor Vehicles," by A. H. Deimel, *SAE Journal* (Transactions), Vol. 53, January, 1945, pp. 10-18.

Discussion on following page ➡

DISCUSSION

Design Trends Toward Ideal Transmission Seen

– W. B. Herndon
Detroit Transmission Division, GMC

THIS paper has offered much informative and factual data on possibilities for improved fuel economy with better engine-transmission relationship. While much of the data is not new, it should give transmission designers new incentive and encouragement to improve fuel economy in future designs. It has been shown that great improvement has been made in constant speed fuel economy since 1915. We must not overlook the fact, however, that the average motorist fares little, if any better, in actual miles per gal than he did in 1915. The reason, of course, is that he drives much faster in modern cars than he did in those of 1915 vintage. In other words, the gains that have been made in engine efficiency have been largely taken in higher speed and greater luxury of travel, and not in fuel economy. A real improvement in miles per gal would be one of the best contributions engineers could make to the cars of the future.

There are two basic principles that govern what can be done in economy and performance. The first is that the slower an engine can be run and still produce the power required, the greater will be the economy. The second is that the maximum performance from an engine can be obtained by allowing the engine to run at full throttle on its horsepower peak, and have the transmission vary the reduction between engine and rear wheels from a maximum reduction at a standing start to a minimum reduction or maximum overdrive ratio at the top speed.

The first of these basic principles has been discussed in this paper. The authors did not, however, explain why this lower speed engine produces better economy. This is certainly no secret and no mystery. It is simply the result of less power going to waste when an engine is running at low speed than when it is running at high speed. An engine is a very inefficient mechanism. In other words, the friction horsepower and pumping losses of the engine are appreciable. Referring to Fig. 16, which plotted road load horsepower against car speed, if you were to lay the engine friction horsepower curve on top of the road load car curve, you would find that up to some rather high speed, perhaps 60 mph, more power is used to drive the engine than is used to drive the automobile. In other words, the average car owner wastes more gasoline in running his engine and overcoming the engine friction and pumping losses, than he uses to propel his car. Since this is true, it is perfectly obvious that the slower an engine can be run and still produce the required amount of power, the better the economy. The same rule applies to a small engine versus the big engine. The smaller engine naturally having the smaller losses will give the best economy at a given horsepower output and engine speed. To summarize, large gains in road load economy are possible by reducing the engine revolutions per mile, per hour, because that reduces the losses of the engine.

The second principle is that maximum performance could be obtained with a transmission that allows the engine to run at its horsepower peak. For my ideal engine transmission combination, I would like to add another feature beyond those described in the paper. I think the ideal transmission should be so designed that when the accelerator is pressed through the detent to the floor, the engine would speed up to its horsepower peak and run there, while the transmission changed ratios automatically as required to keep the engine at its horsepower peak. Such an arrangement would give a tremendous increase in performance. Economy, of course, would be poor, the noise level high, the tire problem difficult, and so forth, but it would add ideal performance as well as ideal economy to the ideal transmission, at the driver's option. Of course, these are all theoretical considerations and impossible of absolute achievement. All engineering solutions, however, are compromises.

In the case of the new automatic transmission, the Dual Range Hydra-Matic, we have recognized these possiblities of low engine speed for economy and high engine speed for performance, and have designed the transmission to take advantage of them. With 1952 models, we are using axle ratios of approximately 3/1 which is the lowest that have ever been used on production passenger cars. These axles give low engine speeds in the cruising gear. Then either automatically with the accelerator, or manually with the finger-tip shift lever, it is possible to step the engine speed up by almost 50% and thereby approach the ideal performance condition. While this Dual Range Hydra-Matic, like all mechanisms, is a compromise, we think it is a good practical compromise that approaches the ideal. It has given us appreciable gains in economy and performance.

I would like to say a word of caution to the transmission designers who attempt to meet the ideal with hydraulic drives. The efficiency of the drive must not be overlooked and it must be very high if the possible gains are to be realized. Even seemingly small losses through the transmission system can completely upset the calculations for improved economy. Referring again to Fig. 16, notice that at 30 mph, for example, the power required to drive the car is only about 6 hp, and that even up to 60 mph no more than 25 hp is required. We think and talk in terms of 200-hp engines but tend to overlook the fact that the average driver seldom uses more than a small fraction of this power. Obviously, if the mechanism that is doing this ratio changing job has losses, even though the losses are small, they can be a high percentage of the total. For example, suppose the transmission system took only 1 hp to drive at 30 mph. One horsepower doesn't seem like much to lose when you have a 200-hp engine. But 1 hp is 17% of the power required at 30 mph. Obviously, such a loss, though small of itself, could easily nullify the expected economy gains.

In designing the Hydra-Matic transmissions, we have been very conscious of the importance of high efficiency. We continue to use gearing to multiply torque as gears are the most efficient means known for doing this job.

We know there is no substitute for the smoothness of a fluid start, so we use a fluid member. To minimize its losses and still retain its smoothness, we partially clutch it out in the driving ranges.

A recent efficiency improvement in the Hydra-Matic is a

new pumping system. The new variable capacity pump, now used in all Hydra-Matics, pumps only the amount of oil required by the transmission, and thereby eliminates unnecessary pumping losses.

I believe the automatic transmission design trend will be toward the ideal. We will have lower engine speeds for economy, quietness and smoothness, high engine speeds at the driver's option for maximum performance, and more emphasis on obtaining the maximum in efficiency in automatic transmissions of the future.

Engine and Transmission Combinations Studied

– V. J. Jandasek
Ford Motor Co.

WHILE the fundamental characteristics of the 4-stroke gasoline engine and its relationship to the passenger-car transmission have remained unchanged since its inception, there have been many improvements in performance and fuel consumption. This paper effectively brings the picture of the efficiency aspect of the engine-transmission relationship uptodate.

In some of our studies of engine and transmission combinations we have considered variable N/V conditions. The results of one such dynamometer study made with a production engine and 4000-lb car are shown in Fig. A. The engine in this case had a compression ratio of 7/1. The upper curve in this figure shows the road load fuel consumption in mpg at constant speeds ranging from 20 to 80 mph with the N/V varied to operate the engine at 75% of its full power output at each speed. The dotted portion of this curve from 20 to 30 mph indicates the range of operation where it is not possible to load the engine to a 75% value because of a minimum speed limitation of 400 rpm.

The lower curve on this figure illustrates the fuel consumption obtained with the same engine and car with a fixed N/V of 44.3. A comparison of these two curves shows some considerable gains with the engine operating at 75% power where possible. At 20 mph this gain is 120%, at 40 and 70 mph the gains are 75 and 26% respectively. Considering the many factors involved this represents good correlation with the curves presented by the co-authors.

Proceeding further with this study, Fig. B shows the road load torque ratio required to obtain the same engine loading of 75% power presented on the previous curve. This ratio represents the optimum for obtaining the maximum efficiency from this particular engine and vehicle combination. Superimposed on this curve are the overall torque ratios of a 3-speed converter transmission. These ratios are required for good starting performance, accelerating at higher vehicle speeds, and hill braking. Combining these with the road load ratio gives the ratio coverage required in an ideal transmission. This indicates a considerable increase in transmission ratio coverage as the cost for attaining a higher engine operating efficiency. An obvious, but also very important item, is the fact that the road load ratio requirement necessitates an infinitely variable transmission mechanism.

Speaking as a transmission engineer I can say that we are certainly aware of these problems. Considerable effort is being applied to develop better transmissions than those presently available and I am sure that we will see improvements in transmissions and engine-transmission combinations in the future.

Offers Suggestion for More Desirable Product

– Forest McFarland
Packard Motor Car Co.

THE authors have clearly indicated the factors affecting fuel economy in connection with the engines and transmissions as we know them today. Due to the fact that the

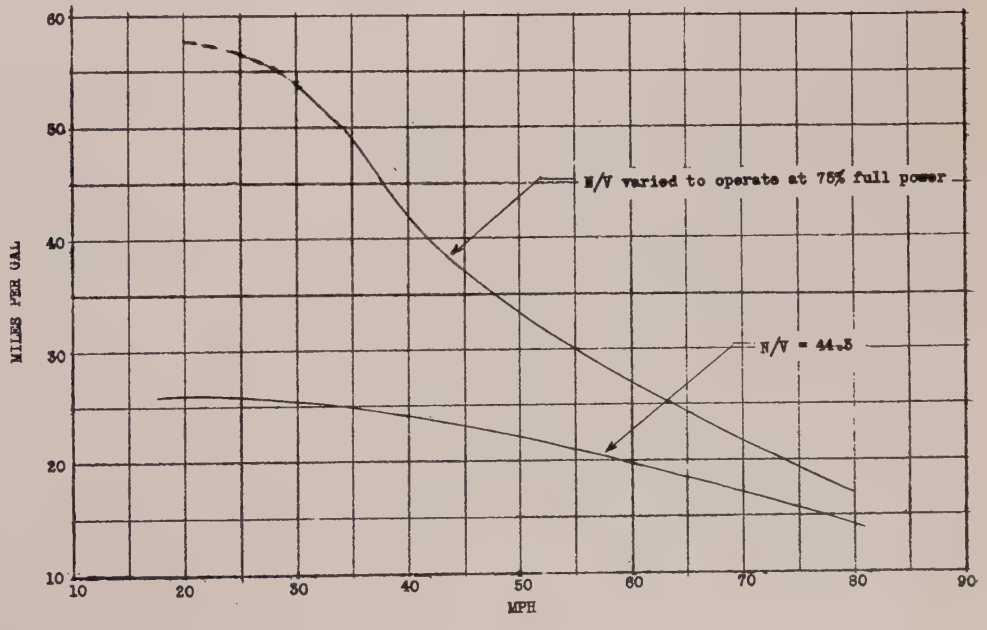


Fig. A – Road load fuel economy with production engine and car. Limitation of 400 rpm per min engine speed does not permit operation at 75% of full power, as indicated by dotted line in 20 to 30 mph range

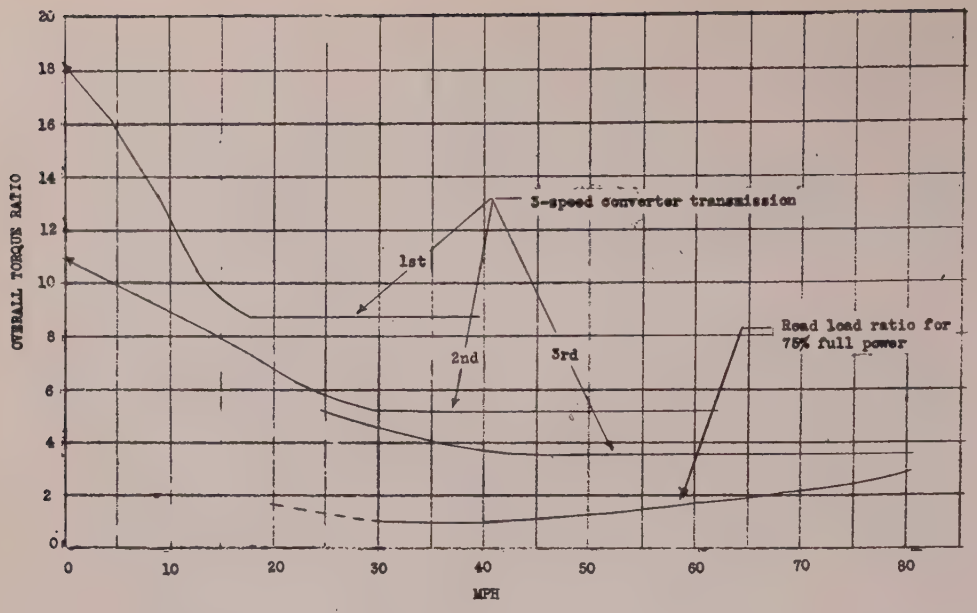


Fig. B – Road load torque ratio requirements. As indicated by dotted line in 20 to 30 mph range, a limitation of 400 rpm per min engine speed does not permit operation at 75% of full power

present engines show much poorer fuel economy at part throttle when the actual compression is low, compared to full throttle when the full compression without throttling effect is obtained, the authors propose to work near the full throttle point to achieve economy, which is certainly one way to solve the problem. This solution compels the driver to shift into a lower speed gear or otherwise obtain the equivalent by transmission manipulation to realize a higher N/V ratio whenever any appreciable increase in power for acceleration is required.

Is there not another line of attack which may give the customer a product which he may find more desirable? Assume that a rear-axle gear ratio giving an N/V of 35 to 40 is used to reduce engine speed and noise to a desirable level at all driving speeds. Assume an engine is selected of sufficient size and power to give performance in top gear to equal or excel that of our best present performance. Assume this engine is such that the compression at part throttle compares with the full throttle value. The resulting efficiency at part throttle or road load should be reasonably near that at full throttle, although admittedly not equal to it.

When more performance is desired a reasonable gain can be obtained by simply depressing the accelerator to full throttle without transmission manipulation. For performance in excess of this value, the conventional "kickdown" arrangements in the transmission would be available, but would be used only occasionally instead of every time any appreciable gain in power is required. Spark-plug fouling and attendant ills would be greatly lessened over that of present engines since nearly constant compression would be obtained. The writer believes an engine transmission combination of this type would be more pleasing to the customer. No solution has been offered leading to the type of engine described, but a number of types have been proposed that may prove very interesting during the course of further development work in an attempt to obtain this result.

The economics of the problem may enter the picture. Is it cheaper to obtain the desired result by a transmission design to suit the present engine – or by an engine design to suit the present transmission? Would the customer not prefer a combination where the amount of transmission manipulation is kept to a minimum?

It will be interesting to see what designs are finally arrived at to obtain the objective of higher engine-transmission efficiency which is surely a worthwhile goal.

Summary of Oral Discussion

Reported by R. L. McWilliams, GMC

THE challenge to design the ideal transmission to overcome the deficiencies of the internal-combustion engine was not denied by the transmission men who commented on the paper. However, several thought that the objectives might be more easily reached through engine improvements; for example, the reduction of engine friction would eliminate high power losses which would be reflected in greatly improved fuel economy. As one expert commented, "The ideal engine might be a variable displacement engine – then the ideal transmission could be simply an extension of the crankshaft to the differential."

In a more serious vein, the ideal transmission, which would permit the engine to operate at near maximum torque at all times through sensing of road-load requirements as dictated by the driver for performance and cruising speeds, will likely develop through evolution as a result of the continuing effort of industry.

To gain this control of engine speed to car speed at near constant torque, thus permitting the engine to operate on minimum brake specific fuel consumption rates all of the time, further automaticity might add to the transmission power losses. This factor should receive close attention, for even in today's automatic transmission the power loss is a considerable percentage of the car's road load resistance.

The aerodynamic aspect of car design should not be overlooked. As pointed out by a discusser, the construction of high-speed turnpikes permits high car speeds. The power to overcome wind resistance varies as the cube of the speed. Thus designs reducing wind resistance can contribute appreciably to economy.

The evolution of the ideal transmission will provide maximum fuel economy without resort to the small or economy car with low performance and small size.

Can the Airlines Use the Helicopter?

F. N. Piasecki and L. S. Wigdortchik,
Piasecki Helicopter Corp.

This paper was presented at the SAE National Aeronautic Meeting, Los Angeles, Oct. 3, 1952.

IN this paper an effort is made to show how transport helicopters can be used by airlines for short-haul services to complement trunk-line airplane services. The objective is to emphasize the present need for airlines to incorporate the transport helicopter in their planning. Similarly, the authors stress the immediate need for the helicopter industry, airlines, and government agencies concerned, to coordinate and plan the development of short-haul services by helicopter.

Multiengined helicopters suitable for airline service can be available in 1956. However, there is an urgent need for practical operational research to establish airline requirements, as well as operating techniques, in order that needed features may be incorporated in the new helicopters. Further research is required to define the nature of ground facilities so that these may be made available in time for operations.

Through coordinated research into the problems of the airlines and the manufacturers, helicopter transport services can soon be a practical reality.

Transport Helicopter Development

Ten years have passed since Igor Sikorsky's first production helicopter, the R-4, began to appear in numbers. Through the years that have followed, we have witnessed the growth of the helicopter into an accepted vehicle capable of performing essential and diverse tasks. The extent of that development may be gaged from Fig. 1 where the change in speed, payload, and range performance

MULTIENGINED helicopters can be available in 1956 for regular airline service, according to these authors. They will be most in demand for short-haul transport, and will be able to operate directly into the center of business or industrial areas. Because of the small ground space or roof space required for a heliport the helicopter will not only solve many local and intercity traffic problems, but will serve also in extension flights to and from airports, thus obviating the present need for passenger limousines.

Moreover, the authors of this paper report that the helicopter has been developed into an accepted vehicle within a very short period. It was only ten years ago that the first production model – the Sikorsky R-4 – began to appear in large numbers.

During this decade the following achievements have been logged: (1) payload increased nearly 15 times; (2) range increased 6 times; (3) cruising speed doubled.

The Authors

F. N. PIASECKI (M '45) is chairman of the board of directors of the Piasecki Helipcopter Corp. He received his aeronautics degree from the Guggenheim School of Aeronautics, and in 1940 founded an engineering group which eventually became the Piasecki Helicopter Corp. He is a recipient of the Lawrence Sperry Award for outstanding contributions in the design and development of helicopters

L. S. WIGDORTCHIK is special projects engineer with the Piasecki Helicopter Corp. He graduated from the College of Aeronautical Engineering in London in 1939, and served with the Royal Air Force from 1940 to 1946. He joined Piasecki in 1951.

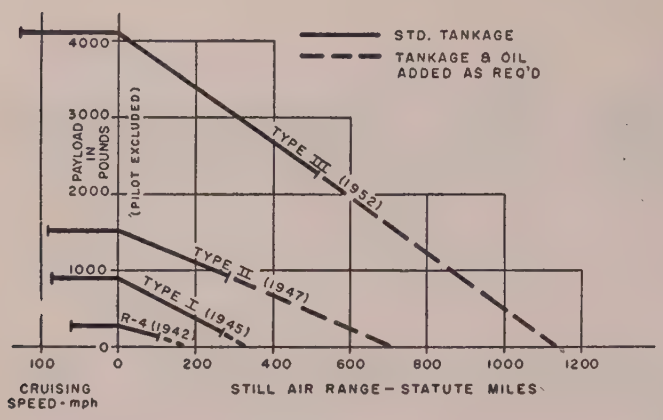


Fig. 1 – Ten years of helicopter development showing payload, range, and cruising speed

is emphasized. From the R-4 to the Type III of 1952, the following achievements may be logged: (1) Payload increased nearly 15 times; (2) range increased 6 times; and (3) cruising speed doubled.

The impetus of military requirements has led to these developments. Much has been learned about the art. Improvements in general performance have come about consistently as experience with successive designs has been gained. The supposed limitations to forward speed are now much more clearly understood, and recent helicopters have

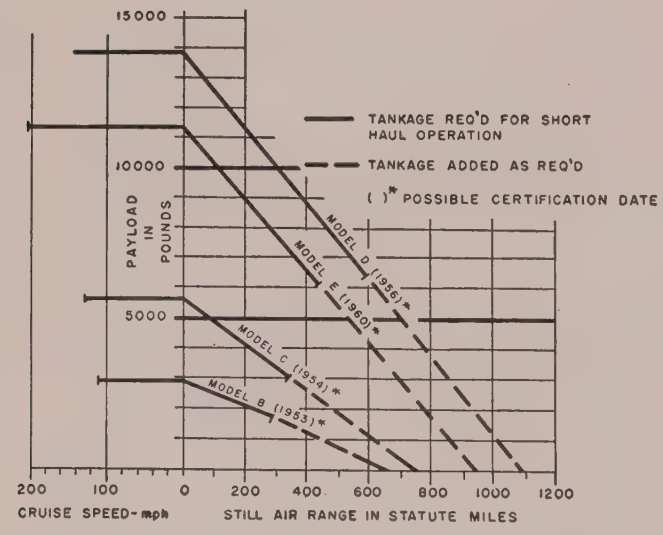


Fig. 2 – Performance of multiengined helicopters for airline operations, showing payload range and cruising speed

flown faster than was believed possible 10 years ago. Supposed limitations to the size of helicopters, although not taken so seriously by some, have proved equally fallacious, and prototype helicopters already exist with gross weights well in excess of 30,000 lb. Such helicopters will have payloads comparable to those of modern airliners. These developments are the natural outcome of a young and energetic industry, able to draw on the vast knowledge and material of the aircraft and mechanical industries.

On the operational side, helicopters have functioned on scheduled mail routes for five years. High mechanical reliabilities have been achieved, and the problem of maintenance has been reduced to a matter of daily routine. Passenger services have been operated in the British Isles with valuable results. Although it has been demonstrated that helicopters will be able to operate under all but the very worst conditions, and operational problems have received serious study, a considerable amount of this type of work remains to be done. As a result of accomplishments to date, British European Airways have declared that it shall be their policy to replace airplanes with helicopters on short-haul routes up to 300 miles as soon as suitable equipment becomes available.

At this stage, it is difficult to speculate further than 10 years as to future developments. Helicopters or other types of rotary-winged aircraft will be built to fly in excess of 200 mph. Helicopters larger than those prototypes existing today, with payloads of 10 tons, may make their appearance in this period. There appears to be no limit to the possible performance or nature of future rotary-winged aircraft, as learned papers have shown.[1, 2] As usual, development will be largely controlled by requirements which, in turn, will depend on the volume of planning and user experience which can be obtained and digested in these early days.

In order to view the future transportation role of the helicopter, an analysis has been made of the performance of representative multiengined transport helicopters which could become available in certificated form before 1960, in the light of present-day technical knowledge and practical experience. Fig. 2 shows the performance characteristics of three conventional medium-speed types, and one advanced high-speed type. These designs all meet the current CAR-6 (Civil Aeronautics Regulations, Sect. 6) requirements for partial-power performance. It is important to note that fuel reserves for transport helicopters will be much lower than those for airplanes in view of the helicopter's inherent flight advantages under IFR (Instrument Flight Rules), and the greater availability of alternate landing areas.

[1] See "New Applications of the Gyrodyne Principle," by J. A. A. Bennett, Technique et Science Aeronautiques, 6 rue Cimarosa, Paris XVI, France.
[2] See *Journal* of the Helicopter Association of Great Britain, Vol. 5, No. 4, 1952: "The Giant Helicopter," by O. L. L. Fitzwilliam.

It will be seen that type B, 12-place, can carry a 2000 lb payload over a still-air range of 200 miles, with a cruising speed of 110 mph. Type C, 20-place, will be able to carry 4100 lb over 200 miles at 125 mph.

Later on, type D, 52-place, which is a considerably larger machine, will be able to carry 11,200 lb over 200 miles at 140 mph. A helicopter of this type would have a passenger payload in excess of existing short-haul airplanes over short-haul blocks. By 1960, a smaller but faster advanced design, type E, 40-place, will be able to carry a payload of 9200 lb for 200 miles at 200 mph.

Short Haul Transportation

The main key to the helicopter's usefulness as a transport vehicle, is its ability to operate into the heart of a city. Like the train, it offers a "no change" service which, on the other hand, the airplane cannot. The fact that no time is lost in changing to other vehicles gives it a higher block speed over short ranges than the airplane, despite the latter's faster cruising speed. Similarly, its cruising speed and block speed are higher than those of the train. Therein lies its great advantage over both forms of transportation.

However, the helicopter must be considered in relation to the entire intercity transportation system, which takes in not only the airplane, but also the railroad, the bus, and the private automobile. Within this system, it is believed that helicopter passenger and freight transportation will divide into two major classes:

Local feeder services – Into and around large cities feeder services will develop to provide rapid transportation over areas where surface travel is congested and/or slow. Typical services will connect a major airport with the center and outlying suburbs of the city concerned. Others will link two major airports.

Intercity services – Short-haul services providing transportation between the centers of cities.

The first of these roles is already well understood and development is in hand.[3] The second, however, is still somewhat obscure. There will be some obvious aspects, however, in relation to the airplane, which the following study will seek to point out.

Potential Advantage of the Helicopter on Short-Haul Routes – The following aspects must be considered:

1. Speed advantages of the helicopter: Today airplane service suffers from two basic delaying factors, apart from those delays due to weather and mechanical defects. The first of these is surface travel time from city center to airport, averaging

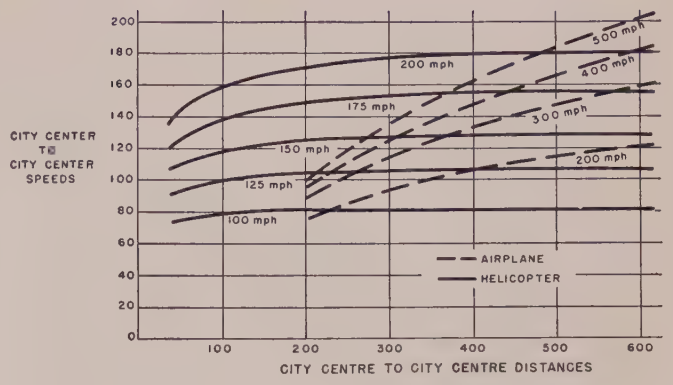


Fig. 3 – City-center to city-center speeds achieved by helicopter and airplane at various cruising speeds

35 min[4] at either end of a journey; the second is the time lost at airports between the ramp and take-off or touch-down. There is also the connection time at airports, for passengers must inevitably arrive ahead of the scheduled departure time, and similarly connect with taxis or limousines on arrival. This all works to lower the passenger's block speed between city centers, and it is only over long stages that a passenger begins to obtain the full advantage of the airplane's high cruising speed.

On the other hand, the helicopter can sidestep these delays almost completely. An analysis of the comparative city-center to city-center speeds over various distances, of both the helicopter and air-

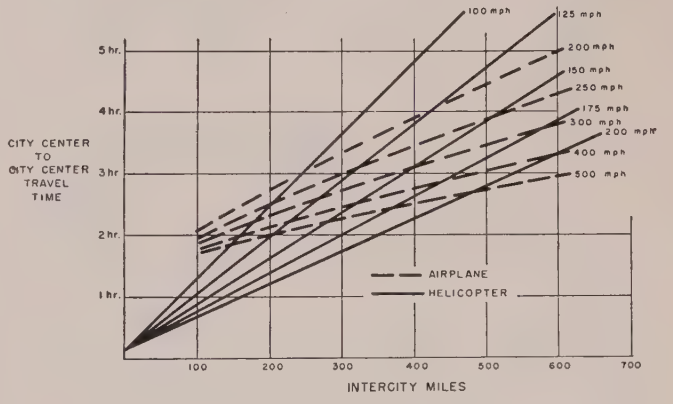


Fig. 4 – City-center to city-center travel time showing helicopter time versus airplane service

[3] Civil Aeronautics Board, Washington 25, D. C.: Exhibits, briefs, testimony, and other documents related to applications for helicopter mail and passenger certificates of convenience and necessity.

[4] "The Airport and Its Neighbors," by the President's Airport Commission May 16, 1952. U. S. Government Printing Office, Washington, D. C.

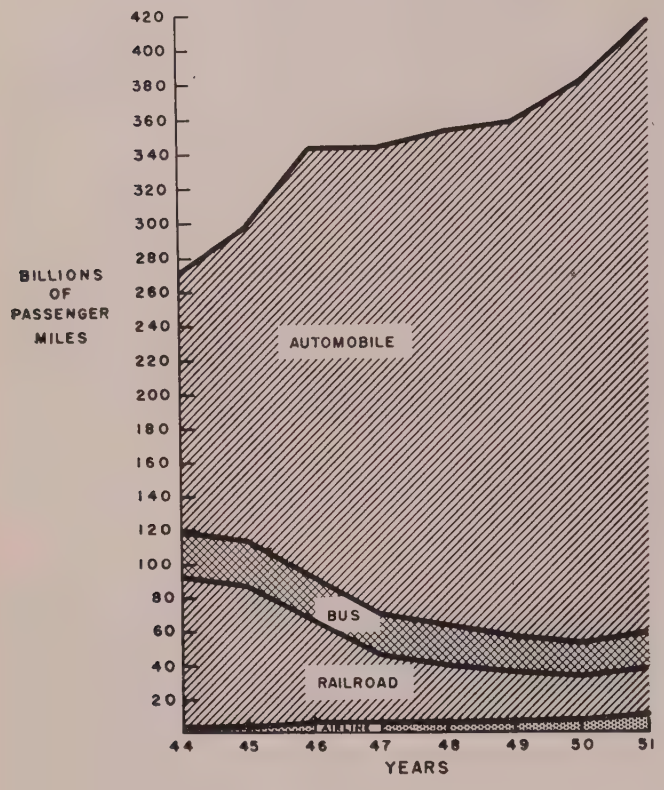


Fig. 5 – Total intercity travel in the United States from 1944-1951

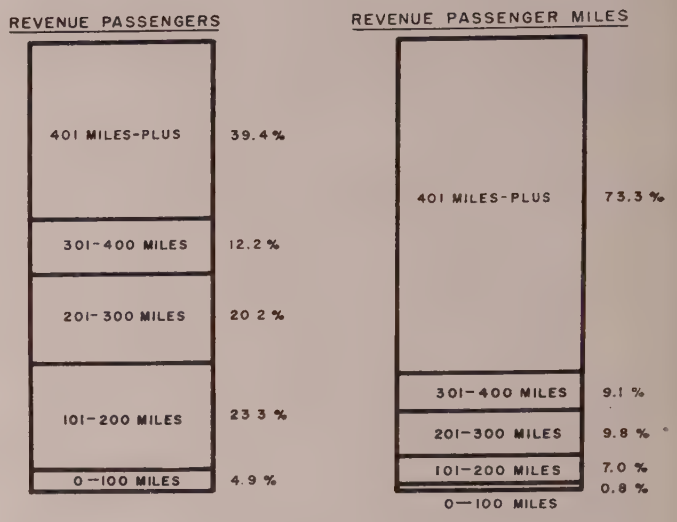


Fig. 6 – Breakdown of domestic United States intercity airline traffic by mileage groups

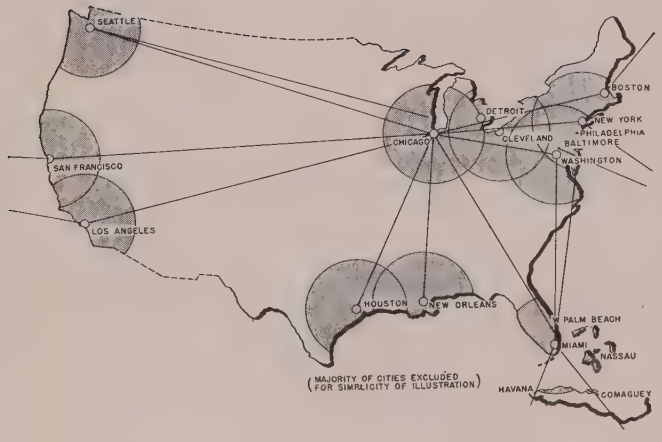


Fig. 7 – Principle of long-distance airline services supplemented by short-haul helicopter intercity services (helicopter operating through 250-mile radius from cities shown)

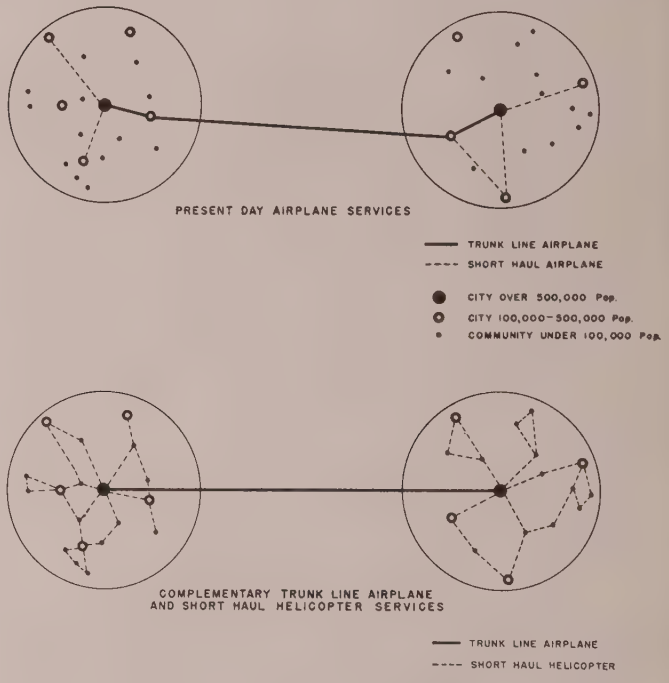


Fig. 8 – Increase in potential air travel market due to short-haul helicopter complementing trunk-line airplanes. To illustrate principle drawing shows helicopter operating through radius of 250 miles with major cities 1300 miles apart

plane, has resulted in the curves shown in Fig. 3. Here the helicopter and airplane are shown cruising at various speeds so that the effect of higher speeds of either vehicle can be assessed. These curves have been constructed on the best available knowledge, with due allowances for operational procedures. It will be seen that the service provided by the airplane with an en route cruising speed of 300 mph, only reaches that of the 125-mph helicopter at a distance of 260 miles. An increase in cruising speed of the airplane to 400 mph or 500 mph makes only a small difference to the competitive position below 300 miles. On the other hand, a much smaller increase in cruising speed of the helicopter from 125 mph to 150 mph – a development which is already practicable – makes a big difference in the competitive position. The advantage over the airplane is then extended from 260 miles to 365 miles. Further development of the helicopter to a cruising speed of 200 mph could extend the helicopter's speed advantage up to 500 or 600 miles.

Fig. 4 shows the picture in terms of time saved. The helicopter cruising at 125 mph can be scheduled over 200 miles at 1 hr and 55 min, or 33 min faster than an airplane cruising at 250 mph.

Table 1 shows how the actual city-center to city-center speeds of present-day airline services over routes up to 300 miles, can be lower than the average assumed in the curves in Fig. 4. In this data an allowance has been made for an average total connecting time of 20 min for both ends of the airplane journey. The average cruising speed for airplanes on these services is 240 to 250 mph, but it will be seen that the city-center to city-center speeds achieved are very low indeed, only averaging around 64 mph. Of course, many of the cities concerned are among the worst in respect to airport to city-center relationship, because they are so large. But for the same reason these are the cities which generate the most traffic. The helicopter will be able to offer a much faster service on such routes.

For these reasons, the helicopter, cruising at 150 mph, can provide the fastest transportation on a large proportion of the high-density intercity services up to block distances of 350 miles, with substantial savings of time to the passenger at distances below this figure. It will thereby bring a much larger potential air travel market to the airplane trunk routes, to the mutual benefit of each vehicle. To understand this better, it is interesting to briefly consider the traffic potential.

2. Total intercity traffic in the United States and the short-haul potential available to the helicopter: Fig. 5 shows how intercity travel has been divided, in its overall growth since 1944, between the airline, the railroad, the bus, and the private automobile. It is well known that the railroad has been losing to the airplane, but it is interesting to note that the automobile currently accounts for 85% of all intercity travel. In spite of traffic congestion and the automobile's low speed, its popularity is due to its convenience in providing local transport to the destination, as well as affording the traveler freedom from schedule. Economically, however, it is by no means cheaper travel for the individual than railroad coach or the bus. Thus the helicopter, in addition to replacing the airplane on short-haul services, can expect further traffic from the great potential of automobile travelers who now travel over routes where other transportation is either nonexistant or slow and infrequent.

Fig. 6 shows that 60.6% of airline passengers travel under 400 miles, and that this accounts for 26.7% of the revenue passenger miles flown. The helicopter can expect to absorb a high proportion of this business, as well as to stimulate heavier traffic from the 0 to 100-mile bracket, which now represents only 0.8% of the revenue passenger miles flown. This, together with the automobile traffic which it will attract, and the additional new business that an improved service will generate, clearly emphasizes that there already exists a very great traffic potential for the short-haul helicopter.

Just as the short-haul helicopter will offer advantages to the passenger, so it will in respect to express and freight. Shipping time will be much

Table 1 – Average Airline Speeds Between City Centers on Typical Routes

Route	Distance Between City Center, miles	Distance Between City Center and Airport, miles		No. of Daily Services	Airline Travel Time: Average Flight Time Ramp-to-Ramp, hr: min	Airline Travel Time: Average Limousine Driving Time, hr: min		Airline Travel Time: Average Waiting Time at Airports hr: min	Airline Travel Time: Total City-Center to City-Center Time, hr: min	Average Speed Between City Centers mph
Cleveland – Detroit	95	12	30	26	0: 48	0: 50	0: 60	0: 20	2: 58	53
Philadelphia – Washington	124	7	4½	16	0: 45	0: 30	0: 20	0: 20	1: 55	65
Chicago – Indianapolis	164	12	8	20	1: 10	0: 50	0: 30	0: 20	2: 50	58
New York – Boston	183	15	3	46	1: 08	0: 55	0: 20	0: 20	2: 43	67
Pittsburgh – Washington	186	14½	4½	16	1: 20	0: 50	0: 20	0: 20	2: 50	66
New York – Washington	209	15	4½	50	1: 17	0: 55	0: 20	0: 20	2: 52	73
Detroit – Chicago	238	30	12	27	1: 15	1: 00	0: 50	0: 20	3: 25	70
Los Angeles – San Francisco	340	14	13	33	1: 40	1: 00	1: 00	0: 20	4: 00	92

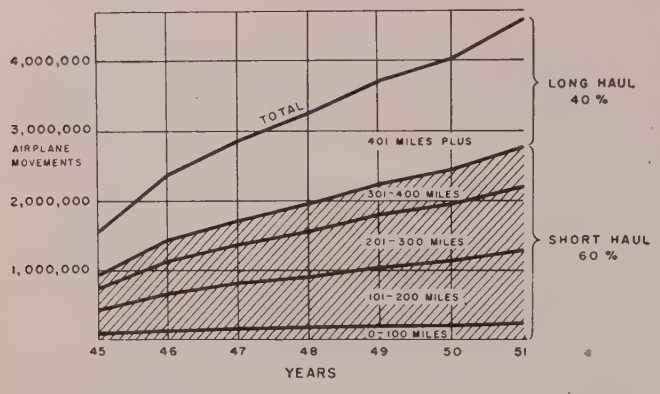


Fig. 9 – Total civil carrier units per year at CAA-controlled airports, and number of movements due to short-haul flights. Breakdown of stage lengths estimated before 1949

less, and thus not only will the helicopter generate new express and freight traffic for itself but it will also greatly stimulate airplane freight. A shipper in a community serviced by a helicopter will have the availability of air shipment to all helicopter-served communities around all of the major cities in the United States.

3. The principle of the complementary operation of airplanes and helicopters and its effect in expanding air transport demand: Fig. 7 shows how the helicopter might be used in conjunction with trunk-line airplane services in the United States. This has been drawn with the helicopter operating through a radius of only 250 miles from a few of the major centers. Even at this range, the helicopter can cover the densely populated areas of the country, and because of its ability to make stops without losing too much overall block speed, it can serve a far greater number of communities within an area than can the airplane. These communities will be able to afford the construction of a heliport where the cost of an airport would be prohibitive. Thus the helicopter will be able to make direct connection with trunk-line airplane services available to a greater number of people, while also affording them short-haul helicopter services. The effect of this is shown schematically in Fig. 8. The widespread and faster short-haul travel around large cities will make long distance air travel much more convenient and will stimulate further traffic for the airplane. Where previously a traveler from a small community had to take a train to the city, and a limousine out to the airport, with all of the connections and delays of surface travel in congested areas, he will now have a convenient helicopter to take him direct to the long distance airplane itself.

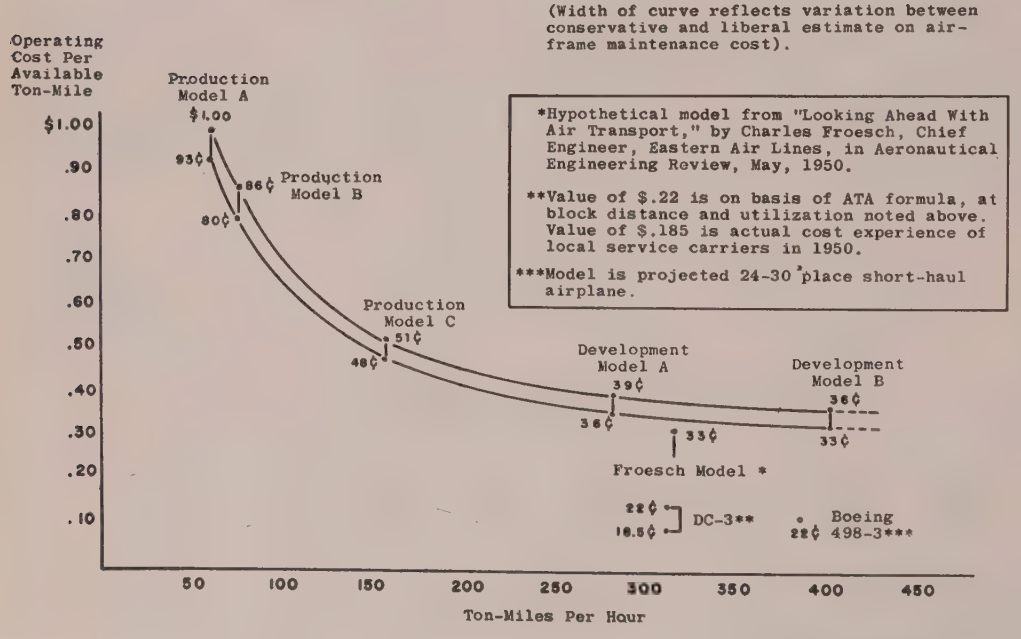


Fig. 10 – Estimated direct operating cost of large transport helicopters per available ton-mile versus work capacity in ton-miles per hour (85-mile block distance and 2750-hour annual utilization)

4. The growing congestion of traffic at airports and aggravation of this problem by the advent of jet airliners: Not only is the air and ramp space at airports now seriously congested, but public feeling concerning the danger and nuisance aspects of living close to airports is growing seriously.

The introduction of jet transports, with their limited fuel reserves, will further aggravate the problem. Longer runways and approaches will be needed. Noise problems will become even more acute. Stack time and holdoffs will have to be much shorter than they are at present, and will probably affect the critical peak movement rate under IFR conditions. The solution may be to build new airports to meet these problems. As stated by the President's Airport Commission,[4] these will have to be built further away from cities than previously. Airports will, therefore, become increasingly remote from cities. At the end of a fast jet flight the passenger will still be far from his destination.

On the other hand, the advent of short-haul helicopter services can go a long way to relieve the problem by taking the short-haul passenger flights away from airports. An indication of the percentage of short-haul movements of civil carriers at airports in recent years is given in Fig. 9. It will be seen that approximately 60% of the movements are associated with short-haul services under 400 miles.

Other Basic Factors in Establishing the Role of the Helicopter in Air Transport Operations – 1. Economics: Many estimates[5, 6, 7, 8] have been made of the operating costs of large transport helicopters. Since helicopters will be operated in a way very unlike the system used for airplanes, particularly in passenger service, such formulas as those of the Air Transport Association will not be strictly applicable. And so the only true comparison can come from a study of the replacement of airplanes by helicopters on a specific route system, when all of the direct and indirect costs can be accurately computed. An example of this type of approach is already available.[9] On the other hand, the Air Coordinating Committee's report[6] showed estimated direct operating costs (see Fig. 10) based on ATA formula, for various helicopters and airplanes. The lowest helicopter figure was higher than that for a comparable airplane – 33 cents per available ton mile. This was presented, however, with the rider that comparison of direct costs alone was misleading. Therefore, it is not intended to present herein an estimate of operating costs.

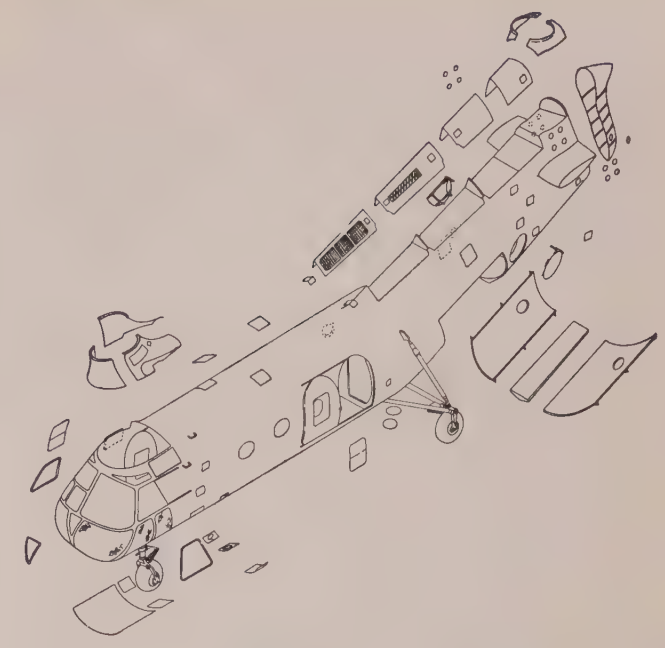


Fig. 11 – Maintenance accessibility panels of modern helicopter

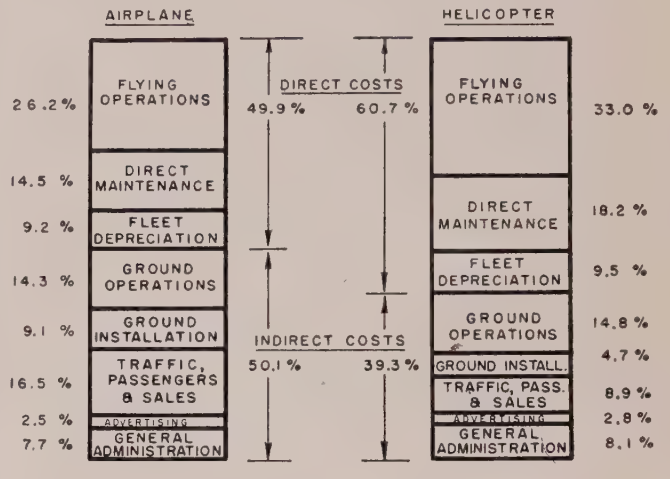


Fig. 12 – Estimated distribution of total airplane and helicopter operating costs per ton-mile over 100 to 200-mile blocks (total operating costs per ton-mile and annual productivity assumed equal)

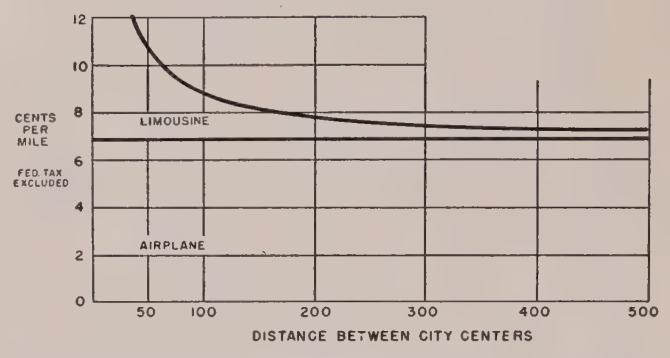


Fig. 13 – Average overall airplane passenger fare per mile between city centers (airplane at 6.8 cents per mile; limousine at $2.00 for journey at each airport)

[5] See *Aero Digest*, Vol. 52, June, 1946, pp. 74, 75, 117, 118: "Helicopter Transport Economy," by F. N. Piasecki.

[6] "Federal Policy Regarding the Development of Commercial Helicopters," Air Coordinating Committee, Washington, D. C., July 19, 1951.

[7] Ministry of Civil Aviation: First Report of the Interdepartmental Helicopter Committee, London, September, 1950.

[8] See *Journal* of the Helicopter Association of Great Britain, Vol. 3, No. 3, 1949: "Some Economics of the Helicopter, Present and Future," by L. S. Wigdortchik.

[9] "Economics of New Aircraft Selection," by Armin Baltonsweiler and W. Guldimann. Presented before SAE National Aeronautic Meeting, New York City, April 22, 1952.

Some of the basic factors affecting the economics of the helicopter in relation to the short-haul airplane are discussed.

a. Initial cost: The transport helicopter will probably cost slightly more per passenger seat than an airliner. Although the helicopter has a mechanical transmission and rotor system, it will have a much simpler structure, less complicated undercarriage and equipment systems, no flaps, or pressurization system.

b. Utilization: The helicopter should be capable of higher utilization than the short-haul airplane. Its time-on-ground will be much shorter, due to the lack of taxiing time. Its higher city-center to city-center speed and operation will induce short-haul travelers to travel over a wider period of the day, particularly under night discount fares.

c. Depreciation: Both the airplane and the helicopter face a similar rate of obsolescence.

d. Insurance: Because it is believed that the helicopter will be found inherently safer than the airplane, insurance rates should eventually be no higher. But in the beginning, it is likely that higher rates will be asked until a safety record has been built up.

e. Maintenance: When operated on regular schedules, the helicopter has proved to have a high mechanical reliability according to maintenance methods established for airplanes, although the early helicopters required high man-hours per flight hour. It has been found, however, that transmissions and rotor systems lend themselves well to airline replacement and overhaul techniques, the work-load being equivalent to that of one extra engine.[10] With provision for rapid replacement now incorporated in current production helicopters, and with fewer specialized systems in helicopters, the overall maintenance man-hours per hour of flight should be similar to that of airplanes. Replacement periods of components are being developed to 1000 hr. An indication of the accessibility of the modern helicopter is given in Fig. 11.

f. Productivity or earning capacity: Although the airplane ramp-to-ramp speed is much lower than its cruising speed, it is still higher than that of the helicopter on hauls of more than 100 miles. In this case, with an equivalent capacity payload, the airplane will be able to produce more capacity ton-miles per hour than the helicopter beyond this distance.

g. Traffic staff: The airplane pays a high price in traffic staff due to duplication at downtown centers and airports. There follows a duplication of booking facilities, administration, offices, and communications, which all combine to make high costs. The helicopter, on the other hand, offers the opportunity for a much simpler organization consolidated downtown. This is one of the cost headings which deserves major attention at an early stage of development to ensure the greatest possible saving.

h. Airport and handling charges: The overall costs of heliports will be much lower than those of airports, and so it is to be expected that heliport charges will be correspondingly less.

From the above it appears that the direct costs per ton mile of the helicopter will be higher than those of the airplane. On the other hand the indirect costs of the helicopter can certainly be lower than those of the airplane. Fig. 12 shows the estimated distribution of operating costs between the helicopter and the short-haul airplane.

However, this must be related to the possible fare structure for helicopter services. The average short-haul airplane basic fare rate is 6.8¢ per passenger mile before federal tax. The airport limousine charges average $2 per journey at each end. This produces an average overall fare rate between city centers as high as 8.8¢ per mile at 100-mile distances, as Fig. 13 shows. It is felt that conventional shaft-drive helicopters with capacity in excess of 40 seats can be operated within this band of costs of 6.8¢ to 8.8¢ per passenger mile.

2. Safety: Experience gained so far with scheduled services has shown freedom from mechanical failure to be dependent on the same factors as for airplanes. High standards of maintenance and adequate endurance testing of prototypes has resulted in the incidence of mechanical failures being kept down to that of airplanes. Mechanical failures have occurred to a larger extent in nonscheduled operations but this has been due to a lowering in maintenance standards under field operations, not representative of air transport operations.

Helicopter accidents due to flight error in scheduled operations have been rare even though operations have been conducted under very low mini-

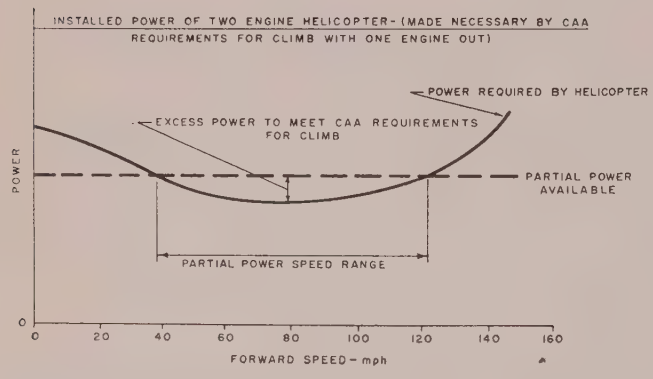


Fig. 14 – Partial-power flight speed range for conventional multiengined helicopter

[10] "Helicopters Applied to Local Air Service Operation – British Experience and Requirements," by N. E. Rowe. Presented at SAE National Aeronautic Meeting, New York City, April 17, 1951.

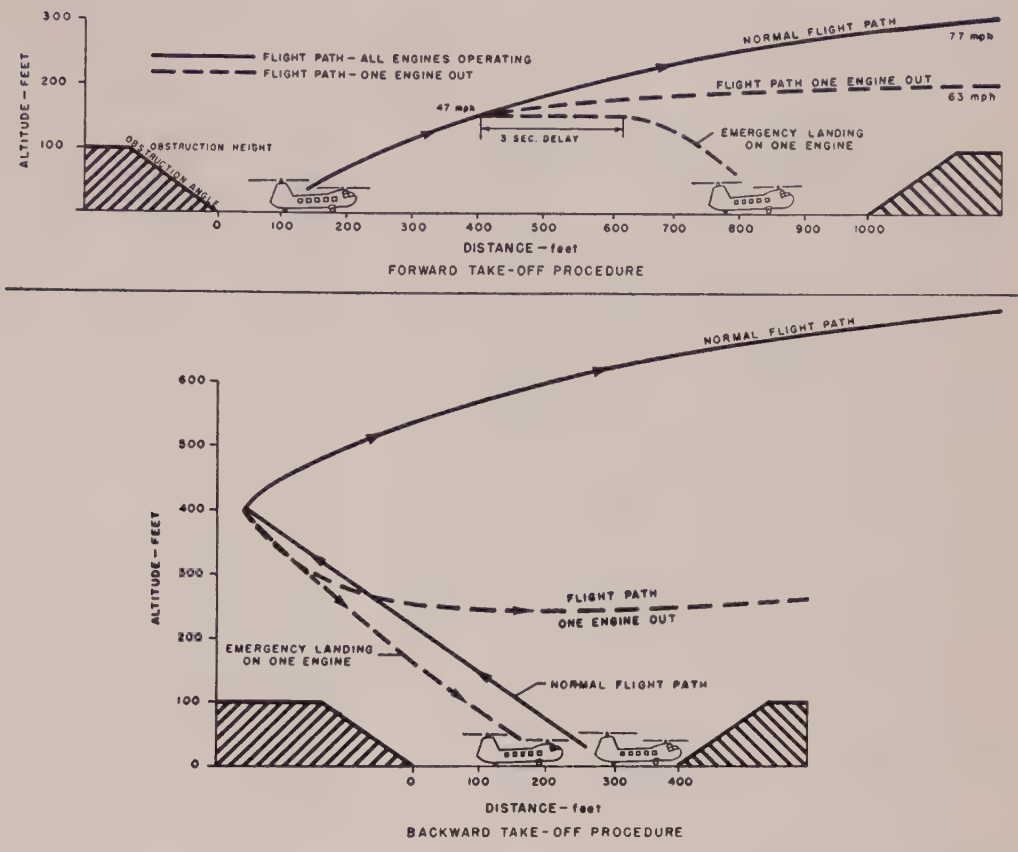


Fig. 15 – Safe take-off procedures and effect on heliport dimensions – partial-power performance to meet CAR-6 requirements

mums comparable, in certain cases, to current airline IFR minimums, even when the flights have been scheduled in built-up areas. The reason for the latter is the slow approach speed of the helicopter which gives the pilot more time in which to recognize obstructions and to take corrective action.

Analyses of airplane accidents[11] have shown how the chances of survival in an accident increase rapidly as the forward impact speed is reduced. And so the conclusion would be that the helicopter's low speed in poor weather or in approach and take-off will greatly lessen the chances of accidents – and in the case of accident will greatly reduce the chances of fatality. It is believed, therefore, that a higher safety record is possible with the helicopter than with the airplane.

It will be for the manufacturer and the Civil Aeronautics Administration to obtain the optimum mechanical reliability by correct endurance proving of prototypes, and by quality control in production; the operator and the CAA will ensure safe operating conditions and high maintenance standards.

3. Regularity and punctuality: The railroad, and to a lesser extent the bus, offer a much higher standard of regularity and punctuality than that yet achieved by the airplane. This standard is virtually 100% on the heavy traffic routes. The airplane has averaged a regularity of 98% of schedules completed in recent years but its standard of punctuality has left much to be desired, particularly in short haul where punctuality is of great importance to a passenger. If a passenger cannot arrange an important meeting closer than 30 min to his scheduled arrival time with certainty of punctuality, then the journey is effectively that much slower.

The helicopter's ability to fly slowly and to stop if necessary will enable it to operate to lower IFR minimums, and therefore eventually achieve a higher regularity than that of the airplane. Accordingly, delays due to poor weather should be less frequent. Apart from safety, airplane regularity and punctuality are one of the greatest deterrents to the remaining first-class travel market adopting

[11] See *Journal of the Royal Aeronautical Society*, Vol. 56, February, 1952, pp. 73-108: "Modern Trends in Civil Airworthiness Requirements," by Walter Tye.

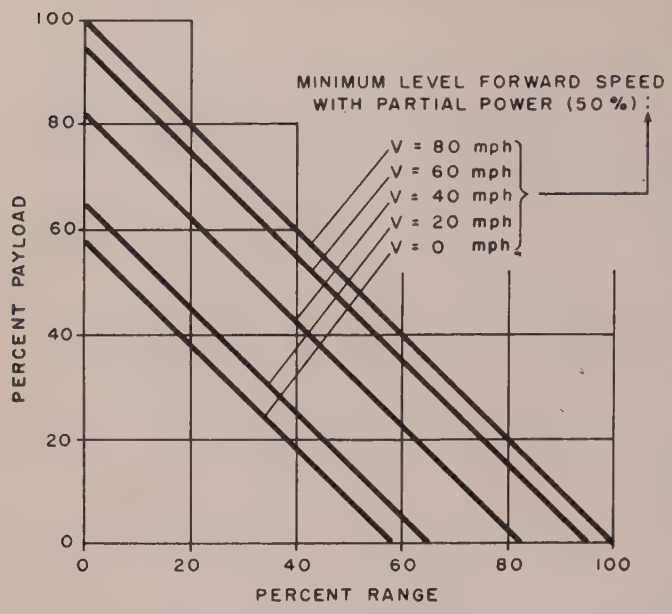


Fig. 16 – Performance penalty of partial-power flight requirements (partial power equals 50% installed power)

air travel. The helicopter affords an excellent opportunity to remedy this situation.

4. Convenience: Convenience plays an important part in a traveler's decision as to his mode of travel. In the general short distance journey, the "no change" facility of the train and bus make them more convenient than the airplane with its limousine connections and delays in relatively remote airports. Similarly the need for reservations makes air travel complicated; it is much more convenient to be able to decide about a journey on short notice, as is possible with railroad travel. The helicopter which will be boarded at the city center should be able to offer a convenience nearer that of the railroad than that of the airplane, and also should achieve reduction of reservation methods, as will be discussed later.

5. Public acceptance: There has been a good general public acceptance of the helicopter wherever it has operated and ready acceptance by passengers in those cases where passenger services have been established. Further, there has been evidence that the public feels that the helicopter is safer than the airplane, and that those who would not travel by airplane would, on the other hand, travel by helicopter. An analysis[12] of carefully derived passenger comments led to the conclusion that there was "a general acceptance of the helicopter as a practical and attractive vehicle."

Helicopter Operational Problems

Although transport helicopters will become available for airline service in the near future, there still remain a large number of problems to be solved. The fundamental performance requirements of the helicopter are already fairly well understood, but a host of operational aspects must soon be clarified if the corresponding requirements are to be embodied in this first generation of transport helicopters. Above all, the helicopter manufacturers need the assistance of the future operators and of the government bodies concerned, so that together they can ensure that the best knowledge goes into the helicopter and all of its associated facilities. Some of the problems are as follows:

The All-Weather Flight Problem – Helicopters are currently operating scheduled routes in the United States under contact conditions of 300-ft ceiling and ½-mile visibility – that is, well below airplane IFR minimums. Early experimental work[10] has already proved that there is no fundamental reason why the helicopter will not eventually be able to operate under lower IFR minimums than the airplane. But only the fringes of the practical problems have been tackled, and a considerable amount of work is required to bring all of the techniques and aids to a high standard of practicality. The Air Navigation and Development Board has recognized this, and they are proposing an experimental flying program to establish techniques and to provide equipment which will permit the all-weather operation of helicopters. Among the aspects which have to be clarified are:

Blind flight: Several operating agencies have shown that existing helicopters with various stability and control characteristics can be flown under completely blind conditions provided that there is adequate instrumentation. Limited IFR certification has been granted in two cases. However, the problem is one of endurance, for pilots tire easily, and it is not believed that such helicopters should be flown blind over one hour. Automatic pilots which solve this problem are already being installed in production helicopters, but emergency techniques for autopilot safe failure are not completely established. Minimum stability and control requirements need to be specified for manual flight. New altitude and low-speed indicators need to be developed for approach flight. Considerable experience needs to be amassed on such new equipment in order that blind flight may be developed into an everyday art.

Air traffic control rules and operating altitudes: Helicopters should be operated at low altitudes, 500 to 2000 ft above the terrain, where they do not encounter the highest winds. Helicopters should be confined to a separate traffic pattern en route, and also when connecting with airplanes at air-

[12] See *American Helicopter*, Vol. 22, May, 1951, pp. 11-14 and 19-20; Vol. 23, June, 1951, pp. 12-15 and 18-19; July, 1951, pp. 14-17 and 19-20: "Some Lessons of Liverpool-Cardiff Helicopter Passenger Service," by L. S. Wigdortchik.

ports. New traffic rules will have to be formulated to avoid en route flight delay, while techniques to avoid conflicting with airplane traffic patterns at airports must be proved.

Navigation aids: Present-day radio navigation systems in the United States are VHF with "line of sight" limitations, in that the signal cannot always be picked up over low terrain, behind hills, or in the lee of city buildings. Helicopters, operating at low altitudes, require an accurate indication of their track and direction, and so considerable modification of existing aids may be necessary. The accuracy of the navigation aid should be sufficient to bring the helicopter into the range of the approach aid.

Approach and take-off aids and ground lighting: Helicopters could be operated either with GCA (Ground Controlled Approach) or ILS (Instrument Landing System), but the cost would be prohibitive for the heliports which will develop in small towns. The slow approach speed of the helicopter gives the pilot more time for recognition and correction. Less elaborate equipment should be adequate, and this might take the form of an unmanned radio beacon which transmits an approach cone based on the optimum angle of approach. Approach direction can be superimposed. Ground-lighting aids must be provided for the change-over to visual reference. Cockpit layouts and view, commensurate with en route and approach flight requirements, differ considerably from that of the airplane. The search for visual reference, such as beacons, will have to be carried out through a greater vertical and azimuth angle.

Weather forecasting: In local operations, the helicopter operator will require knowledge of local, as well as area, weather conditions which are already available for the airplane. This information is vital in order to obtain the high regularity required of local service. New weather stations may be required, as well as new methods of assessing the weather – particularly the visibility at low altitudes.

Terminal minimums: From a full understanding of the foregoing problems will evolve the definition of practical terminal minimums. It is believed that certain terminal minimums may be developed to a 200-ft ceiling, and a 500-ft horizontal visibility. Such developments will require much operational flying under all conditions to ensure their practicality. Completely blind landings may become possible for the helicopter sooner than for the airplane, because of its slower approach and descent speeds.

Ground equipment: Because of the number of heliports which will be required, and the need to keep indirect operating costs to the minimum, heliports should be unmanned in respect to operations and control. All aids should be automatic. Area control could provide the necessary traffic coordination.

Emergency landing equipment and procedures: The helicopter should carry sufficient airborne lighting and emergency flares to enable it to make an en route emergency landing. The ability to do this will provide a great advantage over the airplane. Emergency landing due to partial-power failure will be discussed in detail later.

Partial-Power Safety and Take-Off Area Require-

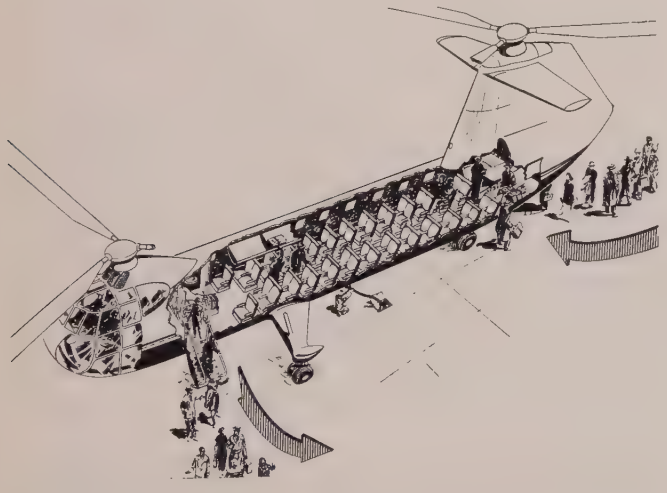


Fig. 17 – Fast traffic and ground-handling technique for large intercity Piasecki H-16 helicopter, designed for one-way passenger flow through rear entry and front exit

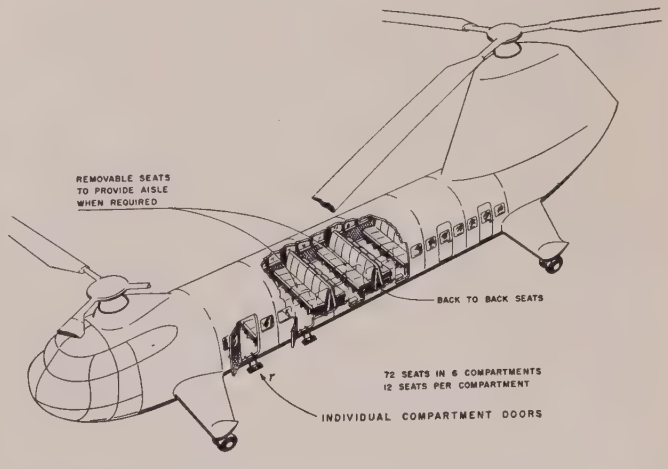


Fig. 18 – High-density seating for local feeder services; patterned after European railroad car, with individual compartment doors

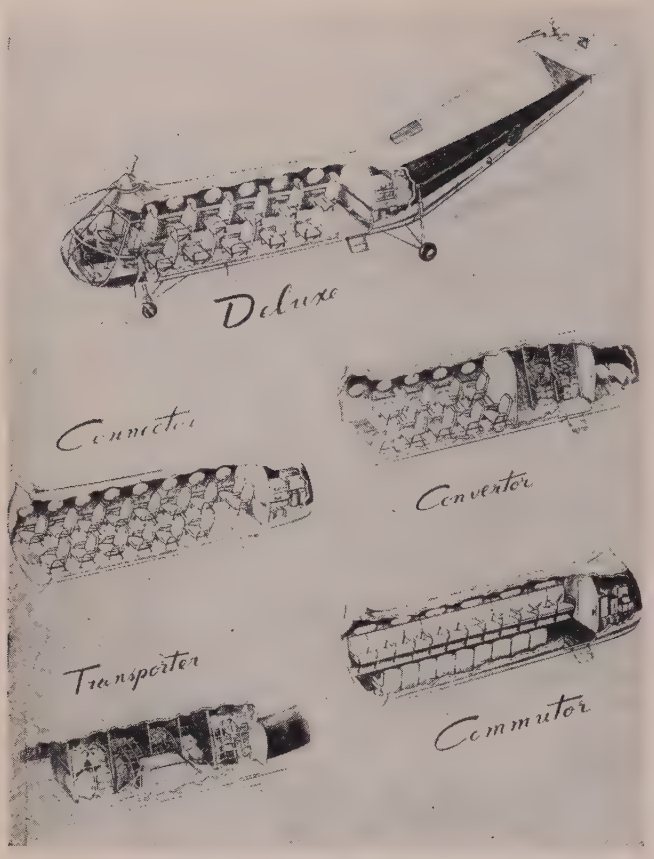


Fig. 19 – The PD-22 helicopter, showing different seating plans and luggage facilities

ments – These two factors are interlinked for the helicopter in the same way as they are for the airplane, except that the helicopter's speed range extends down to zero forward speed, at which point the power required to maintain height rises very steeply, see Fig. 14. The full power of all engines amply covers all flight power requirements. But when the power is halved, as in the case of a twin-engined helicopter with one engine out, the helicopter can only maintain altitude through a much reduced speed range. If an engine fails before the minimum level flight speed has been reached on take-off, then the helicopter has to make a landing unless it can exceed its minimum level flight speed on its descent. If the total power of a twin-engined helicopter were double its hovering power, then this condition could not arise, but the penalty in weight would be high. Realizing this the proposed CAA requirement[13], which is considered to be rigorous, requires a climb performance of 1 in 20 at best forward climb speed with one engine out (between 250 and 400 fpm depending on the speed This sets a compromise between payload performance and safety, but may possibly require that landing areas larger than originally anticipated be available to cover the critical take-off case. Fig. 14 shows how this requirement sets the minimum level flight speed, thus controlling the take-off procedure.

Fig. 15, based on Whitby,[14] shows how take-off condition can be covered with helicopters meeting the CAA requirements. If the worst case, a forward take-off, was considered to be necessary, this would entail a clear area 1000 ft long to be available to cover that stage of the take-off when a landing would be inevitable in case of engine failure. This would be a costly area to provide in the center of a city. The preferred technique is a backward take-off, Fig. 15, when a helicopter can return substantially down its take-off path in event of engine failure, thus permitting the use of an area 400 ft in length, as originally anticipated for large helicopters. This take-off technique has been widely used for other reasons in previous operations, but has to be thoroughly proved as an acceptable procedure against engine failure under all conditions. It will be appreciated that a delicate compromise has to be struck between partial-power flight requirements and the economic performance of the helicopter, on the one hand, and the economics of heliports on the other.

The extent of payload penalty, for a conventional helicopter, in connection with partial-power safety, can be appreciated in Fig. 16, where the effect of providing for lower minimum safe flight speeds is plotted against the payload and range of a hypothetical but conventional mechanical-drive helicopter. This helicopter at a minimum safe level speed, with the partial power required for 80 mph, has a good vertical climb performance with full power. It will be seen that considerable penalties will arise if the minimum safe level speed has to be reduced below 40 mph, in order to permit forward take-off to be carried out from small areas.

It is of the utmost importance that this problem be studied under practical operating conditions in order to establish a workable compromise at the earliest possible time. Until this question is resolved, helicopter and heliport development cannot be based on a firm foundation.

Arrangement of Passenger Accommodation – Helicopter short-haul transportation poses fresh problems in passenger accommodation. Passengers traveling between 10 and 100 miles may accept lower standards than those traveling between 100 and 400 miles. Over very short haul it is important to fit the maximum number of seats into the cabin to fully utilize the available payload. And yet the operator may want day to day flexibility of his fleet over all block distances. What then are the minimum standards compatible with these requirements?

Standards laid down[15] call for nonreclining seats 20 in. wide between armrest centers, and pitched at

[13] "Rotorcraft Airworthiness," Civil Aeronautics Board, Civil Air Regulations – Part 6, January 15, 1951.

[14] See *Journal of the Royal Aeronautical Society*, Vol. 56, January, 1951, pp. 27-42: "Some Operational Problems of Public Transport Helicopters," by R. H. Whitby.

36 in. Although close to the minimum acceptable for 100 to 400-mile services, these are well above the standard considered acceptable for 10 to 100-mile services. Seating to this standard for 50 passengers is shown in Fig. 17. If armrests could be dispensed with and a removable seat used on 10 to 100-mile services, then the arrangement shown for the Piasecki H-16 in Fig. 18 would provide 72 seats in a cabin of the same size, although some weight penalty might be incurred due to the greater number of doors. This European rail coach concept could well speed up passenger movement. Fig. 19 shows the effect of various seating standards in the smaller Piasecki PD-22 helicopter.[16]

An alternative method of providing flexibility of seating capacity is the capsule concept shown in Fig. 20. This type of arrangement, although entailing some weight penalty, will provide extra seating or freight capacity as required. In the latter instance, a freight capsule can speed up handling at turn-rounds, thus increasing potential helicopter utilization as shown in Fig. 21.

It is assumed that toilet facilities will be required, and possibly provision will be made for serving light refreshments. Fig. 17 shows an arrangement of these facilities, and also a portion of baggage storage space for those passengers who wish to carry their own baggage.

Since cabin arrangement and seating are an important factor in design, it is clearly vital that these points be resolved at the earliest possible time.

Traffic and Ground-Handling Techniques – Because the helicopter is a flexible short-haul vehicle, it is vital that the time on ground at stops be as short as possible. Not only must overall block speeds be maintained, but also utilization must be high for the reason expounded previously. Thus high ratios of flight time to ground transit time must be obtained.

Therefore, the helicopter passenger will have to be self-sufficient to a degree approaching that of the railroad passenger. It is essential that helicopter service permit some relaxation of the tedious and costly system of reservations currently used with airplanes. This may be achieved by a partial reservation system on a surcharge basis, and an "on the spot prior to flight" mechanical reservation system at heliports. The fact that the point of departure is in the city center will assist in such relaxation of reservation systems acceptable to the public, since switching to other ground transports can be easily accomplished. In some cases frequency of service will bring about a reduced need for a positive reservation closely approaching city bus or even intercity bus and railroad service. Passengers must be encouraged to carry their own baggage. Traffic documentation must be kept to the minimum, and current regulations requiring that copies of passenger lists be left at each stop should not apply to helicopters. A member of the crew should be responsible for all loading and dispatching duties, so that the requirement for traffic staff at intermediate stops is minimized, if not entirely eliminated.

---

[15] "Outline Specification of Requirements for a Large Helicopter," British European Airways, October, 1951.

[16] "Economic and Design Aspects of Large Transport Helicopter," by Harry S. Pack. Presented at 1950 Annual Forum of American Helicopter Society.

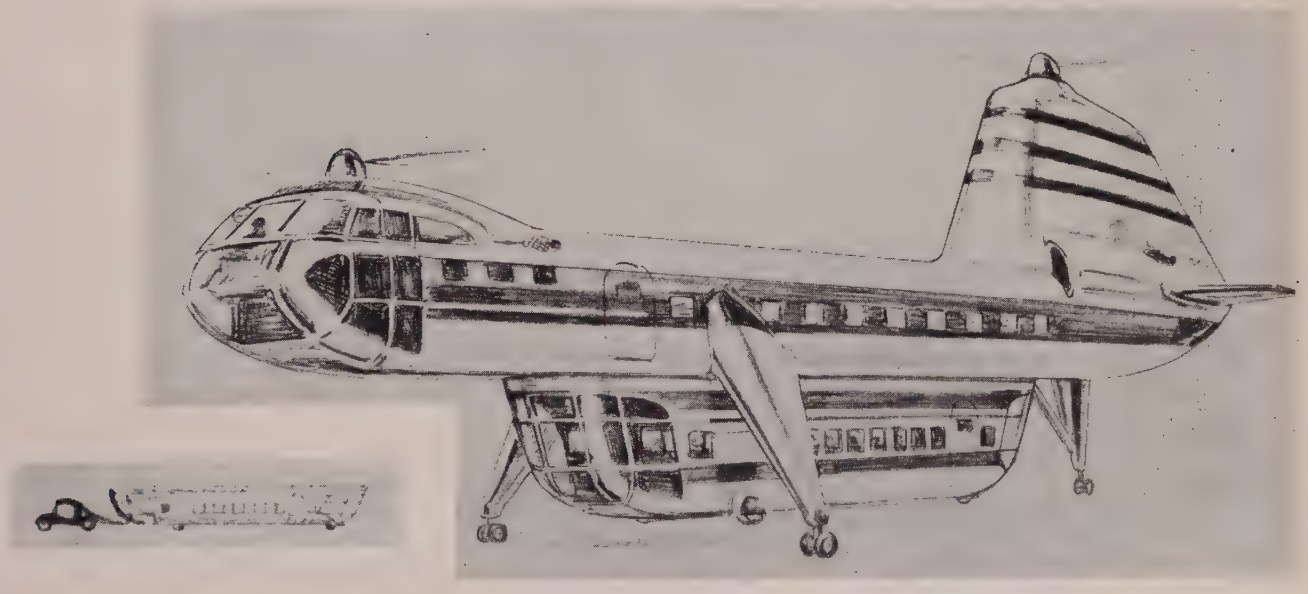


Fig. 20 – Airline connection line version of Piasecki XH-16, showing use of capsule concept to provide extra seating or freight transport

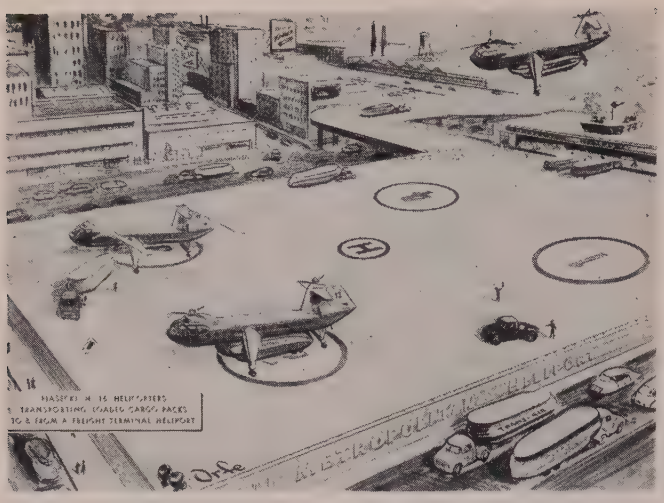


Fig. 21 – Design for a metropolitan heliport showing the use of the prepacked freight capsules

No ground-handling equipment should be required, doors should have built-in steps, refueling points should be accessible immediately under the fuselage, built-in footholds and handholds should enable quick checks to be taken of the rotor system, and engines should be immediately accessible without the need of stands. Baggage holds should be accessible both from inside and outside the fuselage, while the external sills of the holds should be no higher than chest level. Two main doors should be provided to the cabin, so that a one-way flow of disembarking and embarking passengers may be maintained at stops. Passengers should only be able to gain access to the ramp through an unmanned turnstile operated by tokens handed out with tickets.

Fig. 17 shows some of the above principles. Passengers enter at the rear steps and immediately deposit their baggage in holds on either side of the steps, or take their baggage to deep racks above their seats. On arrival, they leave by the forward door, either taking their baggage with them or collecting their baggage from the outside hatches to the rear baggage holds. The crew member responsible for loading stands at the head of the rear steps checking the embarking passengers as they come up the steps. This technique would have to be carefully studied for a 50-seat helicopter of this type, but it is equally obvious that studies of these problems are required now to give the helicopter manufacturer and heliport architect the important requirements.

The Integration of Airplane and Helicopter Operations – The essential link between airplane trunk and helicopter short-haul services will take place at airports. As already stressed, it will be vital for their traffic patterns to be independent of each other, so that neither operation imposes delaying factors upon the other.

The position of the helicopter landing area at major airports is of serious concern. The disembarking helicopter passengers must be within walking distance of the airline check-in counter if the helicopter is to offer the same connection convenience as the limousine, and if the whole operation is to be smooth and simple. If the landing area is even a half mile from the airline check-in counter, the time delay incurred by the use of airport buses would nullify the advantages of the helicopter. This, therefore, implies that the helicopter will have to operate from the very heart of the airport, and that the associated problems of control and flight procedure will have to be resolved. Fig. 22 shows one of the proposals for London Airport which would meet this requirement. Experience gained thus far has shown that this flight problem is by no means as acute as it might appear, mainly because of the low-operating altitude of the helicopter and the ability to maintain an accurate track. The ground facilities would have to be planned on the peak IFR movement rate.

In Fig. 23 a proposed pattern is shown where the helicopters would cross the airplane runways at 90 deg, at an altitude of not less than 500 ft. Clearance to cross the runway would be given by the airport control when the runway was actually free from airplanes taking off or landing, but this would not impose appreciable delay since the helicopter could hover while awaiting permission. The illustration shows the worst case, when the helicopter passes under the traffic pattern prior to crossing the runway. The airplane would be restricted to altitudes well above the helicopter's operating altitudes in all positions other than on the runway approach and take-off zone.

Noise – Operating helicopters into cities will introduce new problems concerning noise. It is considered that engine noise may be the critical factor, particularly when shaft turbines are used. Some degree of silencing will be mandatory.

Acceptable levels of noise must be defined in conjunction with tests in specific locations. The situation may not be so acute if heliports are situated in business centers or industrial areas, unlike airports, which are generally located in new suburban residential areas. However, when the helicopter operates into airports it will be subject to problems similar to those of the airplane.

Some Problems of Heliport Design – Major considerations in heliport design are position, traffic, operating techniques, size, construction, and especially cost, which must be considered in relation to all these points.

Position is controlled by the need to possess good communication with all of the traffic-generating area which is to be served. Heliports will be subject to movement restrictions under IFR. Estimates based on present knowledge have shown that a peak of 17 movements per hr might be achieved under

IFR, which in turn could provide for a maximum of 850 passengers per hr when using 50-passenger helicopters.[14] This capacity would, with the traffic requirements, determine the number of heliports for any one area. The positioning of adjacent heliports entails complete study of the IFR airspace requirements and operating techniques in order to avoid confliction of traffic patterns. Other aspects of operating techniques affecting heliport design, are traffic handling and turn-round technique discussed previously. Surrounding obstructions will have to be kept below a 35 deg angle generated from the periphery of the area or heliport. Locations will be limited accordingly.

It will be best to use ground sites initially until further knowledge is gained, but surrounding obstruction and desired position may leave no alternative but a roof-top site. The latter will require consideration of concentrated loadings and other special construction features. Both types of sites will have to be seriously studied in respect to turbulence and special windbreaks, or other devices to smooth airflow throughout the area.

The heliport size is largely controlled by partial-power emergency landing procedures. The preferred technique of backward take-off will permit the use of a heliport of minimum size, schematically described in Fig. 24. Here a clear into-wind area 400 ft long and 160 ft wide, allows sufficient emergency landing space for a typical 50-passenger helicopter. Parking area for seven such helicopters is provided, five of them with blades folded.

Standardization of Local Community Flight Ordinances – Since the helicopter will serve a greater number of communities than present airplanes, local ordinances concerning flight over built-up areas must be carefully studied with a view to establishing standardization and precedents for future operations. Otherwise public sentiment is likely to be aroused in the same way as it has been in the vicinity of airports. Care must be taken to ensure the best possible flight lanes over built-up areas, from the aspect of nuisance factor and safety. Early action is required to secure the best compromise in order to avoid local legislation which could cripple helicopter services.

Establishment of Transport Helicopter Design Requirements

The helicopter manufacturer labors under the lack of a specification for commercial transport helicopters. The multiplicity of operational aspects which may be encountered by a passenger helicopter have not yet been experienced by a sufficient number of carriers to bring forth a coordinated specification. Thus the manufacturer has had to rely on his own studies of the problem.

Enough is already known about the transport helicopter to enable the operators to study its potential application. Initially, route studies have to be made by the operators in order to arrive at the general type and performance of the helicopter required. Thought must be given subsequently to details of the operational aspects, some of which have been previously discussed, and certain questions must be answered before a specification can be written. From the standpoint of the transportation service to be provided by the airlines, questions of the following type must be resolved:

(1) Over what type of routes will the helicopter be operated – length, traffic, frequency, terrain clearance?

(2) What economic performance will be required – direct cost per ton mile, available utilization?

(3) What operational procedures and allowances will be employed – cruising powers, winds, altitudes, fuel reserves, crew, navigational methods?

(4) What passenger handling and loading methods will be employed – techniques, ticketing reservation, documentation, supervision, alternate seating arrangements, and freight holds?

(5) What other operational equipment will be required – fire-fighting, emergency flares, running lights, landing lights?

(6) What turn-round technique will be employed – ground handling, taxiing, refueling, blade-folding, cabin cleaning, gate system?

From the answers to the above, the following type of questions must be further resolved in order to provide the basis for a transport helicopter specification.

(1) What will be the performance requirements – cruising speed, range, payload, climb, capacity?

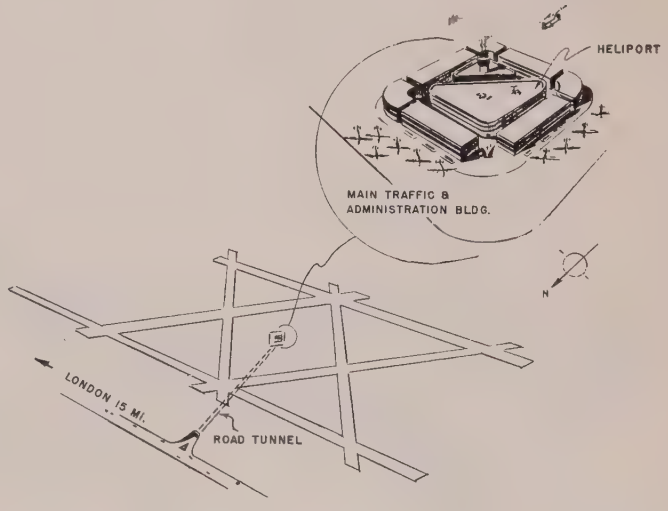


Fig. 22 – Proposed heliport for London airport consists of triangular roof of central car-parking building containing five stories of parking space, enclosed by 550-ft sides. Passengers will descend by elevator to ground level where they will be within 440 yd of all airplane ramps. Proposal is awaiting approval

(2) What safety criteria will be required – partial-power performance, de-icing, flight-handling qualities?

(3) What take-off and landing techniques will be employed – angle of approach, speed, blind-approach aids, flight aids, instruments?

(4) What cockpit layout will be required – position of crew, controls, view, comfort, lighting?

(5) What types of seating, baggage, galley, toilet, will be required – dimensions, arrangement, type?

(6) What will be the comfort standards – external noise vibration, air-conditioning, passenger view, internal noise?

(7) What maintenance characteristics are required – maintenance system, component change time, choice of powerplant?

There is no doubt that the best way to study these problems is through actual experience gained from the operation of experimental services. Although much has already been achieved by the pioneering services in Los Angeles, Chicago, and elsewhere, a great amount of additional information needs to be obtained. Several years of government-supported helicopter operations has enabled British European Airways to prepare such a detailed specification for a transport helicopter, which has proved most informative.[15] Experimental services, carrying freight initially, can be operated over suitable routes in such a way as to simulate and anticipate all of the operating problems. A cadre of trained personnel can be developed who will become cognizant of such problems. Ground equipment and heliports can be developed and tested in representative conditions. Specific aspects can be studied in conjunction with all the bodies concerned.

Program of Development

This paper has sought to further stimulate the already growing awareness of the transport helicopter's potential, and to point out the problems of its operation and associated facilities which need to be solved in time for the transport helicopter now being developed. The following phases of work will have to be undertaken in an overlapping sequence:

Phase A – Further experimental flight operations to determine operational methods, facilities, and requirements.

Phase B – Design and construction of passenger and freight heliports, both in cities and at airports, with ground facilities and operational facilities.

Phase C – Experimental passenger services to prove route systems and techniques.

Phase D – Introduction of regular passenger services on short-haul routes.

Conclusions

(A) Large transport helicopters capable of handling a large proportion of airline traffic can be available for airline operation after 1956.

(B) The transport helicopter of this decade will be able to offer a service to the intercity traveler

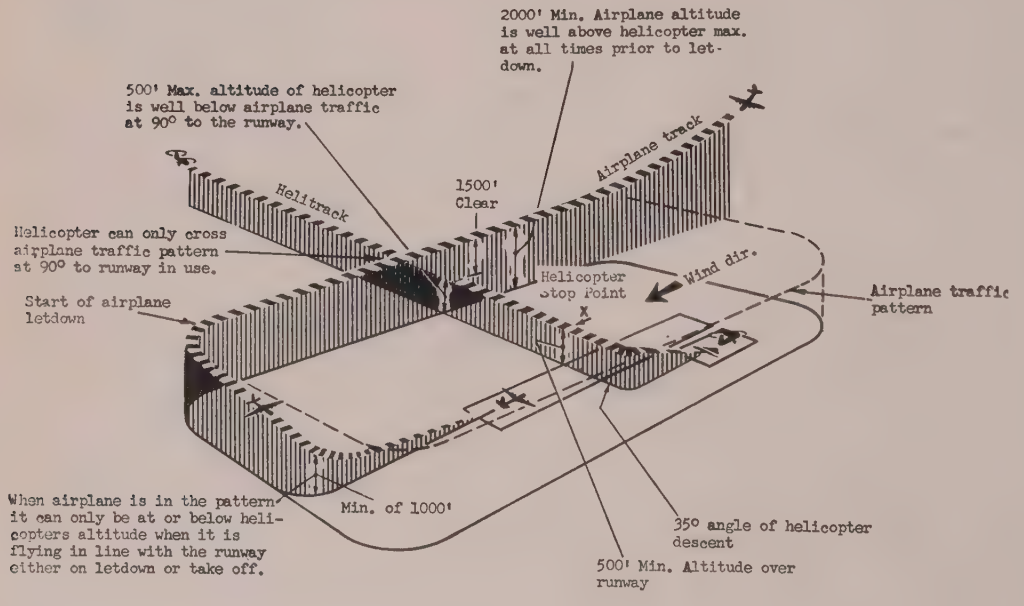


Fig. 23 – Helicopter and airplane flight traffic patterns for major airports, with worst possible landing case illustrated. Take-off procedure is reverse of landing procedure

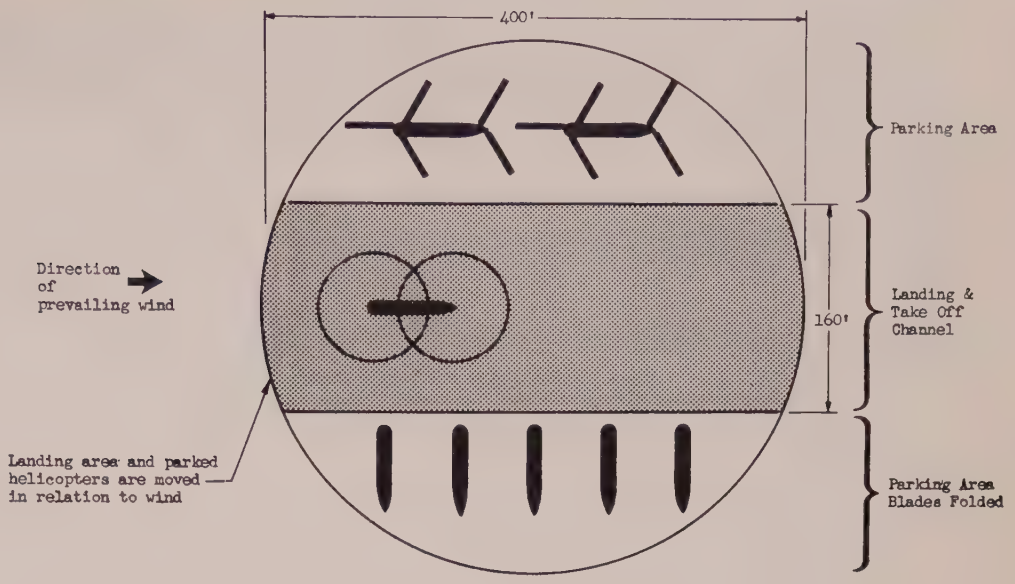


Fig. 24 – Layout of heliport flight area requirements

which will be superior to that currently offered by the airplane, over block distances up to 350 miles.

(C) The short-haul transport helicopter in complementary operation with the trunk-line airplane will further accelerate the growth of all air traffic by bringing the speed of air travel to a much greater number of communities and people.

(D) Widespread development of short-haul helicopter services will enable many of the serious problems of airport and airplane operation to be greatly alleviated.

(E) In view of the above, the airlines should include the transport helicopter in their present planning.

(F) Cooperation between the helicopter industry, the airlines, and government and city authorities, is needed to provide operational methods, facilities, and requirements, to meet the availability of the first large transport helicopters.

Transportation can be likened to a tool with which man broadens and expands his activities. Sociological changes have been implemented by the availability of new vehicles and the particular facilities they provide. First, new vehicles have been used to replace older ones, but later they have created their own traffic, and in so doing they have changed the habits of men. This is the process which the helicopter is likely to follow – in the beginning it will augment and then replace airplane, railroad, and bus traffic, on certain suitable short-haul routes. Later, it will create new relationships between communities, and will thereby increase the overall traffic in the country.

Acknowledgment

To Monroe Brown for his valuable criticism in the preparation of this paper.

To Graham Aldrich of the Air Transport Association for the valuable data furnished.

And to those helicopter lines whose pioneering work in scheduled operations has done so much to further helicopter transportation:

1947 – Los Angeles Airways
1948 – British European Airways
1949 – Helicopter Air Services, Inc.
1949 – Sabena Airlines

Additional References

1. "Etude de Transport par Helicopters," by Paul Codos. Technique et Science Aeronautiques, 6 rue Cimarosa, Paris XVI, France.

2. "Military Aspects of the Transport Helicopter," by F. N. Piasecki, Journal of the Helicopter Association of Great Britain, 1952.

3. "Helicopters Applied to Local Air Service Operation – American Plan," by C. M. Belinn. Presented at SAE National Aeronautic Meeting, New York City, April 17, 1951.

4. "The Status of the Helicopter in Relation to Future Development of Air Transportation and Airport Planning," Aircraft Industries Association, April, 1952.

5. "The Impact of Turbines on Airlines," by E. A. Driessen. Presented at SAE National Aeronautic Meeting, New York City, April 22, 1952.

Growth and

The Author

B. B. BACHMAN (LM '10), vice-president of engineering, Autocar Co., was president of SAE in 1922. He joined Autocar in 1905 and has held his present position since 1929. He has contributed consistently and actively to almost every phase of SAE effort, both technical and administrative. He was the first president of the Coordinating Research Council and was chairman of the SAE War Activities Council in World War II. He has been SAE treasurer since 1944, having previously been chairman of the Finance Committee.

TRANSPORTATION is a service that is vitally necessary, and of all the forms of transportation which serve us daily, the kind which moves over the highways is most important to us and serves us most intimately.

Birth of Mechanical Transport – Until early in the nineteenth century, the winds on the water and the beast of burden on land were the sole aids to man's own muscles in getting from place to place. Then, after the Napoleonic wars ended at Waterloo, the steam engine was harnessed to ships and vehicles. The result was a strange paradox. Here on the North American continent, and particularly in the United States, the railroad made it possible to move out of the coastal areas, into and across the plains, and finally to span the continent. This represented a great expansive movement, dispersing not only our own people but also attracting immigration from abroad.

On the other hand, it was a powerful factor in promoting large centers of population and industry. This was due to the need for industry to be located so as to have siding facilities to bring materials in and take finished products away. At the same time, the workmen had to live sufficiently near the shop so that they could get there by horsecars or by walking, within a reasonable part of the day.

The elevated railroad, electric cars, and subways, in turn afforded progressive means to extend the distance to which it was possible to remove from

GROWTH of mechanical transport since nineteenth-century days, when the steam engine was attached to ships and vehicles, is traced in this historical review. It was appropriately presented in observance of engineering achievements which have taken place during the past hundred years. The author records that development of the internal-combustion engine, replacing steam, led to the appearance of the automobile at the turn of the century.

The horseless carriage increased travel speed, setting up immediate demands for more comfortable travel. Good roads, improved tires, new engine and body designs have followed without interruption for more than 50 years, with the supply of applied ideas never approaching public demands for still greater improvements.

The author summarizes the situation as it exists today, giving many pertinent facts regarding the growth of the trucking industry, its relation to highway facilities and passenger-car travel, together with possibilities for future developments along better lines.

Status of Highway Transport

B. B. Bachman, Autocar Co.

This paper was presented at the Centennial of Engineering, Chicago, Sept. 9, 1952.

the shop or office. The first move toward individual transportation was the bicycle which, even though it used the muscles of the passenger for propulsion, nevertheless, was relatively cheap, and extended the limits of travel materially.

Then, just at the turn of the century, the internal-combustion engine reached the stage of development which made its use in the automobile possible.[1] Now, the emancipation of the individual from dependence on mass transportation became a possibility. Instead of having to live on or near a railroad or car line, a street wide enough to permit the passage of a car suffices to put the individual in touch with all the world. This has resulted in exploding the big cities into the suburbs.

This movement could not have been sustained nor could it have grown to its present proportions unless the passenger car which started it had not soon been followed by the delivery car and truck. Starting with the materials for building new homes, and continuing with fuel, food, furniture and all the necessities of modern life, the truck has been an indispensable means of transportation. Without it, many of these communities could not have come into being, or if they had, could not have flourished or even survived.

While the first automobiles were passenger vehicles and, as a matter of fact, were looked on as strictly for pleasure or sport, the practical implications gradually became evident. To this end, the Ford Model T undoubtedly contributed in no small measure. However, from the earliest days many of the pioneers of the automotive industry thought of the possibilities of the delivery car or truck. Winton is credited with a delivery wagon in 1898; an Autocar delivery truck was built by L. S. Clarke in 1899 (see Fig. 1); and the first White truck was built in 1900. The White truck was a light vehicle, powered with a steam engine, and was delivered to the Denver Dry Goods Co. of Denver, in 1901.

The Mack Brothers began experimenting with motor trucks and buses, and delivered the first Mack to Harris and Maguire, bus concessionaires in Prospect Park, Brooklyn, in 1900.

The first automobiles were literally horseless carriages. In like manner, the first trucks were horseless wagons. Although the Mack, which was built in 1900, had a powerplant located under a hood, in general, the Rapid, White, Reliance, International Harvester, and Autocar followed the practice of locating the engine under the seat, thus preserving the general appearance of the wagons of that day, minus the horse. (See Fig. 2.)

Truck Types

The 1911 Diamond T is an appropriate selection to show an early departure to conventional automobile lines. This trend has continued through the years with a development parallel to the passenger car. (See Fig. 3.) The enclosing of the driver's compartment to provide shelter from inclement weather followed soon, and then lines were refined and proportions modified until the truck became, if not a thing of beauty, at least less ungainly. (See Fig. 4.)

In the truck field, there were reasons for the existence of the type which has come to be known as the engine-under-the-seat or the cab-over-engine type. For a given length of body, the wheelbase can be made as much as 54 in. shorter than the

[1] See "Chronicle of Automotive Industry in America." Revised edition published by Automobile Manufacturers Association, New York, N. Y.

Fig. 1 – First Autocar delivery truck, 1899 – capacity, 700 lb, including driver

conventional type. As the turning circle is approximately four times the wheelbase, this reduction in wheelbase results in a turning circle about 18 ft smaller than a conventional type having the same body length. In city traffic, this is a desirable feature.

It is also possible to obtain a larger proportion of the body and payload weight on the front axle without having an objectionably long wheelbase. With single tires on the front axle and dual tires on the rear axle, all of the same size, the desirable weight distribution between the axles is one-third on the front axle and two-thirds on the rear axle. Such a distribution can be attained with the COE type, but in a conventional type about 28% is the maximum that can be obtained and still have a reasonable length. For these reasons, there has been, over the last twenty years, a return to the earlier form of construction using more modern styling. (See Fig. 5.)

Following the reappearance of this type of vehicle, there arose in the industry a considerable opinion that it would replace the conventional type entirely. Although this prediction has not been fulfilled, there are many of the COE type being used in all sizes.

Purely as a personal opinion, it is my feeling that this type has its principal field of usefulness wherever the requirement is for a truck for relatively short-haul work, involving a considerable portion of operation in congested areas. There are many factors, such as regulations governing overall length, which create a pressure for the use of this type, to the sacrifice of more desirable characteristics of the longer conventional vehicle. This opinion may be violently rejected by other competent authorities.

Adequate Roads Become Important

The degree to which a car or truck can function is limited only by the dependability of the vehicle and the extent and character of the roads. The advent of the bicycle in the latter part of the nineteenth century provided an awakening to the need for better roads. When the automobile came, the desire to go places and see things pushed the idea along. Such events as the Glidden tours provided the kind of publicity that helped to make the need for more roads and better roads known to the nation. During the early years of the century, as the industry grew and the number of cars multiplied, road building was started as a local movement and rapidly grew to a national program.

Until World War I, there was no chance to use trucks outside the areas in and around towns and cities. Roads were inadequate, solid tires were used on all trucks, and the limitations of the capacity and ability of the vehicles combined to make intercity highway transport impossible. During the war, the need for assistance to the railroads became painfully evident. As a result, when the war ended, activity started at once to get roads on a national scale.

The Lincoln Highway became a symbol around which the people rallied. In 1913, a motor caravan traveled from Indianapolis to San Francisco. In 1919, the army sent a motor convoy from the initial milestone of the federal road system to San Francisco.

These did great service by making it known that, except for a comparatively limited area in the northeastern section of the country comprising parts of New England, New York, Pennsylvania, New Jersey, Delaware, and Maryland, there were no highways adequate for either passenger or freight haulage. Then a miracle happened. Almost as though Aladdin had rubbed his lamp, road building started up all over the land, until today it is almost impossible to find the kind of roads we were happy to ride on thirty years ago.

Roads Needed in Quality and Quantity – The thousands of miles of modern highways which have resulted from this program have pulled us out of the mud, but have not solved all our problems. Great as the building program was, it has been interrupted by World War II and its aftermath. The enforced limitation of labor and material and the increased cost of both since the war, have prevented picking up the prewar tempo, and have adversely affected maintenance. These things, coupled with the tremendous increase in the number of all types of vehicles, have overloaded our highway system to such a degree that, in many sections of the country, the situation is critical.

The highest type of highway constructed to date is typified by the Pennsylvania and New Jersey turnpikes. That these two roads have filled a need is made evident by the great use made of them, not only by passenger cars, but also by trucks.

There is a divided opinion as to the soundness of the policy of building toll roads, and this is not the place to discuss the merits of the differing points of view. They have served the purpose of demonstrating the benefits derived from such features as wide traffic lanes, divided lanes, limited grades, large radii of curvature, long sight distances, and limited access.

Tires

The rubber tire was a prerequisite to a successful road vehicle. The first tires used on trucks were solid bands of rubber vulcanized onto a steel rim. This, in turn, was pressed onto a wheel made of wood, with a steel felloe band shrunk on to hold the spoke and felloe assembly in place. The wheel assembly was provided with a hub made as a malleable iron or steel casting. Over the first 15 years, various types and sizes of solid and cushion tires were developed and used.

The first type had a trapezoidal cross-section, and had a perfectly plain tread. Modifications were made to this basic type by increasing the height of the section, and by providing a tread pattern designed to improve traction and to allow the rubber to flow, so as to give better cushioning against road shocks. The ultimate in this development was to core out the section to still further promote shock absorption. At best, these tires provided inadequate protection against road shock and poor traction and life. There was, moreover, a very definite limit to the speed at which they could be operated.

These tires were made in a variety of sizes. One list[2] shows a range from 32 x 3 rated at 1000-lb capacity, up to 40 x 14 rated at 12,000 lb. The largest size reported[3] is 44 x 16, and the rating would probably have been 14,000 lb. In many cases, preference was shown for mounting two tires; that is, two 40 x 7's instead of a single 40 x 14. While the rating of the two tires was not equal to the single tire, many users of that time felt that the service obtained was better.

Sometime around 1910, a start was made in the use of pneumatic tires. This was on vehicles of 3000 to 4000-lb payload capacity, and gross weights of less than 10,000 lb. The size used was 36 x 6 or 38 x 6. The latter size was developed in 1908[3] for use on fire trucks, but my own memory of the size used on Autocars was 36 x 6. These were single tires on all wheels and were for the limited mileage of those days, and on a small vehicle in delivery service they furnished an acceptable improvement over the solid tires.

Serious work on pneumatic tires for trucks started during the war years of 1914 to 1916.[3, 4] The introduction of cord fabric and its demonstrated superiority to the older square-woven fabric was a start. During World War I, dual wheels for pneumatic tires were produced by Michelin in France, and were used on trucks in the French army. A set of dual wheels was applied to a Locomobile used by General Pershing as a staff car.

The Budd Co. acquired rights to the Michelin dual disc wheel in 1920, and they give credit to the White Motor Co. as having been their first customer in 1921. This marked the beginning of the practical use of pneumatic tires on heavy vehicles.

The tire sizes which were available at this time ranged from the 34 x 5 size, rated at 1700 lb with 80-lb inflation pressure, to 48 x 12, rated at 8500 lb at 140-lb inflation pressure. The first figure of the size designated (34 x 5, and so forth) in this instance is the nominal outside diameter of the tire, and the second figure is the nominal diameter of the cross-section. It will be noted that the diameter of the tire seat was 24 in.

Through several years, the merits of the large single tire versus dual tires were weighed, and resulted in the universal adoption of dual tires for highway vehicles. While the movement in this direction was started by the introduction of the disc wheel, the cast-type wheel, suitable for the application of demountable rims, was not far behind.

The next development was the low-pressure or balloon-type tire which came around 1925. These tires are designated by the nominal cross-section and the tire seat diameter, as 6.00-20. The range of sizes available starts at 6.00-17 with a rated capacity of 1250 lb at 50-lb inflation pressure, up to 14.00-24 rated at 8525 lb when inflated to 80 lb pressure. Due to the limitations imposed by the legal provision for a maximum vehicle width of 96 in., which is prevalent in a large number of the

Fig. 2 – Illustration of early engine-under-seat construction, 1910

[2] See SAE Handbook, 1921.

[3] See "Notes on Development of the Truck Tire," by Earle C. Zimmerman, Issued July 2, 1952 by Firestone Tire and Rubber Co., Akron, Ohio.

[4] See "Development and Use of Pneumatic Tires for Trucks," by W. E. Shively, Goodyear Tire and Rubber Co., Akron, Ohio.

states, the largest tire commonly used is the 11.00 size.

There are three tire seat diameters in use – 20, 22 and 24 in. It is hard to say why this condition developed, and still harder to explain why it persists. In recent years, a trend in favor of the 22-in. diameter has seemed to develop, but because of the cost of changing existing equipment and the desire for interchangeability between tractor and trailer, it does not seem probable that the 20 and 24-in. sizes will soon be entirely replaced.

The service life of pneumatic tires has increased almost continuously since they were first introduced. (See Fig. 6.) Between 1915 and 1935, there was a rapid and continuous improvement in the service life of tires. This improvement was the fruit of the use of new cord fabric, changes made in the compounding of rubber, the adoption of lower inflation pressures, and also a reflection of better roads.

In recent years, the adoption of rayon cord has made a further improvement. First adopted in 1937, it came into extensive use during the war, and now is used for practically all truck tires. Nylon cord has been tried, and is apparently prevented from being more generally used only by the increase in cost. Wire cords have also been used, having been introduced by Michelin in Europe. There seems to be considerable difference in opinion as to the merits of this construction, particularly in highway use.

Accurate data on current mileage are not available. It can be seen that in 1935 the curve was showing a tendency to gain less rapidly than it had 10 years earlier. The dotted line that has been added in an attempt to extrapolate Hale's[5] curve is frankly a guess, based on the known inclination for curves of this character to become asymptotic. The value of 44,000 miles is probably not far wrong for present-day experience where the best operating practice is followed.

Fig. 3 – Diamond T, 1911, illustrating early conventional type of construction

The statement has been made that the number of tires sold in 1949 was about the same as in 1941.[4] In Fig. 7 it is shown that the total number of vehicles in 1949 was 8,000,000, and in 1941, 5,000,000, or an increase of 60%. There is no doubt that the total annual mileage has increased, at least to as great an amount. From this, it could be estimated that tire life had increased by 60%. Undoubtedly, this is not correct because many other factors make such a method of estimating inaccurate. Among other things, the scarcity of new tires during the war served to produce greatly improved methods for recapping tires, and this practice is being more generally followed than before. However, the figures are still significant in showing that improvements in tires are still continuing.

Highway Transport Starts Moving

With roads coming into existence that would permit high-speed, long-distance operation, and tires available that could cushion the shock for both the vehicle and the road, the era of modern highway transport came into being.

We have seen that the earliest trucks were essentially horseless wagons. Later, the truck followed the passenger car in having the engine located under a hood. However, the single vehicle was the predominant type. As with many innovations, the use of a third axle was first adopted on the West Coast. This area has been a leader in providing relatively liberal regulations for highway users. Of course, the incentive for using trucks was great. The railroads either ran east and west or hugged the coast line. This left the great valley between the Coast Range and the Sierras dependent on highways. The great amount of agricultural products that rewarded the provision of water to this area furnished the burdens which called for a means of reaching the markets.

There was discovered, at the same time, fabulous stores of petroleum, and all of these factors, together with the native energy of the people of that region, resulted in the trial of many things.

It was here that the 6-cyl engine was first used. The initial trials were made by converting passenger cars into buses to carry sightseers into the great scenic areas of the states. Dual pneumatic tires and then dual axles were tried. At first, the extra axle was little more than an appendage which had little practical value, but the idea was sound and soon developed into a practical construction that spread into the East.

Multiple Unit Vehicles

During this period, which witnessed many beginnings, tractors and semitrailers made their appearance. The Columbia Terminals Co. of St. Louis started experimenting with this combination in 1919, and by 1924 had converted from horsedrawn

to mechanical transport. The possibility of keeping a separable power unit busy while the load-carrying unit was being loaded and unloaded, provided evident and profitable economies as compared to the single vehicle.

The trailer manufacturers who began to develop this market had an erroneous idea of the basic value of their product. A current slogan which said that a vehicle could pull more than it could carry, and which cited the ability of the horse in this respect, was obviously one of those half-truths which so often cloud the real facts. Although it was true that some excess horsepower and torque were provided to meet the demands of grade and road conditions, it was also true that the capabilities of the engines of that period were inadequate to provide the power for the most efficient operation of even a single vehicle.

The big and impelling reasons which promoted the development and use of the tractor-semitrailer were: (1) the flexibility in operation made possible by the separable power and load-carrying units; (2) the ability to use longer and more capacious bodies than could be mounted on a single vehicle.

The first advantage is so obvious that it is hardly necessary to elaborate. The ability to move a truck around in terminal areas with what amounts to the equivalent of a shifting engine, to service the power unit without the space requirement for a big body mounted on it, to send a spare tractor to pick up a box on the road when a failure in the power unit happens, all can be readily seen and appreciated.

The second advantage is equally important. The articulation provided by the fifth wheel reduces the space needed to maneuver the vehicle and makes it practicable to turn around and turn corners in city streets and in traffic, that would be impossible for a single vehicle with a body only a fraction of the length of what can be accommodated with the combination. The longest bodies mounted on a single vehicle rarely exceed 20 ft, while many trailers are 33 and 35 ft long.

Another very important factor is the ability to provide additional axles. In a single vehicle, three axles have been the maximum number generally used. It is true that 4-axle designs have been built and found useful in military and off-road operations, and several designs have been produced in England and Europe, principally for export to colonial areas. These have been relatively few in number, and the mechanical complications in providing for the equalization of loading and the articulation in the front bogie, together with the need for provision for steering four wheels, is difficult and complicated, even though it is not insoluble.

[5] See *SAE Journal* (Transactions), Vol. 38, January, 1936, pp. 25-44: "When Is a Truck Tire Overloaded?" by J. E. Hale.

[6] See "1951 Motor Truck Facts." Published by Automobile Manufacturers Association, New York, N. Y.

Fig. 4 – Autocar, 1952 – modern conventional type

On the other hand, a 2- or even 3-axle tractor to which a 2-axle semitrailer is attached, provides 4 or 5 axles in a 2-unit combination which it is relatively easy to construct, with designs that are by comparison much simpler.

Therefore, it is not hard to see why, for long-distance highway hauling, the tractor-semitrailer combination has become the accepted unit for all but a very few special cases.

Figures to Show Growth

With all of these developments sprouting up, the middle of the second decade of this century witnessed the start of highway transport on a large scale and over long distances. The start was made with perishable and valuable commodities first, but rapidly grew till the current statistics are almost fantastic.

Truck sales to the domestic and export markets and the totals are shown in Fig. 8.[6] Here, the rise from 1921 to the depression years is plain. The drop from 1929 to 1932 was followed by an equally steep climb to 1937. The real jump came when civilian production was restored in 1945.

The cumulative effect of these sales on total vehicles in service is shown in Fig. 7.[6] Here, we see that in 1919, at the close of World War I, there were about 900,000 trucks in use. In 1951, this has grown to 8,775,000 with 375,000 publicly-owned vehicles in addition, or a total of 9,150,000. This is just over half the trucks in the world. The latest figures on United States and world populations are 151,132,000 for the United States, and 2,400,000,000 for the world. Thus it is with highway transport as with many other services, our country and our people have an abundant supply.

The largest number of trucks, 2,300,000[6] are owned and operated by farmers. They serve in the production and distribution of food and other farm products. In 10 out of 63 major markets, all livestock is delivered by truck. More than half of this number receives 90% or more of all shipments by truck, and the average for all 63 is 75%.[5] Half of the vegetables and fruits shipped out of Florida are hauled by truck.[6]

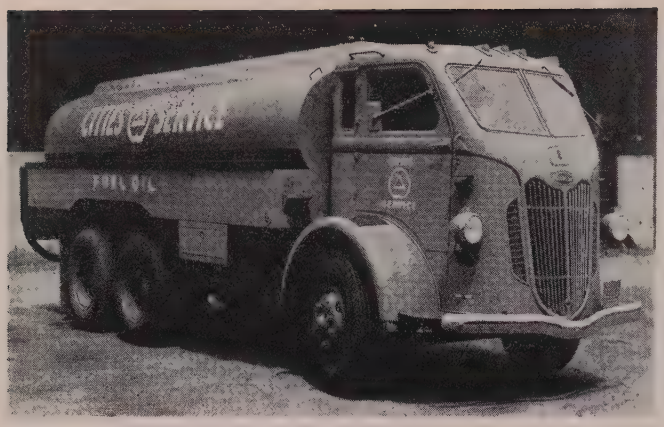


Fig. 5 – Autocar, 1952 – modern engine-under-seat type

There are 1,000,000 trucks operated by for-hire carriers; 360,000 are used by industrial establishments; 425,000 serve the needs of wholesale stores; and 625,000 are used in retail delivery.[6]

Out of 11,100,000,000 tons of freight hauled in 1949, 75% was carried in trucks. Total ton-miles amounted to 1,146,000,000,000, or an average haul of 100 miles. The truck share of this was 122,000,000,000 ton-miles, an average of 15 miles.[6]

The number of people specifically employed to drive these vehicles is 5,137,000. When the number employed in production and sales and service are added, the total is over 5,500,000.[6]

Statistics Not Adequate

While these figures are stupendous, they are, like all statistics, dry and uninteresting. To arrive at an understanding of the real meaning of highway transportation, it is necessary to exercise the imagination and visualize the myriads of points of origin and of destination between which these vehicles are traveling every minute of every day. When this is done, we can obtain some comprehension of the degree of individual service that is provided by it.

Before mechanical highway transport became available, anyone who was located more than a very few miles from a seaport, navigable stream, or railroad, was isolated and dependent on his own resources for everything not obtainable in a very circumscribed area. Today, the highway and truck have made it possible to burst out of these limitations and have integrated every place where it is desirable to live into one community.

Trucks Do Pay for Highway Use – The number of trucks in use is 17% of the total of all motor vehicles in the United States. In 1950, the total of special motor vehicle taxes was $4,354,033,000, and of this sum $1,201,532,000, or 27.6%, was paid by this 17% of all vehicles. A large part of this sum was to pay gasoline taxes which amounted to $1,667,229,000 total, and of this sum $435,000,000, or 26%, was paid by trucks.[7]

These figures are cited to show that trucks pay a good share of the special taxes which are levied against the motor vehicle. The attempt is made repeatedly and mistakenly to create the impression that the provision of highways over which motor trucks are operated is a form of subsidy. It must be apparent that the highway system we have could not exist except for the facility it affords to trucks to supply the necessities of life to the users to whom these highways lead. That any highway system could be built and maintained solely for the use of passenger vehicles should hardly need discussion. Such a system could never have promoted the extensive growth of the last 20 or 30 years. Neither could the many areas which have no other means of communication have been opened to development.

All other forms of transportation developed up to this time are limited to travel between fixed terminals. Therefore, in order to get the articles of commerce to the loading point, and from the unloading point to the ultimate destination, an auxiliary means of transportation must be provided. The time consumed in transfer from one means to another forms such a large portion of the total movement time, that the elimination of this delay results in a great saving. The benefits of this saving are most evident in small shipments.

A most important aspect of the flexibility of truck transport is the benefit which comes from the increase in turnover of inventory that can be effected. Because shipments can be scheduled and completed with such great facility and certainty, banks of material needed to insure against interruption of operations can be reduced from weeks to days. All of these advantages result in direct service and savings to the consumer for whose benefit the highways exist. These are strong arguments against the thesis that trucks are getting a free ride as the result of unwarranted subsidies.

The Engine

The internal-combustion engine has been the nucleus around which the automobile and truck have developed. There are other essentials to making a successful machine, that fortunately became available at the same time. A liquid fuel having the properties of being relatively easily vaporized and containing a large amount of energy was one, and the other was a material that could be used in a pneumatic tire.

When crude oil was discovered in Pennsylvania and a refining industry was created to produce a better illuminant than the candle, the way was opened for the production of gasoline and fuel oil that provides the fuel for cars, trucks, buses,

[7] See "1951 Automobile Facts and Figures." Published by Automobile Manufacturers Association, New York, N. Y.

locomotives, farm equipment, and many other types of equipment. Likewise, when Goodyear in England discovered the process of vulcanizing rubber, the start had been made to making tires that would cushion road shocks, and provide a material having a high natural adhesion on the road.

The way in which processes for producing fuel of various and desirable characteristics has developed, and the work of Kettering and Midgley in learning about the causes and nature of detonation, is a story of immense interest and importance. The same is true with regard to the development of the tire. However, these subjects, important as they are, do not properly come within the scope of this account for detailed consideration.

The engine, however, even though it is basically similar to the passenger-car engine, has required sufficiently different design to meet service demands, that a study of these characteristics is desirable. The earliest engines used had two or four cylinders. The speed was limited to 1400 rpm, or 1800 rpm at the most. Cylinder heads were integral with the block, and on the 4-cyl engines not more than two cylinders were cast into one block which was detachable from the crankcase. As the valves had to be passed through an opening in the head which was generally closed with a screw plug, the placing of the valves on opposite sides of the bore was a popular arrangement because it reduced the length of the engine, as compared to a side-by-side location as in the L-head. The valves themselves had cast-iron heads threaded and brazed to low-carbon steel stems. This provided a seat material that was compatible with the seat in the block, but the weight ruled out any attempt at speed.

Crankshafts looked like hairpins in comparison to present-day practice, and they had no sign of a counterweight or a dampener.

Vibration and noise were deafening. The valve gear was open, and no attempt was made to provide quieting ramps or to control the acceleration of the valves. Flywheels were also out in the open, and in some cases even timing-gears ran naked.

Bearings were made of babbitt, directly cast into the big end of the rods and frequently into the main bearing caps, and in some instances in the case as well.

Accessories were limited to a cooling fan, a carburetor, and an ignition device, generally a high-tension magneto, except in the very earliest days.

Carburetors were of the most elementary types, with almost no provision for obtaining or maintaining a constant mixture ratio, and there was no means for furnishing heat to promote vaporization. This, of course, was made possible by the extreme volatility of the gasoline of that day.

Compression ratios were around 4/1, and the power output was low, as was the fuel efficiency.

Even up to 1917, when a group of specialists was recruited by the SAE to work under the control of the Quartermaster Corps, and later the Motor Transport Corps, slow speed and moderate performance were characteristics of the engine developed for the Class B military truck. After the war, the advent of pneumatic tires and the road improvement already commented upon stirred the idea of intercity hauling, and the need for more power and better performance became apparent. It soon became evident that the inherent limitations of the 4-cyl engine, both in displacement and speed, made it necessary to take a new look. At about this time, in the passenger-car field, 6-, 8-, 12- and even 16-cyl engines were being produced, and this gave added impetus to the development of new engines.

Not long after 1920, some old 6-cyl passenger-car engines were modified and put into trucks, but their use principally showed up weaknesses that must be corrected to provide a desirable truck engine. In 1925, Mack produced a 6-cyl truck engine, and by the early '30's the move to six cylinders was in full swing, and this type has been used almost to the exclusion of any other for the past 20 years. Ford, with the 8-cyl V-type engine, has been almost, if not entirely alone, in the use of anything other than a 6-cyl engine. The LeRoi Co. has produced an 8-cyl V-type engine of 540 cu in. displacement for Autocar, and vehicles with this engine have been offered just recently. The 6-cyl engines which were built at first had about 500 cu in. displacement, and at that time would produce about 100 hp at a speed of 2000 rpm. Since that time, engines have been built as large as 1000 cu in., developing up to 300 hp at about 2200 rpm.

However, in the middle '30's these larger engines were not available, and if they had been, there would have been considerable resistance to using them because of the cost of fuel. There was an insistent demand for more power, particularly on the West Coast, where gross combination weights of 72,000 lb were permitted. Also, in this area, many long and severe grades were encountered.

This demand was met by the Cummins Engine Co. with a 6-cyl diesel engine of 677 cu in. displacement and developing 150 hp at 1800 rpm. Several things occurred at about the same time to make the introduction of this engine auspicious. First, there was the demand for more power to provide better performance. Second, there was available a large supply of fuel oil of adequate quality at a very low price compared to gasoline. Third, the lower specific fuel consumption of the engine gave added economy.

The first installations were effected by replacing the gasoline engines with which the vehicles were equipped as a field change. Soon vehicles equipped with these engines were offered by several companies, and for some years this type of equipment was used almost to the exclusion of anything else on the West Coast. Naturally, these exceptionally favorable conditions did not last. The demand created for fuel oil in trucks, together with the industrial development and railroad use, combined to raise prices. Also, it was inevitable that the

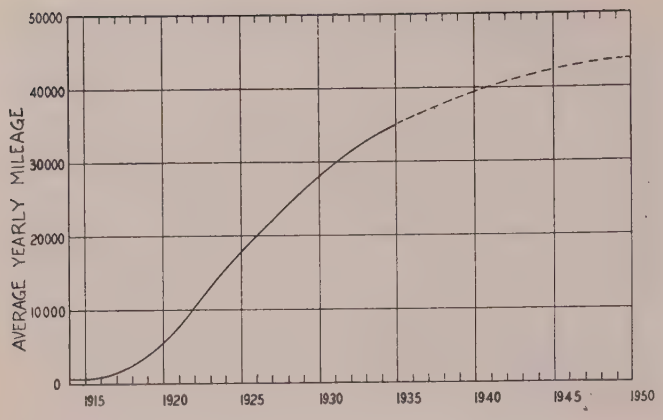


Fig. 6 – Year-by-year trend of increase in service life of pneumatic truck tires

absence of tax on fuel oil for highway use would soon be noted and changed.

Although other manufacturers have produced competitive engines, Cummins has really had only one serious competitor in the highway field in the GMC diesel. This engine operates on the 2-stroke principle, and in this respect is unique in the vehicle field. It has been developed to a high degree, and together with other engines, has contributed to the marked swing to diesel equipment in long-distance highway hauling. There have been many developments of design and material during the twenty or more years since the 6-cyl engine was adopted, that have caused a steady improvement in performance and service.

In the field of design, the work on combustion chambers by Ricardo in England, and Horning, Taub, and Janeway, in this country, paved the way for combustion control. At the same time, the developments in fuel characteristics which grew out

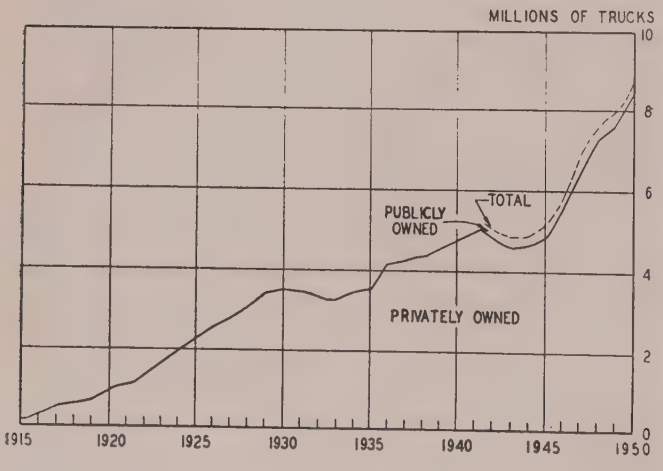


Fig. 7 – Total trucks in United States

of the researches of Kettering, Midgley, and Boyd, led to an improvement in performance which is still continuing to reach new records, and for which the end is not yet in sight. Crankshafts and bearing designs have also been improved steadily, so that higher speeds, which coupled with higher mep produce more power, are made possible. This has been accomplished by making the shaft diameters larger and the cheeks heavier, and by keeping the strokes short so that the main and rod bearings have a considerable overlap.

The added torsional and longitudinal stiffness thus provided has reduced the localized overloading of bearings, as well as raised the critical frequency of vibration of the shaft above the operating speed range. The use of counterbalance weights has further reduced the loads imposed on the bearings by the centrifugal effects of the rotating elements of the shaft and the connecting rod. Naturally, these masses, which added to the weight of the shaft but contributed nothing to its stiffness, resulted in reducing the frequency. This gave rise to the need for some type of torsional damper which can act to control the amplitude of vibration of the shaft and keep it within permissible limits.

Even such seemingly small details as the size, accuracy, and finish of the radii which join the bearings to the cheeks have had careful study and have yielded large returns in longer life. Although for many years the value of a hard surface for the bearings was realized, the means for attaining such a surface was a long time coming. Mack, for many years, even in their 4-cyl engine, used a carburized and hardened shaft, but the production difficulties were so great that they had no followers.

Then Dunn, at the Ohio Crankshaft Co. succeeded in perfecting the induction-hardening process known as "Tocco." With this process, it is possible to select a steel and heat-treat it to provide the basic requirements for strength and toughness. This leaves the shaft at a hardness that permits performing the necessary machining operations. The bearing surfaces can then be hardened, and the hardened zones can be localized so that they do not enter the highly stressed areas. The increased quantities of heat liberated to produce greater horsepower and torque per cu in. in these new designs, have made demands for improvement in valves and valve seats, piston rings, bearings, and lubrication.

Valve life and performance have been improved by such design features as sodium cooling, first used in aircraft engines. A great deal of detail study has been applied to cam design that has resulted in reducing the acceleration of the valves and operating mechanism, and has lowered the velocity with which the valve is picked up and placed back on its seat. In order to remove the vagaries of inexpert or careless adjustment, constant clearance is attained by the use of automatically adjustable lifters. The type most generally used is the hydraulic lifter.

Even with these improvements, a valve seat

formed directly in the cylinder block or head has been found unsatisfactory, and for exhaust valves particularly, seat inserts have been provided. Great benefit to increased life and reduction in maintenance has resulted from the use of valve rotators. Particularly with the use of a positive type, such as that produced by Thompson Products Co., the tendency toward the formation of localized pockmarks that induce blow-by and burning is eliminated almost entirely.

Piston rings have been made thinner in the axial direction of the bore, and increased radially. The group of rings has been moved away from the top of the piston so that they may have better protection from flame and products of combustion entering the space between the cylinder bore and piston. Probably the biggest single improvement in piston rings has been the chrome plating of the top ring. Various expedients have been introduced for producing a predetermined and controlled taper to control lubrication, oil passing, and blowby. There is seemingly an infinite variety in oil-control rings. Some are made of cast iron and some of steel. Many, if not all of the most successful types, have some means, besides the elasticity of the ring itself, provided to maintain a constant wall pressure.

The general pattern of engine development is illustrated with reasonable accuracy by our own experience. Beginning in 1907, when we produced our first truck, a 1½-ton delivery wagon, we used a 2-cyl opposed engine, having a displacement of 160 cu in. and developing 16.5 hp at 1400 rpm, and 90 lb-ft torque at 600 rpm. The imep was 100 psi, and the specific fuel consumption was 0.860 lb per bhp-hr. The compression ratio ranged from 4.0/1 to 4.2/1.

In 1919, we produced a 4-cyl engine of 276 cu in. displacement. However, the 39 hp developed by this engine was due to increased displacement and speed, 1800 rpm. The compression ratio remained unchanged, and the imep as well. Fuel consumption dropped to 0.800 lb per bhp-hr.

By 1929, we had a 4-cyl engine of 350 cu in. displacement, and had increased compression ratios to 5.0/1. However, the speed was still 1800 rpm and imep had not been improved, but we did get 48 hp, and fuel consumption was under 0.70 lb per bhp-hr.

Then came the 6-cyl engine, and through the years there has been a progressive development until we now have experimental engines that can operate on standard grade fuel with compression ratios of 7.0/1, which run at 2800 rpm, developing 200 hp and 430 lb-ft torque, which is 150 psi imep, and specific fuel consumption 0.535 lb per bhp-hr.

The curves of hp (Fig. 9), imep (Fig. 10), and specific fuel consumption (Fig. 11), plotted against years, give a graphic illustration of these changes.

With diesel engines which, as has been stated, were first put into trucks around 1930, the story has been much the same. The improvements in materials already described have played an equally important role in developing the performance and

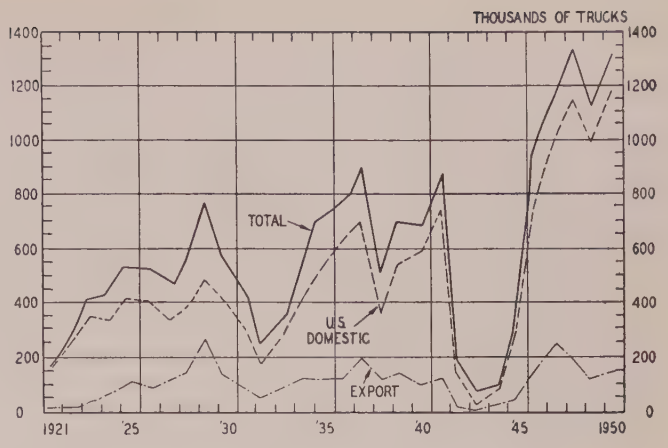


Fig. 8 – Sales of trucks per year

dependability of this type of engine. Starting out with an advantage over the gasoline engine, in that the fuel consumption was lower, it was handicapped somewhat in the greater weight and cost of the original engines.

Lower fuel consumption was obtained from the higher compression ratios used, which resulted in better thermal efficiency at full load. Also, this better efficiency was retained at part load where the gasoline engine dropped, due to the fact that speed and load variations were accomplished by throttling the air intake. In the diesel engine, variations in speed and load demand are produced by changing the quantity of fuel injected, and the quantity of air taken in remaining unchanged. As a result, the compression pressure is maintained and full-load efficiency is retained through a great part of the operating range. In a highway vehicle subject to frequent and sudden changes in power requirement and speed, this is an important factor.

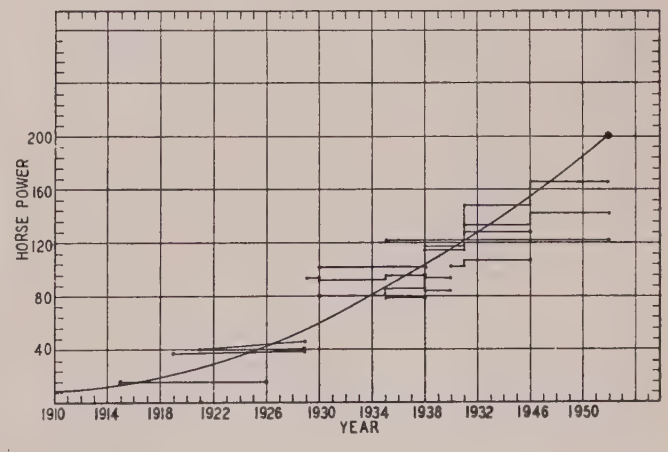


Fig. 9 – Increase in horsepower from 1910 to 1950

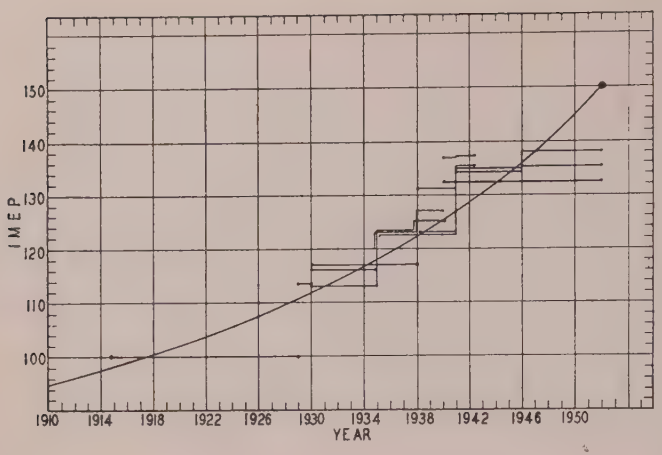


Fig. 10 – Increase in indicated mean effective pressure during 1910 to 1950

The use of lighter materials and design changes have continuously reduced the weight, and the larger volume of production has resulted in reducing cost. Nevertheless, it is only in recent years that the introduction of supercharging has made it possible to produce an engine which, for the same displacement and weight, develops equal power, and can be operated at speeds comparable to those of gasoline engines. The additional cost of the supercharging apparatus has not helped the disparity in cost.

Before this latest development, the penalty in weight for a complete installation was at least 1000 lb. This difference and the greater cost limited the use of the diesel engine to long distance operations, where the element of fuel cost was a sufficiently large proportion of the operating cost to permit the savings from this source to overbalance the loss in payload and increase in investment.

To date, the application of supercharging has been more in the area of increasing horsepower. The engines of 743 cu in. displacement, naturally aspirated, develop from 165 hp at 1800 rpm to 200 hp at 2100 rpm. By supercharging, the output is increased from 165 to 225 at 1800 rpm, and from 200 to 300 at 2100 rpm. The increase in power produces economies in running time which affect the labor costs that, under present conditions, are such a significant part of operating expenses. However, in these large engines, the gross weight of the engines has been increased by the addition of the supercharger, even though the weight per horsepower has been greatly reduced as a result of the increased horsepower developed. Unfortunately, to operate within the limits of the gross weights permitted by the various regulations, it is the gross weight of the engine which is important.

Within the last few years, several engines, one made by Cummins and one by Buda, have been produced which have the objective of using the supercharging principle to reduce the gross weight. The Cummins JBS-600 engine has 401 cu in. displacement and develops 150 hp at 2500 rpm, 360 lb-ft torque at 1400 rpm, and the installed weight is 1543 lb.

Buda has two engines. One has 468 cu in. displacement, developing 135 hp at 2400 rpm and 341 lb-ft torque at 1600 rpm, and having an installed weight of 1750 lb. The second engine has 516 cu in. displacement, develops 165 hp at 2400 rpm and 400 lb-ft torque at 1600 rpm, and weighs 1850 lb.

These engines are closer in weight and performance to the gasoline engines generally used in vehicles of around 50,000 lb gross weight, the size unit acceptable under the regulations of many states in the East and Middle West. There has not been sufficient operating experience with these newer engines to be able to determine the degree to which they will fulfill the objective set for them. However, enough has been learned to indicate that there is distinct promise that they will provide an engine that will make it possible to produce a chassis that can better meet the weight requirements and provide comparable performance and improved fuel consumption in relation to gasoline-powered units.

The 2-stroke diesel engine built by the Detroit Diesel Engine Division of GMC entered the automotive field in 1938. Three engines having three, four, and six cylinders were offered. All three engines had cylinders of 71 cu in. displacement, and a number of parts were identical.

Taking the 6-cyl engine with a total displacement of 426 cu in. as an example, we learn that it started out in 1938 with a rating of 165 bhp at 2000 rpm. In 1946, the rating was increased to 200 bhp at 2000 rpm, and in 1951 the rating was again increased to 225 bhp at 2100 rpm.

The means employed to attain this improvement in performance is much the same as has been explained with respect to the gasoline engine. Detail changes in materials for bearings, head gaskets, and timing gears, are typical of one path. Changes in oil-hole location and roll-burnishing fillets at crankpin journals on the crankshaft, together with adoption of a torsional damper of the rubber-bonded type, and later one of the viscous-friction type to replace the earlier springpack type harmonic balancer, were other examples of improvement.

In short, in this instance again, careful attention to service history, coupled with detail design, production, and material changes, has resulted in continuous improvement in performance and life.

Power/Weight Ratio

One of the big problems that the industry has struggled with is the amount of power required. In the early days, when single units were the predominant type and with the limitations of solid tires and poor roads, even the low horsepower of the engines then available still produced an acceptable

performance as measured against the standard of that day. When roads were improved, pneumatic tires soon arrived, and long-distance hauling was started, and the question of how much power should be provided became very important.

Beginning in the middle '20's and coincident with the introduction of 6-cyl engines, the trend to more power developed and seemed to be here to stay. The depression of 1929 and the early years of the 1930 decade brought a change. With the introduction of tractor-semitrailer combinations, there began a period during which the use of light tractors with engines of moderate power came into use. These tractors were originally modifications of smaller trucks made in large volume and, consequently, were relatively low in price. Naturally, the builders of these vehicles exploited this new market. While trials showed that there was an area in which they were successful and practical, they also served to point out the desirable features inherent in better performance potentials. As a result, around 1940, a trend toward larger and more powerful tractors started and is still continuing.

Fig. 12 shows this general trend[6] pretty clearly. Prior to 1946, the classification was different than for the period between 1946 and 1951, but there is enough similarity to make a reasonably good comparison. The vehicles up to and including 14,000 lb gross weight are probably all single units used in retail delivery, on farms, and in similar services. From 16,000-lb gross weight and above, considerable numbers are used as tractors, and the proportion increases as the gross weight goes up.

This trend toward larger vehicles is a natural reflection of the lower cost per ton or other unit, which can be obtained from more capacity and better performance. Unfortunately, there is not much information readily available and in a form which can be used to provide quantitative figures to support this statement.

Pennsylvania Pilot Study

Several years ago, the Highway Research Board formed a committee with the imposing title, "Committee on Economics of Motor Vehicle Size and Weight, Department of Economics, Finance and Administration." The committee has set up as a task the attempt to determine what size of vehicle or combination can be most economically operated, both from the standpoint of the vehicle and the type of road that must be provided for it.

The first part of the program was a project designed to determine the variation in fuel cost and time for a large range of vehicle sizes and weight-power ratios over two different types of roads.

The project became known as the "Pennsylvania Pilot Study," which was not a very appropriate title. Associated in this cooperative program were: the Automobile Manufacturers Association, which provided the powered vehicles, mechanics and insurance; the Truck-Trailer Manufacturers Association, which supplied the semitrailers and full trailers; the American Trucking Association, Inc., and the National Council of Private Motor Truck Owners, which furnished the drivers; the Department of the Army, which sent a wrecker for loading the vehicles; the Pennsylvania Department of Highways, which made available the facilities of two State Highway garages for bases of operation and personnel for recording data; the Pennsylvania Turnpike Commission, which permitted toll-free operation on the Pennsylvania Turnpike; the Bureau of Public Roads, which supplied the instrumentation and supervising personnel; and eight major eastern petroleum companies, which furnished the gasoline and lubricating oil.

Carl Saal, Highway Transport Research Engineer, Highway Transport Branch, Bureau of Public Roads, was in charge of the project.

The route selected was a closed circuit comprised of the Pennsylvania Turnpike from Carlisle to New Stanton Interchange, and return via State Highways, U. S. 30 and 11.

The Pennsylvania Turnpike is representative of the best in modern roads, with divided traffic lanes, maximum grades of 3%, large radii of curvature, and long sight distances. U. S. 30 and 11 are good state highways through mountainous country, and though they have been rebuilt and improved tremendously in the last 20 years, in general they form a 2-lane road with sharp corners and restricted sight distances, and grades as high as 12%. Some grades are several miles long, averaging 6 to 8%. The types of vehicles used in the test are listed in Table 1.

As can be seen from Table 1, each vehicle was tested with three different loads, and the complete range extended far beyond any currently recognized limits. While there is some overlap between the different vehicles as far as gross weights are concerned and also as to power/weight ratio, it is doubtful that comparisons can be made to compare

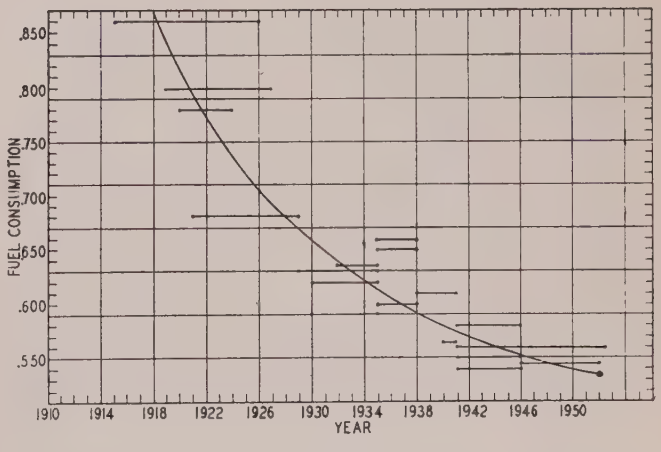


Fig. 11 – Increase in specific fuel consumption from 1910 to 1950

Table 1 – Pennsylvania Pilot Study

Test No.	Vehicle Type	Gross Vehicle Weight lb	Weight/Power Ratio
1	2-axle single	20,000	177
		24,000	211
		26,500	236
2	3-axle single	36,500	250
		44,500	305
		54,000	369
3	2-axle tractor	33,000	293
	Single axle	41,500	372
	Semitrailer	50,000	448
4	2-axle tractor	50,000	313
	2-axle semitrailer	62,000	387
		74,500	465
5	3-axle tractor	63,000	350
	2-axle semitrailer	80,000	445
		96,000	534
6	3-axle truck	80,000	358
	3-axle full trailer	100,500	449
		119,500	534
7	3-axle tractor	93,500	339
	2-axle semitrailer	116,500	421
	2-axle full trailer	139,500	505

the different vehicles and combinations. Thus, Nos. 2, 3, and 4 overlap in both these items, but it cannot be determined with certainty that the difference in performance is due to the No. 3 truck having a dual drive, whereas the other two have single axles.

Other comparisons are still possible which provide interesting and valuable information relating to the effect of gross weight and highway characteristics on fuel consumption and running time. Fig. 13 shows the average speed in mph plotted against gross weight. Two curves illustrate the difference in performance resulting from the difference in the roads. It should be emphasized that the drivers were representative of the most skilled and competent available. Also, the vehicles were operated in accordance with safe practice and regulations. Therefore, the higher speed on the turnpike was a direct result of the greater speed that could be reached on lower grades and longer curves.

It will be seen that the speed of the No. 1 vehicle with the lightest load was about 10% less on U. S. 30 and 11 than No. 7 vehicle with the heaviest load on the turnpike. Whereas the speed on the turnpike decreased from 47 mph for No. 1 vehicle at 20,000 lb, to 36 mph for No. 7 vehicle at 140,000 lb on the turnpike, the same comparison on U. S. 30 and 11 was 30 mph against 20 mph. The difference between No. 1 vehicle at 20,000 lb on the two roads was 45 mph against 30 mph or a loss of 33%. On No. 7 vehicle at 140,000 lb, the comparison was 36 mph against 20 mph, or 45%.

Although it has already been pointed out that the weight/hp ratio is not the only factor that affects the performance, there are other interesting comments that can be made. The weight-to-power ratio for the No. 1 vehicle at 20,000 lb is 177 lb per bhp; for No. 7 vehicle at 140,000 lb, it is 505. To provide the same ratio for No. 7 vehicle would call for 790 bhp instead of 276 net bhp which this vehicle had. Even then, the speed would not equal the 36 mph attained by it on the turnpike.

Fig. 14 shows the fuel consumption in miles per gallon versus gross vehicle weight. Here the ratio between the lightest and heaviest vehicle is shown to be approximately 3/1 on either road. However, the difference between the two roads, 27% for No. 1 vehicle at 20,000 lb and 73% for No. 7 vehicle at 140,000 lb, both being the increase in miles per gallon for operation on the turnpike compared to U. S. 30 and 11, is also considerable.

Comparing miles per gallon is instructive, but it gives no information about the work done. The product of the weight by the distance covered while consuming 1 gal of fuel (ton-miles per gallon) is also worth considering. In Fig. 15, this is shown versus gross weight. Here we see that, even though increasing the weight requires the consumption of larger quantities of fuel (less miles per gallon) (see Fig. 14), the weight carried increases more rapidly than miles per gallon decreases and the ton-miles per gallon increases.

The effect of the road is shown in two ways. The rate of increase is much greater on the turnpike and the difference in results between the two roads is apparent. For No. 1 vehicle at 20,000 lb, the difference is 36% and for No. 7 vehicle at 140,000 lb, it is 74%.

There is still another factor that must be taken into consideration, which is the rate at which the work is done. Therefore, if the weight multiplied by the rate of fuel consumption (ton-miles per gallon) is further multiplied by the speed (miles per hour), we obtain a factor which seems to have great significance. Fig. 16 shows this relation. While it is evident that there is a similarity between this pair of curves and those in Fig. 15, there are very evident differences. In Fig. 15, the ton-miles per gal-

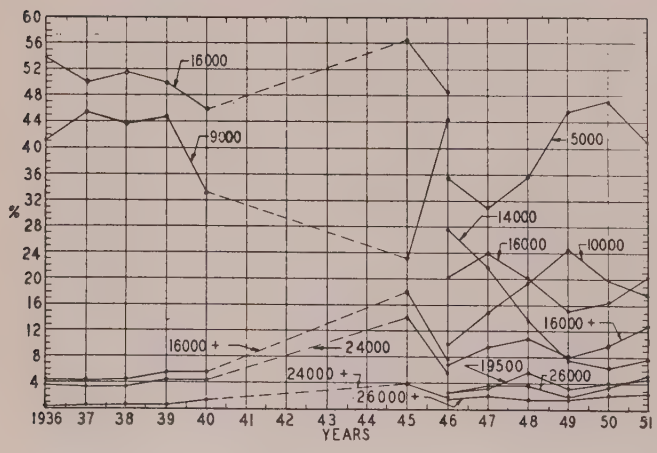


Fig. 12 – Illustrating sales of trucks of various sizes

lon on U. S. 30 and 11 increased slightly as the weight was increased. In Fig. 16, a horizontal line provides as good an average for the plotted points as can be made. This means that the speed has decreased at such a rate that it has canceled the increase in ton-miles per gallon.

On the turnpike, there is a similar change in the slope of the curve, but not so drastically. However, it can be seen that the effect of the road differences is made more distinct. For the No. 1 vehicle at 20,000 lb, there is a difference of 90%, and for vehicle No. 7 at 140,000 lb, there is a difference of 210%.

Lest somebody gets the impression that the attempt is being made to promote 140,000 lb vehicles, let it be said that this was not the purpose of the study, nor is it the purpose of this discussion.

These data are the only material of the kind that I know of, and it certainly seems that the great differences in performance and fuel consumption that are brought to light must be taken into consideration in planning roads for the future. The fact that vehicles can be built that can capitalize on the characteristics of these high-type roads is also encouraging in that we can see great areas of opportunity still to be exploited in highway transport.

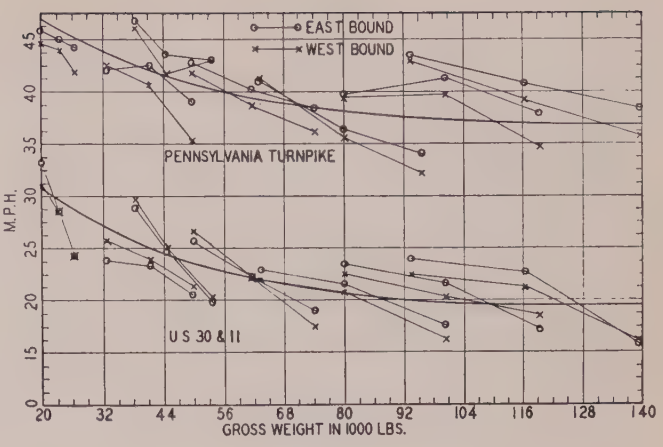


Fig. 13 – Effect of road and load on speed

Other Mechanical Features

There are, of course, other units in the truck that are important, even though the contribution they have made is not so spectacular. The clutch is essential because the internal-combustion engine cannot be started under load. The clutch performs the function of coupling the engine to the driving mechanism in a manner which makes it possible to effect the connection at will.

The cone clutch was the earliest type to be used. While it was simple and effective after engagement, it was difficult to construct in a manner which would permit smooth engagement. Various other types have been used, such as multiple discs, both dry and in oil, but the plate clutch has become the surviving type in general use today. The single-plate type has proved to be adequate for engines of 500 lb-ft torque, and beyond this 2-plate types are used most generally. The distinction is in the number of driven plates. While this type provides easy and smooth engagement and produces a powerful drive after engagement, it is far from being an ideal device. Being a friction device, the time of the engaging is limited; otherwise, the heat generated by slipping will destroy it. Although the fluid coupling has many desirable features, the comparative cost and susceptibility to misuse has prevented the use of this device.

The transmission operates in conjunction with the clutch to provide for the limited torque range of the engine. Within the range of speed at which the engine will run satisfactorily, the torque developed is essentially constant. Therefore, to meet the widely varying requirements, some means must be provided to multiply this torque. The device which was originated, the clash gear, is still in use, albeit with some improvements. There probably is no similar example of such an unsuitable device surviving for so long a time anywhere.

The original transmissions had three speeds and were of the progressive type; that is, a single sliding element on the mainshaft progressively engaged mating driving gears. The next step was a 4-speed selective type; that is, one having separate sliding elements on the mainshaft with control means that could be selected so as to place one element in driving connection, while the others were secured in a neutral position. Succeeding improvements have provided dog or toothed clutches for engagement instead of the driving gear teeth, and the application of clutch brakes or synchronizers has facilitated making the required changes with less skill, effort, noise, and destruction. Tribute must also be paid to the metallurgists for having developed types of steel and heat-treatments which have resulted in metal capable of providing an astonishing resistance to the abuse received.

In an effort to provide more flexibility, a 5-speed transmission was produced, also of the selective type, and this has become the predominant type except in the smaller vehicles. A division of practice has developed these 5-speed boxes into overdrive and direct-drive types. From the standpoint of performance, there is no difference between these two types. The range of gear ratios which can be secured practically in five steps is approximately 8.0/1, and whether one starts with an overdrive top gear of 0.75/1 and ends up with a low gear with 6.0/1, or starts with a top gear of 1.0/1 and ends up with a low gear of 8.0/1, makes no difference if an appropriate rear-axle ratio is selected.

The only essential difference is that driveshaft speeds are greater with the overdrive type, and

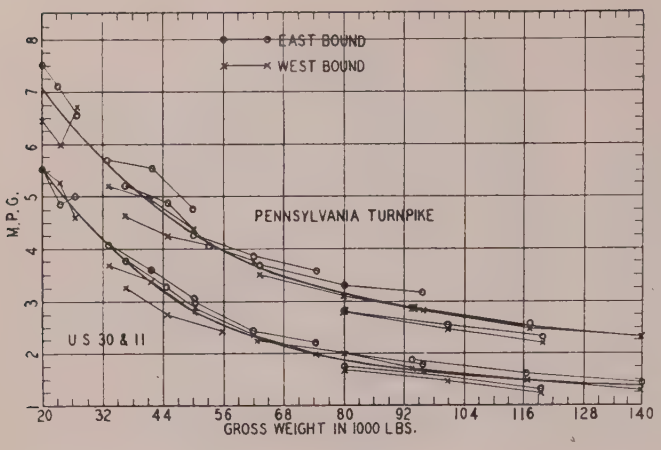


Fig. 14 – Effect of road and load on fuel consumption

gear and bearing loads are greater, and the size and weight of the box is greater for the direct-gear type. In practice, both types have given satisfactory results.

As the size of vehicles increased and the range of operation was extended, the need for more gear steps was provided by dual-range transmissions that had 10 speeds, or auxiliary transmissions which provided 15 speeds. This trend is a recognition of the fact that the ideal would be a device which would provide an infinite range of speeds without definite steps. Many attempts have been made to produce such a device, but none have yet been successful because of size, weight, cost, or low efficiency.

To a very large degree, the continuing use of the friction clutch and shiftable gear transmission with the degree of service it gives, is due to the skill exhibited by truck drivers. The final drive for automobiles has generally been through the rear wheels. None of the designs of front-wheel drive have survived. In trucks, 4-wheel drives were produced early and have continued to be used largely as special purpose types. The large majority of trucks are 2-axle types with the drive in the rear axle.

The first type of final drive was by a jackshaft, sprockets, and chain. This type had many advantages, but one disadvantage was of a serious nature. The chain, being exposed, was subject to wear, was hard to lubricate, and was noisy. Then, too, it had limits regarding the speed at which it could be operated.

In the effort to provide an enclosed drive, several paths have been followed. Pierce-Arrow imported the worm gear from England. Autocar developed the double-reduction gear set which it had first put on an automobile in 1901. Both of these types have continued in production and have supplanted the chain drive except for some off-road operations, and even here the design, such as Euclid uses, having a planetary gear set in the wheels, is in the greater number of vehicles.

The worm-gear axle was developed to its present status by Col. Herbert W. Alden and his associate, L. R. Buckendale. It had one great advantage – silence – but there were many problems attending the design, selection of materials, and lubrication, that had to be solved.

In the endeavor to secure silence, the straight-tooth bevel gears were replaced by spiral bevels, and more recently by hypoid gears, and the straight-tooth spur gears by helical types. Today, there is no difference in quietness between these different gear types.

Axles have also been designed in multiratio types in the effort to furnish a larger number of speed ratio changes between the engine and the wheels. The Eaton axle is a single-reduction axle with a planetary gearing located between the drive gear and axle shafts. With the planetary set locked, there is direct connection between the gear and axle shafts giving the high range. With the sun gear locked to the case, an additional reduction of about 1.33/1 is introduced.

Timken has a double-reduction type in which there are two sets of spur gears driven by a single-hypoid gear set. The gears are attached to the differential, and the pinions run free on the jackshaft, and can be selectively clutched to the jackshaft, furnishing two ratios that differ in the same order as the Eaton axle. Shifting from one speed to the other is by a power mechanism which permits a preselection of the desired change; the actual shifts follow a release of the throttle or the clutch.

The value of dual-speed axles and auxiliary transmissions, together with the desirable steps between ratios, has been and still is being vigorously debated. No attempt will be made here to add to this discussion, beyond the observation that where they are applied intelligently each will give a helpful improvement in flexibility, and thus make a given vehicle able to cope with a wider range of load and road conditions. On the other hand, gear ratios are mechanical means for relating torque and speed, and have no ability to change horsepower. As horsepower is a function of the engine, it should be obvious that gear ratio changes are useless as a means of increasing maximum speed, either on the level or on grades. There probably are many instances where the ability to use a slow ratio when loaded, and a faster ratio for a return empty, is an economy feature that is worth while.

With the more powerful engines and higher road speeds of today, the need for slow gear ratios is reduced, and single-reduction axles using spiral bevel or hypoid gears, have come into increasing use. For equal capacity and serviceability, there is little difference between any of these types as far as size, weight, and cost are concerned.

The fact that gear sets have been developed of such relatively small size, weight, and cost, is little less than a miracle. If the classical concepts of gear

design had been followed, there never would have been such a device built. By breaking away from these rules, and through many trials and errors, failures were studied and weaknesses corrected that have resulted in a triumph that has been accorded little in the way of recognition or even of understanding.

Controls

The ability to control the speed and direction of a highway vehicle is obviously of paramount importance. Steering, the control of direction, has until comparatively recently been by the muscular effort of the driver through a steering gear and a system of links connecting the steering gear to wheels pivoted on the front axle. By providing a satisfactory mechanical reduction in the steering gear and linkage and by conforming to simple geometric requirements in arranging the links, the effort necessary for steering was kept within the limits set by the driver's strength, and stable directional control was obtained.

There is a wide range of effort required as the speed of the vehicle changes. When the vehicle is stationary or at very slow speeds, such as obtained in docking, the effort is high. As the speed is increased, the wheels roll through a change of course instead of being twisted against the road, requiring a torsional slip between the tire and the road, and the effort needed is reduced. As the vehicle speed is increased, the importance of speed in ability to change the direction of the steering wheels increases. If the mechanical advantage is made large enough to provide the desirable ease of steering at low vehicle speeds, then the required speed of control for high vehicle speeds is sacrificed.

It has been possible through the use of efficient types of steering gears currently available to provide a reasonably satisfactory solution to these conflicting requirements.

A number of designs calculated to improve steering have been proposed and have been used with reasonable success. Francis Davis, some 20 years ago, started to design a hydraulically assisted type of steering gear, which has resulted in a design recently incorporated in steering gears that have been adopted for passenger cars. The impetus thus provided will probably produce similar equipment suitable for truck use. Units of this type are now available, and it remains to determine how to apply them in the most practical manner.

With regard to speed control, there are two requirements – acceleration and deceleration. For facility in meeting emergencies, both are equally important. Acceleration is limited by the reserve drawbar pull available, and deceleration is limited by the adhesion between the tire and the road surface. Indeed, this is also the limit for acceleration, but providing the engine power to reach this limit is not so easy on heavy vehicles as it is to provide brakes equal to slide the wheels. There is one catch to sliding the wheels with the brakes. When they are sliding, they have no effective directional control.

The first brakes were located on the rear wheels only and were generally external band or shoe types operated by manual effort. When front-wheel brakes were put on passenger cars, they soon followed on trucks. Extensive work was done in developing servo mechanisms of various types in the endeavor to lessen the manual effort required, but none of these were adequate. The introduction of a hydraulic system to replace rods and levers simplified the installation of controls and made them more efficient, but they were still inadequate for the larger, heavier vehicles, and the higher speeds which were being required.

The vacuum booster produced by Bendix was the first practical application of auxiliary power to assist the driver. At about the same time, a full power system was produced by Westinghouse. Both systems have been intensively developed and are used on all but the lightest trucks. While they have made it possible to provide all the power needed to operate the brakes with minimum manual effort on the part of the driver, there is some inevitable and, so far, irreducible loss of application time. When it is recognized that at 60 mph each 0.1 sec represents distance traveled of 8.8 ft, and when it also is recognized that the maximum rate of deceleration cannot generally exceed 15 ft per sec per sec, the importance of reducing this lag in application time to the maximum can be realized. With apparatus presently available, application time of the order of 0.5 sec can be obtained and retained with careful inspection and maintenance.

Regulation

There can be no rational disagreement about the need for regulations to control the characteristics

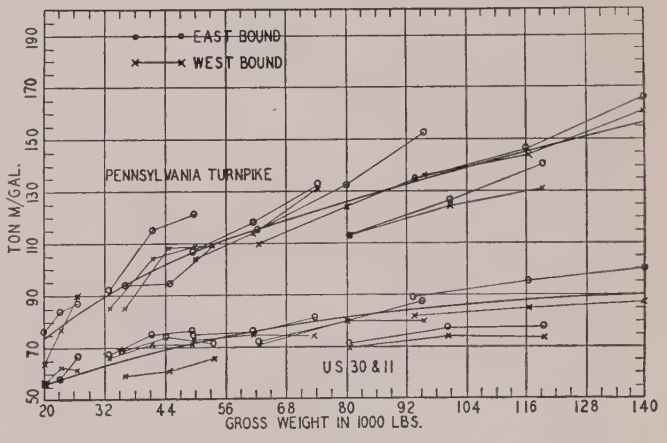


Fig. 15 – Effect of road and load on work done relative to fuel consumption

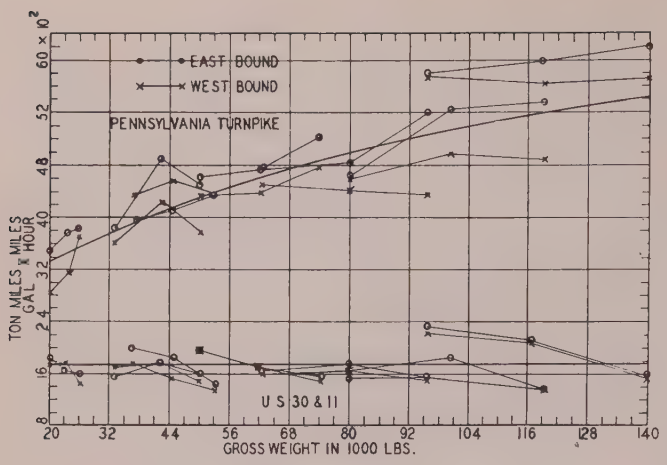


Fig. 16 – Effect of road and load rate of doing work relative to fuel consumption

and performance of the vehicles that use the highways. Ideally, such regulation should be uniform throughout the whole nation. It is also desirable that such greatly different objectives as license fees, and size and weight regulation, should not be confused. Neither should there be anything which tends to influence design in a manner that produces undesirable, inefficient, and unsafe vehicles.

When the regulations of the different states are compared for uniformity, it can be seen that there is a great deal that remains to be done. It is practically impossible to take a vehicle meeting the regulations of a specific state into a neighboring state.

In Pennsylvania, license fees are based on an arbitrary classification of chassis weights, and to make matters worse, the gross weight which each class is permitted to carry is also specified. Such regulations take out of the manufacturer's hands the rating of the vehicle he builds. While no one can deny the right of the states to determine the schedule of fees for vehicles registered in it, and while it may be argued that the powers of the state should be used to guard against vehicles being loaded beyond their capacity in the interest of safety, this has not been shown as a good way to accomplish these purposes.

The so-called bridge formula has either been written into or followed in establishing limits for vehicle gross weight. In its proper place and for the purpose for which it was originally set up, it was a good rule. However, the manner in which it is applied in many regulations results in making it impossible to build vehicles with the most desirable features.

The regulations are set up on overall length limit, which is a perfectly proper thing to do. Then the gross weight limit is determined by the distance between the first and the last axle in the combination, modified in most cases by the distance between each of the axles. This latter provision guards against a too large concentration of weight where axles are closely spaced. The other provision places a premium on the placing of the front axle as close to the front of the vehicle and the rearmost axle as close to the rear of the vehicle as possible. Locating the rear axle close to the back end is not the best selection, but placing the front axle so far forward reduces the weight which can be carried by it. Seeing that the front axle does rarely, and can only with difficulty be made to carry more than half the weight carried by the other axles, the slight variation produced by a change in location of not more than 2 ft is of no practical importance as far as either bridge or highway loading is concerned.

Forty-five feet is the length limit provided in many of the states for a tractor-semitrailer. Placing the axles as close to either end as possible gives a distance between them of approximately 40 ft. It is hard to understand why an arrangement which gives a distance of 38 ft should be penalized as much as 1500 lb.

In answer to the question, "What difference does it make?" it can be pointed out that locating the front axle back has been proved to produce better riding qualities and better steering. After all, the steering forces developed between the tire and the road are limited by the weight carried by the steering axle. Anyone who has experienced the lack of directional control on ice, where the steering forces are reduced almost to zero by the slippery road, can appreciate what this means.

Mechanical highway freight transport has just crossed its fiftieth birthday at the Centennial of Engineering. The contribution it has made and is making to our mode and scale of living is that it has brought to everyone's doorstep the means of sending and receiving the necessities which are beyond his capacity to produce for himself. In our modern life, urban, suburban, and rural, this means almost everything.

In these pages, the effort has been made to outline in a broad way rather than in detail, the need for and the place of trucks and highways in the nation's transport system, and also to trace the development of the vehicle.

Acknowledgments

Statistics used have been taken from sources that are listed in the footnotes.

Additionally, I wish to acknowledge the courtesy and help received from the Cummins Engine Co., the Budd Co., the Eaton Mfg. Co., the Diesel Division of the General Motors Corp., the Firestone Tire and Rubber Co., the Goodyear Tire and Rubber Co., the Diamond T Motor Car Co., the Mack Mfg. Corp., the White Motor Co., and the Columbia Terminals Co.

Engines to Digest the Vitamin-Enriched Fuel—Elpeegee

J. E. Glidewell, Hall-Scott Motor Division, ACF-Brill Motors Co.

This paper was presented at SAE National West Coast Meeting, San Francisco, Aug. 12, 1952.

TO use LPG efficiently as fuel for trucks, buses, or industrial engines, the compression ratio must be raised to take advantage of the fuel octane number. This point is explained in this paper, together with other pertinent facts relative to converting to LPG or using an engine already set up for the purpose.

The author also details the many safety factors which have been considered in the handling of LPG. Developments to date provide that, with reasonable care and attention, LPG is as safe as, if not safer than, other types of fuel in common use.

He touches on the economic advantages which can be realized now that LPG is in excess supply.

SINCE you are seriously considering the use of LPG as fuel for your truck, bus, or industrial type of engine, you are interested in its characteristics, handling, and use. You are quite right in adopting this fuel because of the possibility of getting increased power output, economy of operation, and lower cost maintenance.

First, considering the fuel itself, it is just one of the many products obtained from petroleum. LPG or liquefied petroleum gas is composed of those lighter cuts of crude oil which won't stay in an uncorked bottle. These constituents are chiefly those chemicals known as butane and propane, with some isobutane, possibly propylene, and smaller amounts of other "anes" and "enes" together with slight amounts of entrained oil, and so forth, which are present to justify use of filters and sediment traps. Liquefied petroleum gases are, as the name implies, normally gaseous at ordinary temperatures and are, therefore, most economically stored and transported in the liquid state under moderate pressures. LPG has been used as a motor fuel for the past 20 or 25 years, and was commonly known as commercial butane. Butane is normally a liquid below about 34 F at atmospheric pressure, while propane is a gas down to about 44 deg below zero. This means that a lower test tank can be used for butane; however, some propane must be used with it in order to bring pressures up to a safe value, especially in winter season use. For commercial butane which consists largely of butane, although propane percentage may run from 10% to about 50%, tanks of 125-psi pressure rating have been used. In the last few years, however, the constituent butane has been used more and more for such purposes as making synthetic rubber, improving the fuel known as gasoline, and so forth, and propane has been obtained in increasing quantities. Therefore, LPG is tending toward higher percentages of propane, which fuel having a vapor pressure at 100 F of

Fig. 1 – Typical LPG fuel tank with center external valve arrangement and guard

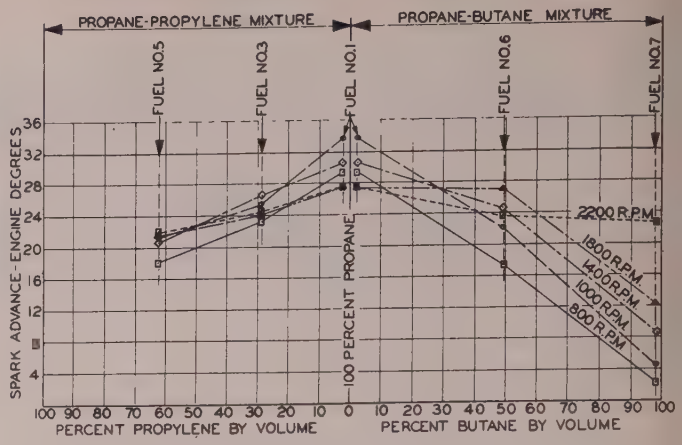


Fig. 2 – Trace knock comparison of LPG fuels – propane-propylene versus propane-butane mixtures

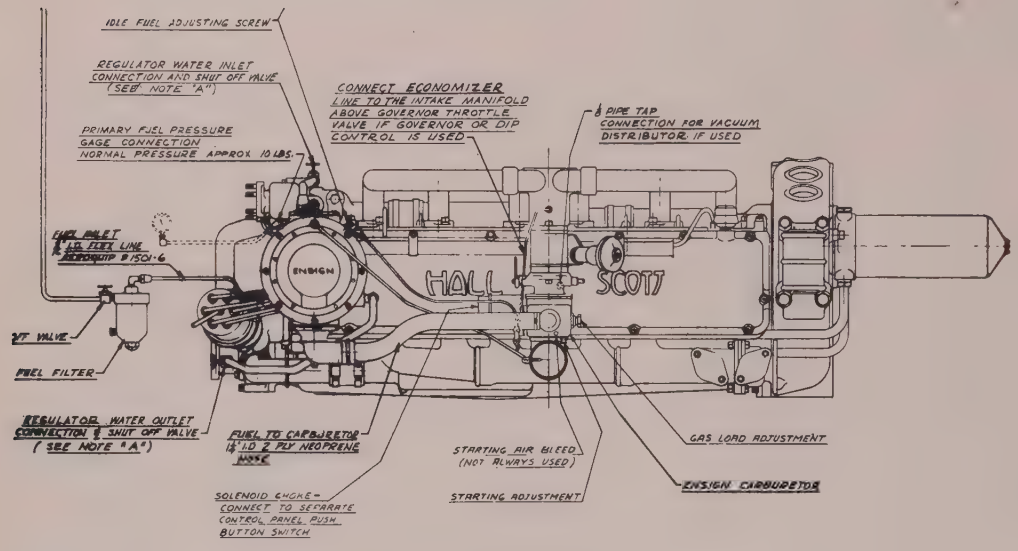


Fig. 3 – Equipment required on vehicle in order to use LPG fuel

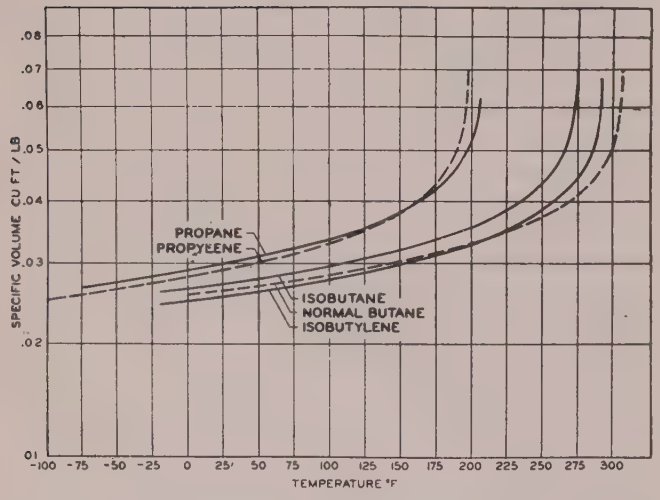


Fig. 4 – Liquid specific volumes of pure liquefied petroleum gases

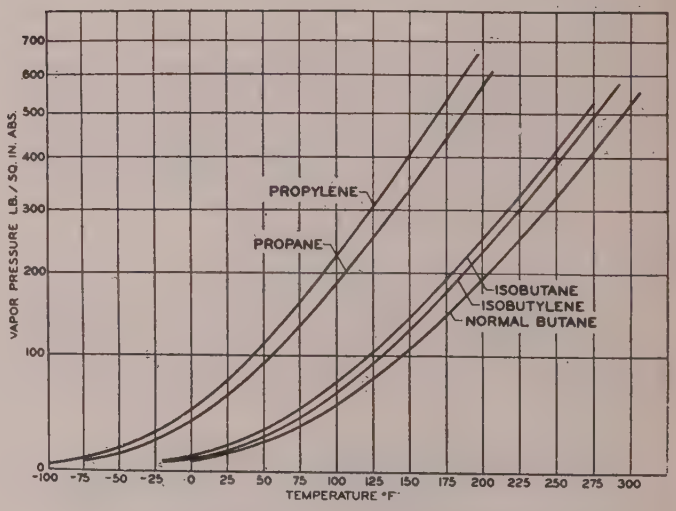


Fig. 5 – Vapor pressure of pure liquefied petroleum gases

about 180 to 190 psi requires tanks and containers of 250-psi rated pressure capacity. Such tanks are necessarily heavier than tanks of the same fuel capacity which are used for gasoline because of the higher pressures carried. Also, due to the higher pressures, the tanks are built with cylindrical section and rounded ends. (See Fig. 1.) The ratings, capacities, and pressures are regulated by National Board of Fire Underwriters and also by various state agencies. Such regulations are well enforced, especially on equipment such as buses carrying the general public. It is recommended that anyone considering LPG installations of any kind should obtain copies of not only the regulations published by the National Board of Fire Underwriters, but also of the regulations of the state in which an installation is to be made and used. There are also some interesting and informative handbooks sold by various organizations.[1]

It may be interesting to compare weights of propane and gasoline tanks both empty and loaded to capacity with their respective fuels. As an example, in Table 1 comparisons are made on fuel tanks used on ACF-Brill buses.

The capacity shown for the propane tanks is water capacity less 21% outage as required by the National Board of Fire Underwriters' regulations. Some states do not require such a high percentage of outage capacity. Some outage capacity is always required, however, to allow for possible liquid expansion, since LPG in liquid form has rather a high expansion coefficient, and an appreciable air space must be provided to allow for liquid expansion without rupturing the tank.

The figures in Table 1 show empty tank weights for propane ranging from 4¼ to 5¾ lb per gal capacity against only 1 to 1½ lb per gal capacity for gasoline. This weight advantage for gasoline diminishes for full tanks, however, due to the fact that the weight of propane is roughly two-thirds that of gasoline (the exact ratio being 4.23 to 6.23). The weights of full tanks are roughly 8½ to 10 lb per gal capacity for propane against 7¼ to 7¾ lb per gal for gasoline. In general, still larger tank capacities would shift the weight advantage more in favor of propane.

Table 2 shows a comparison of fuel density and heat content.

It is apparent from this table that although propane and butane have somewhat greater heat units per pound they have, being lighter in density, less heat content per gallon than the more ordinary liquid fuels. Since all these fuels are sold on gallonage basis, LPG must be sold for lower cost per gallon to be competitive regarding fuel cost per mile.

There have been various rumors circulated that LPG is scarce in supply and so any appreciable increase in its use as motor fuel would soon drive the price up out of all proportion to its advantages.

[1] Published by Jenkins Publications, Inc., 1709 West Eighth St., Los Angeles 14, Calif.

The Author

J. E. GLIDEWELL (M'29) has been chief engineer of the Hall-Scott Motor Division, ACF-Brill Motors Co., since 1947. He received his B.S. from the University of California in 1926. Mr. Glidewell was vice-chairman for the SAE Northern California Section for 1951-52.

Statistics issued by some of the large and reputable oil companies, however, seem to refute such rumors. It is reported that there is a great oversupply of LPG, especially in the oil-producing regions. This is especially so in the last two or three years due to conservation laws preventing waste of natural resources and also due to the fact that natural gas distributing lines are replacing the bottled gas requirements of many rural communities, thus leaving still more unused LPG. A possible basis for the rumor of scarcity mentioned above is that distribution of this fuel is not yet worked out for wide coverage, but is being taken care of as fuel usage creates a demand.

Historically, the consumption of LPG has been chiefly for heating and cooking in more or less rural districts. This type of market has caused a seasonal variation in consumption, the demand being larger in the winter season and falling off during the warmer seasons. This condition in the past has caused seasonal price fluctuations which has worked a hardship on use of the fuel in trucking, for instance, which in most areas is a year-round operation. Recently, however, there has been a very great increase in the use of LPG in farm equipment, which condition increases demand during the warmer seasons, thus tending to balance the fluctuation due to seasonal use of fuel for heating pur-

Table 1 – Comparison of Tank Weights

Bus Passenger Capacity	Tank Capacity, gal	Weight Empty, lb	Weight Full, lb	Lb per Gal, Empty	Lb per Gal, Full
		Gasoline Tanks			
36 city	88	125	681	1.42	7.75
44 city	110	151	847	1.37	7.70
41 intercity	125	123	914	0.98	7.30
		Propane Tanks			
36 city	95	400	803	4.21	8.46
44 city	95	400	803	4.21	8.46
41 intercity	115	650	1138	5.65	9.89

Table 2 – Fuel Density and Heat Content

Fuel	Lb per Gal	Btu per Lb	Btu per Gal
Commercial propane	4.23	21,500	91,200
Commercial butane	4.84	21,200	102,500
Gasoline	6.23	20,300	126,500
Diesel	7.00	19,500	136,000

Table 3 – LPG Test Samples

Fuel Samples	No. 1	No. 3	No. 5	No. 6	No.7
NGAA Vapor Pressure at 100 F, psig	191	215	210	120	42
Ethane, % by vol.	1.8	4.4	1.7	2.8	....
Propane, % by vol.	93.1	64.0	34.1	44.2	....
Propylene, % by vol.	2.8	28.7	62.1	....	....
Butanes, % by vol.	2.3	2.9	2.1	49.5	98.6
Pentanes and Heavier, % by vol.	....	?	....	2.1	1.4
Unsaturates, % by vol.	....	?	....	1.4	....

Table 4 – LPG Fuel Samples

Sample No.	1	2	3	4	5	6	7	8	9	10
Propane	53.9	48.1	55.4	39.8	47.8	36.5	37.2	33.1	39.7	38.4
Isobutane	3.0	6.1	Trace	22.5	18.4	4.2	13.1	18.0	4.1	3.2
Normal Butane	42.7	45.8	42.8	37.2	33.5	56.3	49.3	48.3	55.6	57.5
Propylene	....	....	....	....	....	1.8	....	....	....	....
Ethane	0.4	2.0	1.7	0.2	0.1	0.6	0.3	0.6	0.3	0.6
Isopentane	....	....	0.1	0.3	0.2	0.6	0.1	....	0.3	0.3
Unsaturated Hydrocarbons	Nil	Nil	Nil	Nil	Nil	Nil	Nil	....	....	....

poses. Thus, a more constant market is being built up which should reflect in more favorable fuel prices because of easier production planning and fewer storage facilities.

If an operator can put in his own storage facilities, thus enabling him to buy direct from a refinery in appreciable quantity, a very favorable fuel cost can be readily obtained. Buying direct in quantity also insures getting fuel to more uniform specifications.

Butane has an octane rating of 90 while propane has an "octane rating" of 110 to 125. Therefore, a higher compression ratio can be used with propane which, other things being equal, should result in increased power output and improved fuel economy. The higher compression ratios, suitable for propane fuel, cannot be safely used unless it is assured that the fuel is actually propane, and not a mixture having a lower octane number rating. As a direct comparison, an engine setup for use of ethyl gasoline cannot be operated on regular gasoline without detonation and possible damage, if high load factors are imposed on the engine. An LPG fuel having a high percentage of butane will stand a considerable increase in compression ratio over gasoline, but propane will stand a still greater increase as indicated, of course, by the relative octane ratings. The terms butane and propane have in the past been rather loosely used, either name being applied to an LPG which may be composed of largely one or the other, and most likely both. However, in buying from a refinery or large dealer organization in appreciable quantity, fuel can be obtained to a specification desired, or recommended for the engine setup.

An alternative, of course, is to have the engine set up with compression ratio and spark advance suitable for about a 60-40 butane-propane, so that a wider range of LPG fuels would be used without fear of possible imperfect results on engine maintenance. This procedure is recommended particularly if, as in the case of truck use over long distances, fuel must be purchased at various sources. However, such procedure will not produce optimum fuel mileage, especially if straight propane might in some cases be used. This again would be similar to the case of using high octane gasoline in an engine of low compression ratio, in that the full potentialities of the fuel are not being realized.

Butane, commercial mixture, has a spark tolerance curve similar to gasoline in that the spark can be advanced with increasing engine rpm. With propane, on the other hand, the engine tends to knock at high speeds rather than at low speeds; in other words, it may be found, depending on compression ratio, and so forth, that propane can have more spark advance at lower speeds than in the high-speed range. Propylene is a constitutent of LPG, very similar to propane, and exhibits a similar spark tolerance characteristic. This constituent is not usually found in very large amounts in LPG as marketed in the west, although it may comprise 50% or more of LPG sold in the east. It is usually extracted from LPG for other uses, especially if the fuel is a refinery product. At the same compression ratio, propylene exhibits a greater tendency toward high-speed knock than propane. Reference to Fig. 2 will indicate the trend in relative spark advance characteristics. Data for this series of spark advance curves were obtained from test stand results on a 6-cyl 779 cu in. displacement Hall-Scott bus engine set up for 8.25 compression ratio, and run on the fuels shown in Table 3. These results indicate trends, although they are not claimed to be entirely comprehensive, since only one compression ratio was used.

Table 4 shows analysis of 10 LPG fuel samples taken at random from various trucks in the immediate vicinity of San Francisco Bay area. These were all taken during the spring season and no effort was made or intended to be made to find the origin of the various samples or to determine where they were marketed. Either sample might be called propane or butane at random, and usually is.

LPG fuel for use in internal combustion engines is, as explained above, supplied in liquid state under moderate pressure in containers, the construction of which is governed by strict regulations. In general, the liquid fuel is supplied to the engine through two stages of pressure regulation to a mixing valve or carburetor, where it enters the engine as a dry gas. Being a dry gas and not a liquid, it has no tendency to wash lubricating oil from cylinder walls, or to pass down by piston rings and dilute the oil in the engine sump. Therefore, due to this characteristic, the service life of cylinder bores and piston rings is extended a surprising amount. In many cases the mileage between overhaul has been

Fig. 6 – Close-up view of 300-psi tested propane tank showing valves and fittings

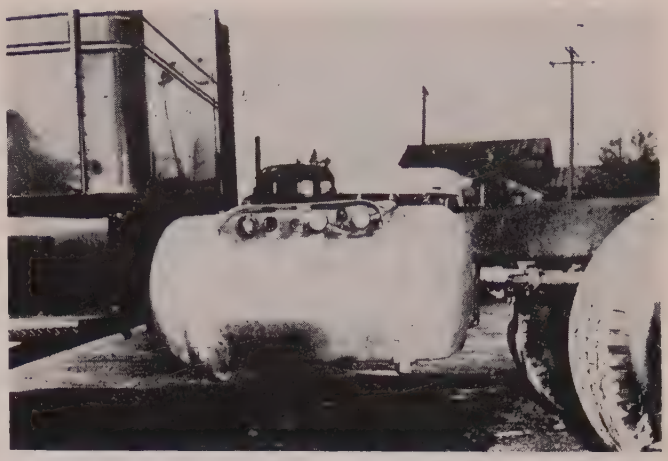


Fig. 7 – Kenworth truck-tractor, equipped for LPG, powered by Hall-Scott engine

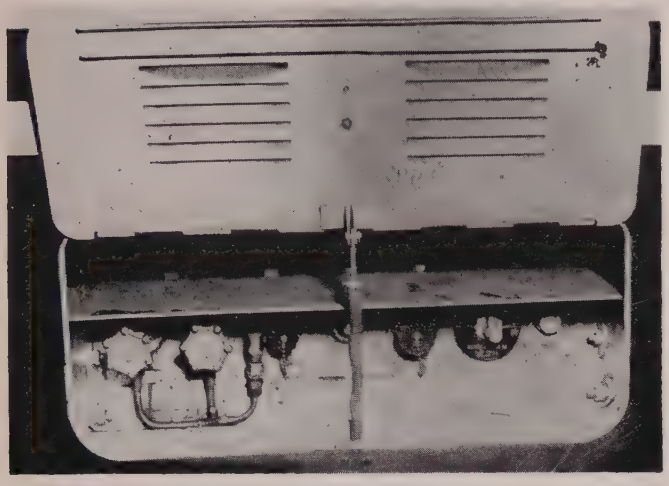


Fig. 8 – LPG tank and valve arrangement, rear coach view

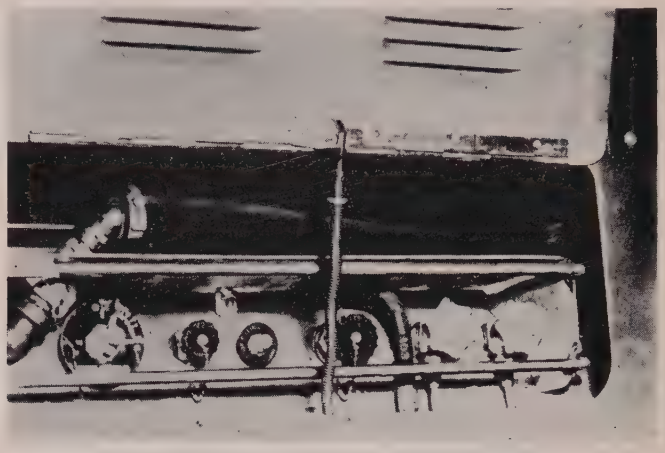


Fig. 9 – LPG fuel tank installation showing safety relief valve piped to outside

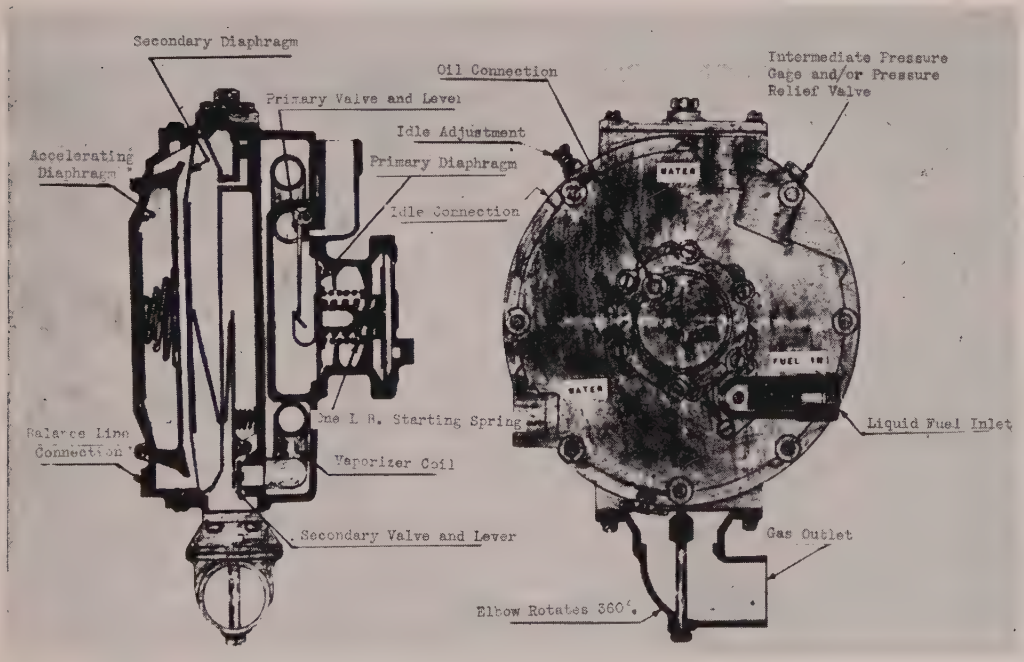


Fig. 10 – Sectional view of model S Ensign regulator

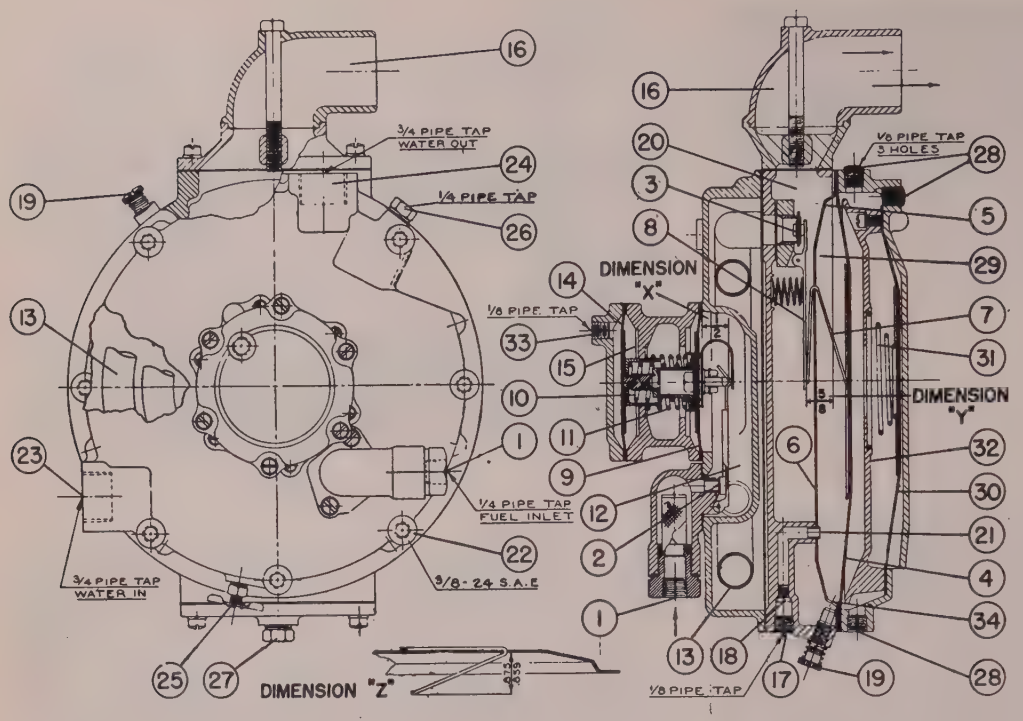


Fig. 11 – Ensign butane regulating unit

twice or three times that obtainable with ordinary liquid fuels.

Due to the higher compression ratio generally used, higher outputs and higher combustion-chamber temperatures can be expected, and so it is considered advisable to use exhaust valves and seat inserts faced with a hard material, and top piston rings with chrome face. This has been Hall-Scott practice.

Since LPG does enter the engine cylinder as a dry gas, then, of course, no intake hot spot is required. In fact, every effort should be made to separate the intake and exhaust manifolds, so that the intake charge cannot be heated. Running a cold intake manifold improves volumetric efficiency and hence engine output. Also, since distribution with a dry gas is not the problem that it is with vaporized fuel, the intake manifold can be somewhat larger in cross-section, thus further improving volumetric efficiency.

The equipment required on a vehicle in order to use LPG fuel, which differs from similar equipment used with gasoline, generally includes fuel tank or tanks, fuel filter, converter or regulator, carburetor, and piping for both fuel and water. (See Fig. 3.)

The fuel tank for LPG must have a designed pressure of 1.25 times the vapor pressure of the liquid carried, and must be hydrostatically tested to 1½ times their rated working pressure. Of particular importance is outage space of at least 10% of the tank capacity to allow for thermal expansion of the liquid fuel. This is usually accomplished by use of a 90% full valve. The Parkhill-Wade Co. makes a special LPG tank with a small tank of regulation outage capacity built inside it, and arranged so that, by use of a special fitting, during the filling operation the small tank is closed off and cannot receive fuel. After the tank is filled and a special filling fitting withdrawn, the small inner tank is then connected automatically to form the required outage capacity or expansion space. For relation of liquid volume at various temperatures, refer to Fig.

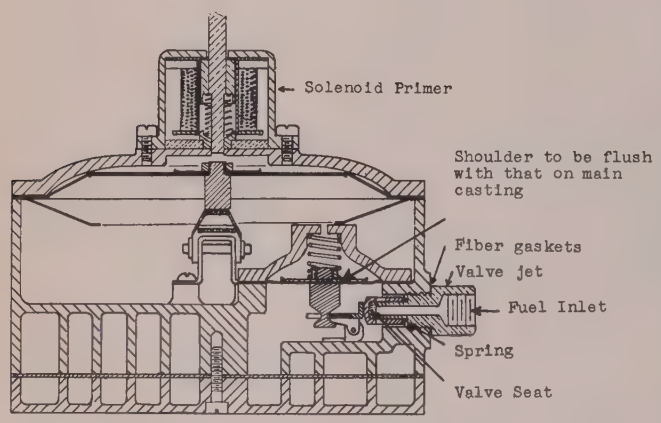


Fig. 12 – Cross-sectional view of high-pressure valve on Century converter

4 which is a graph obtained from Phillips Petroleum Co. Fig. 5 was obtained from the same source and shows relation of vapor pressure with temperature for the various possible constituents of LPG. A pressure relief valve must be provided which will start to discharge at 125% of the rated pressure of the tank, and be full open at 150% of the rated pressure.

Most installations have both a liquid fuel valve which draws from the bottom of the tank, and a vapor valve which draws from the top of the tank. Both the liquid fuel and the vapor are piped together after the valves, and a single line leads to the vaporizer at the engine. The vapor valve is not always necessary, as its principal function is to build up the line pressure gradually so as to prevent damage to the vaporizer. Once the line pressure has been brought up to the tank pressure, the liquid valve is opened and the vapor valve closed. Except on the lightest loads, the vapor valve will not discharge sufficient fuel to operate the engine. With a butane-propane blend, the more volatile propane, when taken through the vapor valve, will be used more rapidly than the butane, so that the remaining mixture will gradually become richer in butane. This drops both the tank pressure and the octane number, either or both of which might be disadvantageous in a given installation. When the fuel is taken through the liquid valve, on the other hand, the mixture delivered and the mixture remaining stay practically constant.

All tubing and fittings must be of high-pressure type and fuel lines must be kept away from exhaust lines. Fuel lines are ordinarily protected with a loom and clamped to the frame in the same fashion as gasoline lines. Refer to Figs. 6, 7, 8, and 9, illustrating various automotive-type tanks, their fittings and method of location on the vehicle.

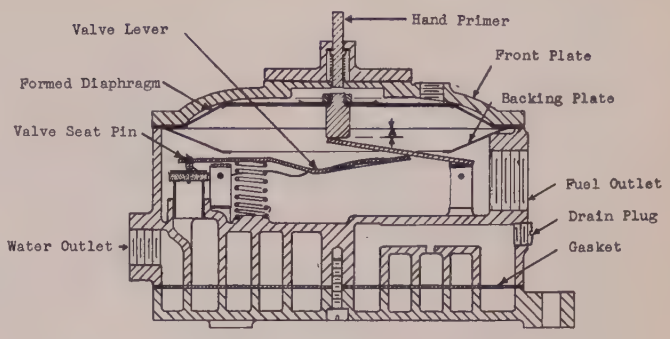


Fig. 13 – Cross-sectional view of low-pressure valve on Century converter

Several manufacturers make equipment for vaporizing LPG and mixing it with air for use in the engine. Details of their designs show more variety than is found in gasoline carburetors, but there are several features common to all vehicle converters and carburetors. See Figs. 10, 11, 12, 13, 14, 15, and 16.

The vaporizing function is handled by the converter, which usually combines two stages of pressure reduction through diaphragm-operated valves, and a heat exchanger which supplies the heat of vaporization of the liquid fuel from engine jacket water. The first stage of pressure reduction drops the pressure from tank pressure to a value of 2 to

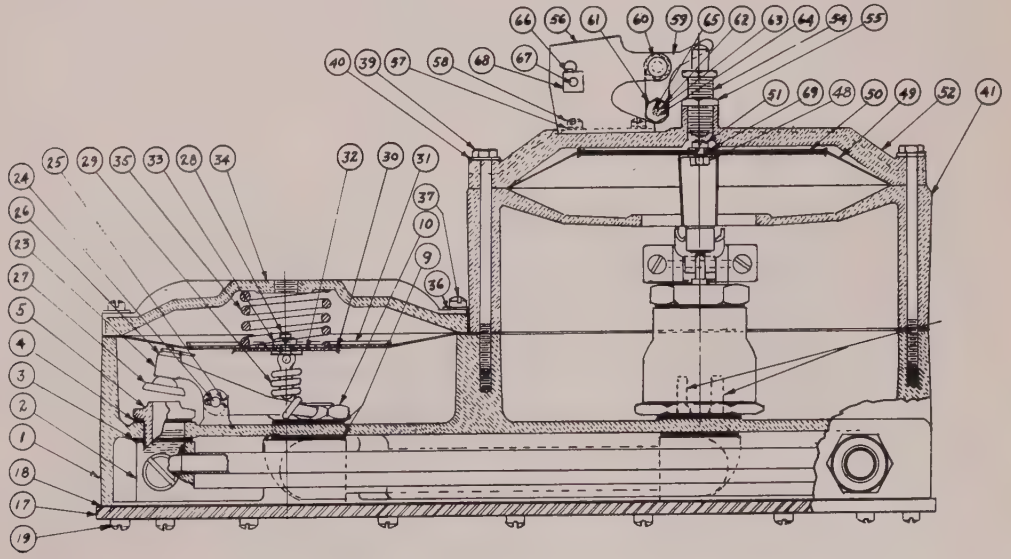


Fig. 14 – Details of design in Algas converter

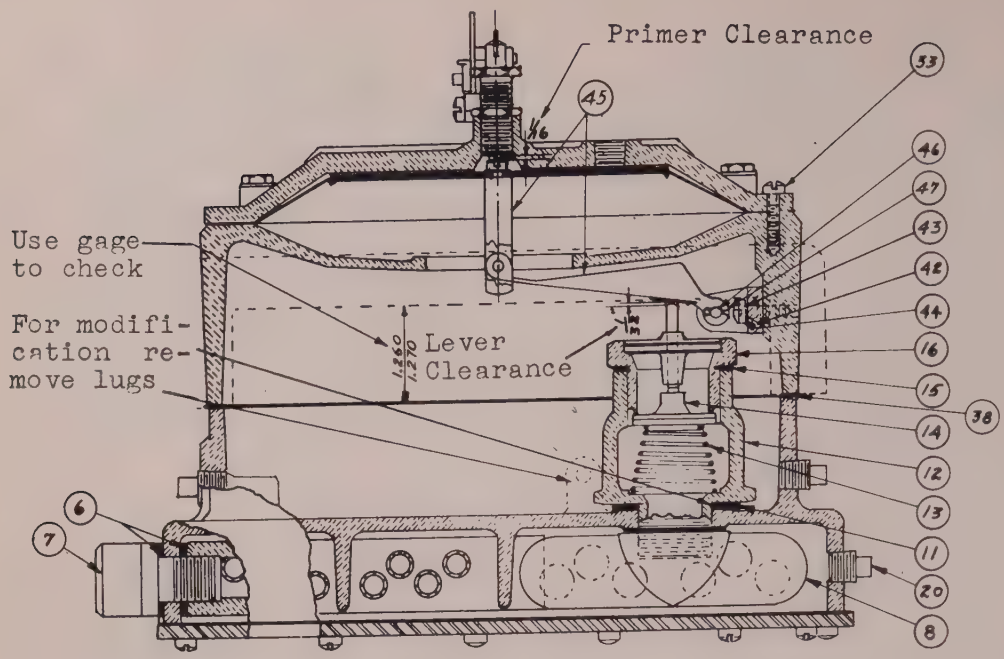


Fig. 15–Algas converter, showing details of design

12 psi, depending on the make. The liquid flashes to vapor in the primary valve. The heat exchanger is generally between the two states of pressure regulation, although one manufacturer heats the fuel before the primary regulator as well as between the two regulators. This is generally necessary only on larger engines, as the refrigerating effect of vaporizing the gas at high flow rates causes frosting at the primary valve. In all cases the fuel is delivered to the carburetor as a dry gas at less than an inch of water vacuum. A highly desirable feature is an atmospheric lockoff which tends to close the fuel line to the carburetor tighter as the pressure of the gas behind the valve increases. When this lockoff is a separate unit, mounted between the secondary regulator and the carburetor, the secondary regulator maintains a positive pressure of a few inches of water. When the lockoff is incorporated in the secondary regulator valve, the secondary regulator pressure is substantially the same as the supply pressure to the carburetor, less than an inch of water vacuum.

Neoprene or other synthetic rubber is used for the regulator diaphragms and for regulator valve seats, although some of the smaller units use metal needle valves in the primary regulator.

There is even wider variation in carburetor or mixing valve design than in converter design. Some designs depend on the carburetor for all fuel metering, while the secondary regulator furnishes dry gas at a constant pressure to the demand of the carburetor. In other designs, the metering func-

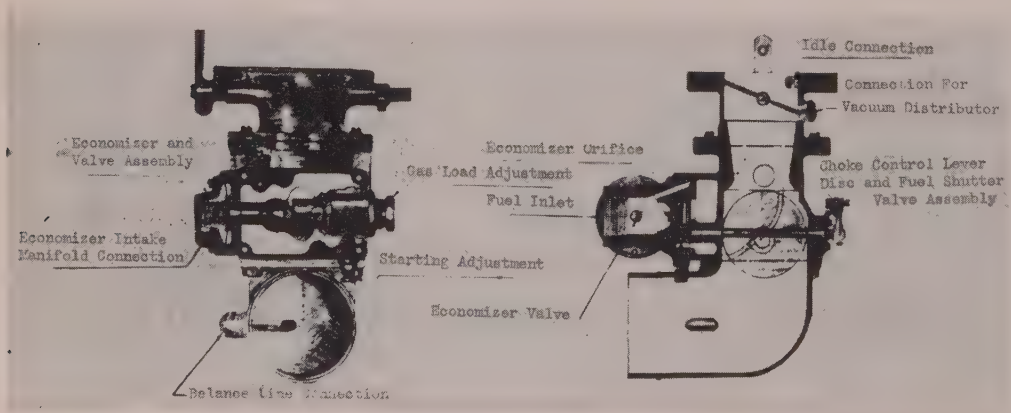


Fig. 16 – Illustrating features common to all LPG equipment

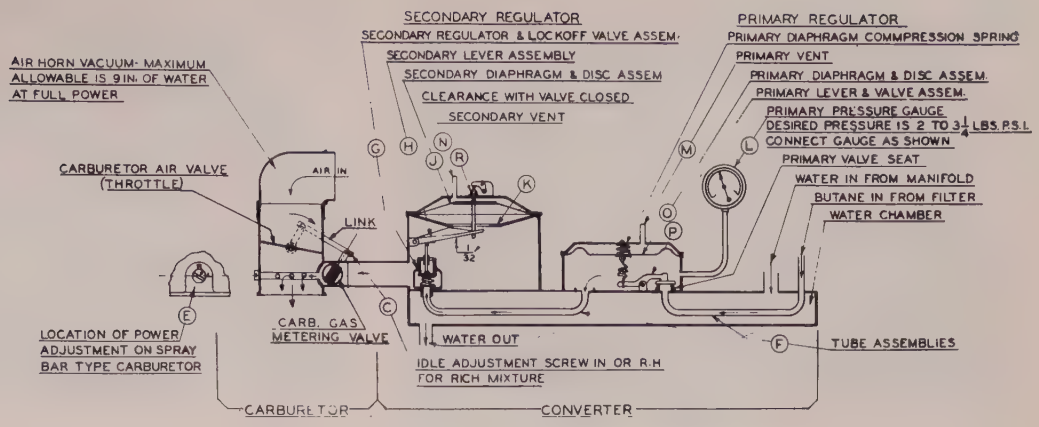


Fig. 17 – Diagrammatic illustration of fuel system

tion is partly handled by the secondary regulator through varying the pressure with demand. Some carburetors draw in the fuel at the venturi throat in the same manner as a gasoline carburetor and have air-fuel mixture in the idle tube which enters at the edge of the throttle plate in the usual manner.

One design has no venturi. The throttle plate is upstream of the fuel tube and the fuel is metered by a barrel valve linked to the throttle plate. On this carburetor, the engine is started with the throttle in the idle position. On models incorporating lockoffs, the lockoff must be opened manually or by a solenoid to prime the engine.

Some carburetors are furnished with a choke plate which is operated in the normal manner.

One design of carburetor has a choke plate which is closed and left closed for starting. The engine will idle indefinitely on this carburetor with the choke fully closed, but the choke must be opened before load is applied. This design incorporates a balance line from the carburetor air horn to the atmospheric side of the secondary regulator, to prevent enriching the mixture due to air cleaner restriction. Optional is an auxiliary diaphragm behind the secondary regulator diaphragm, the function of which is to enrich the mixture for acceleration. This auxiliary diaphragm is normally held against the cover by a very light spring, but upon a sudden depression in the air horn on acceleration, the spring force is overcome and the space between the diaphragms is reduced by movement of the auxiliary diaphragm rather than the secondary regulator diaphragm, which allows the fuel valve to remain open rather more than would be the case at the same air horn depression on steady load.

Economizers are used on the above design as well as most others, and function much the same as gasoline economizers. The common type is actuated by a small diaphragm on the side of the fuel tube at the carburetor. This diaphragm is spring-loaded and holds the power valve closed until a predetermined manifold vacuum, usually 5 to 6 in. Hg, is reached.

Maintenance expense on LPG carburetors and converters is very slight. Diaphragms seldom wear out, although dirt, sulfur, tank, or line corrosion

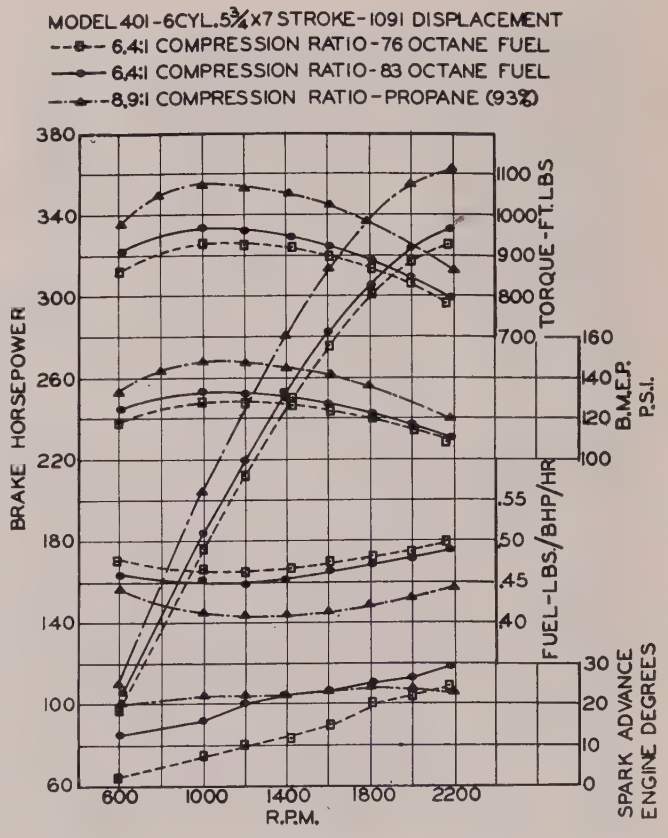


Fig. 18 – Performance curves of engine used in comparing gasoline and LPG output

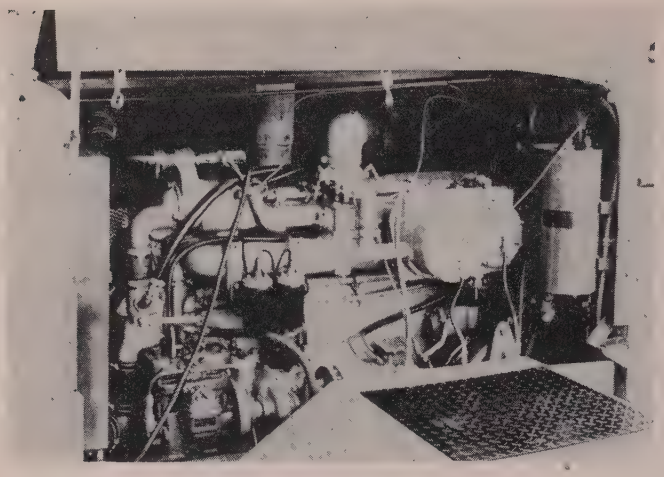


Fig. 19 – Oshkosh truck equipped to use LPG as fuel

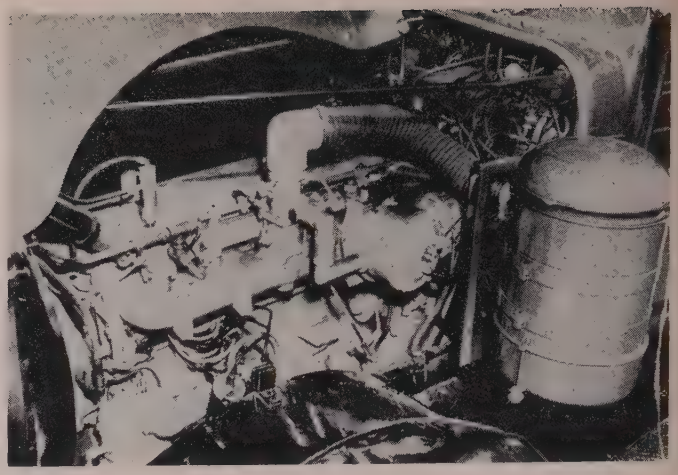


Fig. 20 – Kenworth truck-tractor, LPG-equipped

deposits sometimes build up on the valves and cause some internal damage. There is some wear in the levers and pivots of the valves. Springs sometimes break. Attention to the internal parts is seldom necessary between engine overhauls if effective fuel filters are used ahead of the converter. Kits containing valve seats, gaskets, diaphragms, and springs are serviced by the manufacturers of the equipment. Cleanliness and low sulfur content of the fuel as purchased is the best safeguard of the LPG equipment.

The fuel filter is usually a small unit with a shutoff valve incorporated in it, and it is used in the liquid line between the fuel tank and the regulator or vaporizer unit. It is a wise precaution to mount this filter just aft of the cab on the driver's side, so that in case of any emergency the driver can easily reach the shutoff valve to stop all flow of fuel from the tank.

Fig. 17 shows diagrammatically the fuel system.

Fig. 18 shows performance curves of a 6-cyl Hall-Scott truck engine used in comparing gasoline and LPG output. These curves serve to indicate increased performance that can be expected. All runs were on stripped engine. Note that specific fuel consumption is much lower on propane, but remember that specific consumption is in pounds and not in gallons. Checking specific consumption against density, propane would need to sell at around three-quarters the cost of gasoline in order to equalize cost of fuel per mile. Propane in many places is sold at a better relative price than this, so it is suggested that the territorial price setup be considered. This, combined with the other many advantages of LPG, may quite definitely prove to be of benefit.

Figs. 19, 20, 21, and 22 serve to illustrate some various installations using LPG as fuel.

And so the upshot of the whole deal is whether LPG should be used to fuel engines. If it is a question of converting an engine to a higher compression ratio, the first step is to consult the engine manufacturer, since he most likely has the parts to make the conversion or at least can explain

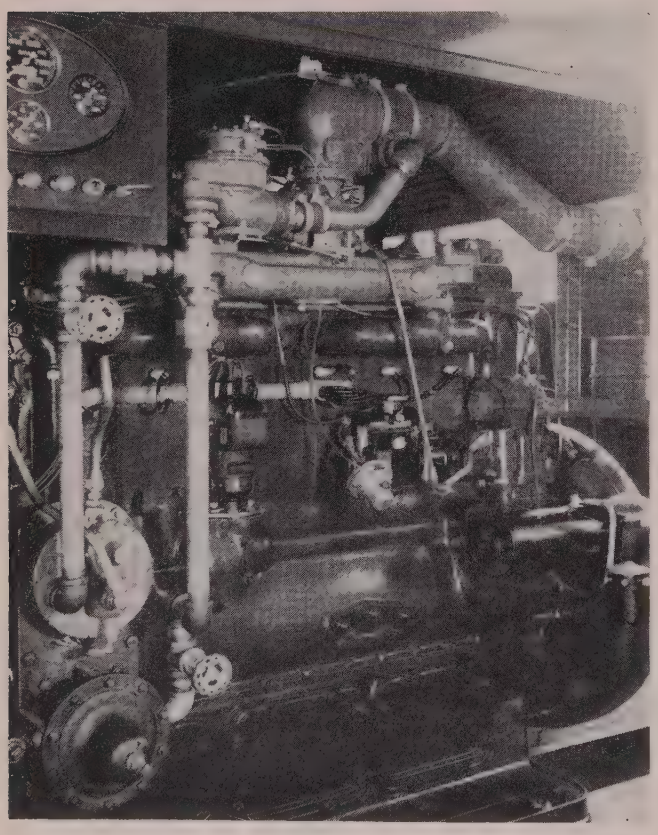


Fig. 21 – Installation for use of LPG as fuel

how to do it and how far it is safe to go. Naturally it would be best to buy an engine already set up for LPG, such as a Hall-Scott for instance. Then a study should be made of availability of LPG in the operating area, and of the cost relative to fuel now being used. Depending on the projected fuel usage per month or year, a check with the nearest large supplier on cost of tank car lots delivered will determine the economics of installing special storage facilities. Converting to LPG will give increased engine power, providing compression ratio is raised to take advantage of the fuel octane number. Maintenance costs should be considerably reduced, as proved by past practice. ("Ask the man who owns one.") Mileage between oil changes can be extended. It is usually considered advisable to use an oil grade lighter than otherwise, because with LPG there is no dilution; and, therefore, oil in the crankcase tends to get somewhat heavier with use. Spark plugs of a grade "colder" will probably be found best in operation depending on load factors, and so forth. In obtaining carburetion and converter equipment, there are several manufacturers of such equipment who could be consulted, including the Ensign Carburetor Co., Century Carburetor Co., Algas, and J & S Carburetor Co.

LPG is a good clean-burning fuel which is normally easy on an engine. It can be safely handled provided regulations governing its use are followed. It smells so nice (nonodorous) that the law requires that an odor be added to it so that it can be readily detected if a leak occurs. The vapor is heavier than air and therefore in still air it tends to settle in any sump or low area, so use your nose and a ventilation draft instead of a match to check any leaks in closed buildings, and so forth. Because of the heavy tanks and fittings required, no fuel is lost through evaporation. The tanks are strong enough to go through wrecks or collisions without being ruptured, with resulting addition of fuel to a possible fire. If a line or valve should be open and the escaping fuel be ignited, it will burn as a localized torch without exploding the tank. Regulations call for excess-flow valves which automatically shut off all flow of fuel from the tank if, due to some emergency such as a line break, the flow is suddenly increased over the designed flow capacity. All tank fittings should be protected against being knocked off, either by a protruding baffle arranged to shield them, or by having them set into a recessed area in the side of the tank. Fuel tanks should, of course, be mounted so that all valves and fittings are readily accessible, and should be electrically grounded. A fire extinguisher in working condition should be conveniently placed. Reasonable care and attention makes use of LPG as safe as, if not considerably safer than, other types of fuel in common use.

LPG is a most excellent fuel and is in excess supply. An engine need not be of excessively heavy construction to use it so some payload advantage may be realized. Distribution of the fuel over wider areas is being taken care of and expansion of coverage can be accelerated with demand of consumers. Therefore, the possible use of LPG should be carefully investigated by anyone operating trucks, farm equipment or industrial engines. There is reason to believe that the use of LPG as a motor fuel will gather momentum in the future as its virtues become better known, and it becomes more readily available on a widespread basis. Improvement of fuel quality and uniformity is also helping to hasten acceptance on a wider scale. All these developments, however, must go hand in hand, as they are interdependent.

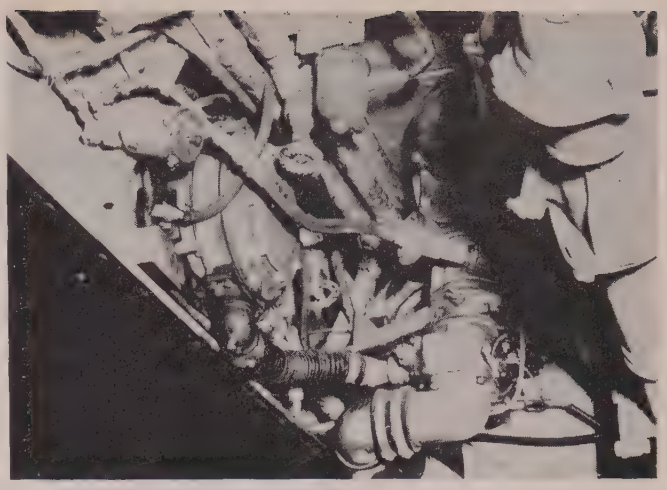


Fig. 22 – LPG engine installation under floor, showing view from under transmission, forward and to left side of coach

ORAL DISCUSSION

Gilbert Way, Ethyl Corp.: When using LPG, can the heat of vaporization be used as a refrigerant in trucks?

Mr. Glidewell: A small panel truck was equipped with a pipe running through the panel body where the LPG was allowed to vaporize, and the latent heat of vaporization was utilized as a refrigerant. Some measure of success was obtained in reducing the temperature inside the panel. However, if this application were to be tried in a large installation, such as a transit bus, the amount of refrigeration obtained by the vaporization of the fuel that is burned, would be insignificant. In other words, the amount of fuel that is burned controls the amount of energy that is available for cooling; and if the unit is large, the temperature drop will be small.

Mr. Way: Under what conditions were the spark advance curves obtained?

Mr. Glidewell: The spark advance curves were obtained at 8.6 compression ratio. The curves were obtained at the incipient ping point and not at maximum power.

Hydraulic Torque Converter—

FROM the outset of the use of the hydraulic torque converter in excavators, tractors, earthmoving machinery and off-highway trucks, it was generally conceded that the converter would help start heavy loads smoothly, reduce shock and torsional loads, and eliminate stalled and lugging engines due to overloads. This understanding and reduction of operator effort led to the wide acceptance of the converter drive by the earthmoving industry, and ore and coal mine operators all over the country.

At the same time it was indicated that the converter drive was contributing directly to longer engine, transmission and differential life by the elimination of shock loads which many operators claim were responsible for a great percentage of the transmission, driveline, and rear axle failures.

Hand-in-hand with speculative converter characteristics and reports of longer life for its power-train components, Allison began to accumulate a wide experience in building many different models of hydraulic transmissions for both commercial and military applications. After one of the first

MANY recent design improvements in hydraulic transmissions have followed the trend toward longer life of power-train components established by the use of a torque converter. The need for actual data relating to the characteristics of torque-converter operations led to the series of field tests reported in this paper. Off-highway trucks used in earthmoving, coal, and ore hauling were made available for the tests, together with experienced operators. Measurements of driveline stresses were recorded for trucks equipped with converter drives, and for those having mechanical transmissions.

Comparisons indicated that, with a converter, equipment can be operated at a higher load factor in order to get more work done, and also that the equipment will have a greater life factor. These two factors will result, of course, in reduced operating costs to the user.

The Authors

R. M. SCHAEFER (M '29) is manager, Transmission Engineering Department, Allison Division, GMC. He is in charge of both commercial and ordnance transmission programs. Mr. Schaefer received his mechanical engineering degree from the University of Munich.

J. A. WINTER is senior project engineer, Research and Development Section, Transmission Engineering Department, Allison Division, GMC. He joined General Motors in 1934. Before obtaining his present position he was assigned to the General Motors Aircraft Development Section as development engineer.

Its Effect on the Power Train

R. M. Schaefer and J. A. Winter, Allison Division, GMC

This paper was presented at the SAE National Tractor Meeting, Milwaukee, Sept. 9, 1952.

hydraulic torque converter transmissions had been in service for several years with good results, it was felt that the capacity of this transmission could be increased. The present rating permits its use in connection with 50% larger engines without experiencing any difficulties such as shorter life and increased failures due to excessive stresses.

This experience indicated that the normal load and stresses on the parts of hydraulic transmissions could be increased over the values previously used due to the use of the converter in the power train.

Even with operator's records indicating a trend towards longer life of power-train components when using a torque converter, and our design and field experience in hydraulic transmissions, there was still felt by everybody directly concerned with the use of transmissions, the need for actual data relating to the characteristics of torque converter operation.

With this interest established as a subject for the 1952 SAE National Tractor Meeting in Milwaukee, the impetus necessary was provided to develop and carry through a series of actual field tests. In consideration of the accessibility of the transmission and the entire power train, it was decided to conduct this test program on off-highway trucks used in the earthmoving, coal, and ore hauling industry. In order to accumulate the information desired, several operators and users of this equipment cooperated very generously. They agreed to make their equipment available in order to measure driveline stresses in the power train on identical trucks using both mechanical and converter drives operating over similar haul roads.

Also included in the plan was an investigation into the gear stresses of various common transmissions used in off-highway trucks and a comparison of the design stresses with those to be obtained from the field tests later on.

In addition to measuring the driveline stresses and comparing them to design specifications, operators further cooperated by making their operating records available on the maintenance of the power-train components in both mechanical and converter-equipped trucks which they had been operating over long periods of time.

Test Equipment Used

A method was needed for determining power and torque distribution in mechanical and torque-converter power trains used in off-highway trucks under field conditions, which could be used without making changes in the original power-train components thus changing their characteristics. The ordinary slip-ring collector had many undesirable characteristics and the mercury-bath collector was, therefore, selected. With this in mind, it was decided to follow the procedure outlined by D. E. Burroughs, Purdue University, and use the mercury-bath commutator in combination with strain gages and suitable amplifying and recording instruments. Burroughs' method allows the original drivelines to be used without change except for the addition of strain gages and the mercury-bath commutator.

For our use it was necessary to design, develop, and build our own commutator, because the Burroughs' commutator was only 1½ in. in diameter and end-mounted on shaft. Shafts which were to

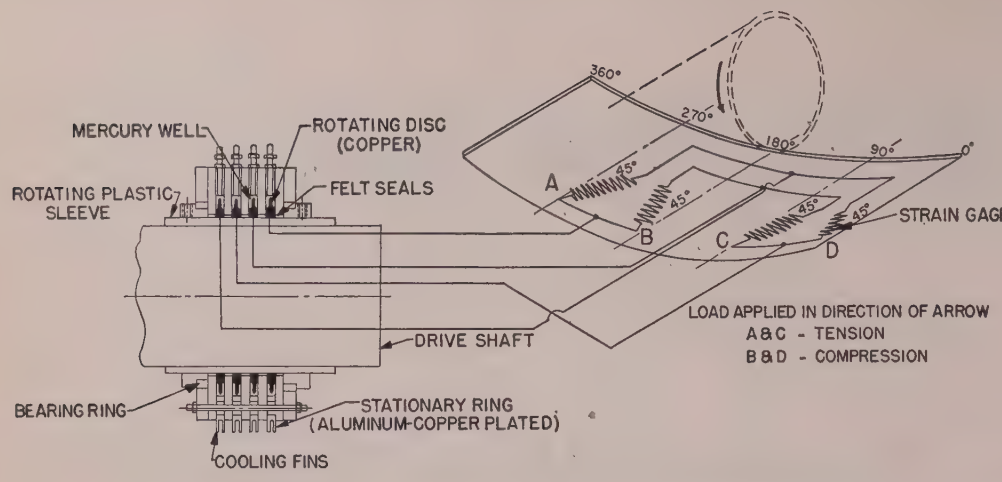


Fig. 1 – Schematic wiring of strain-gage bridge arrangement and mercury-bath collector

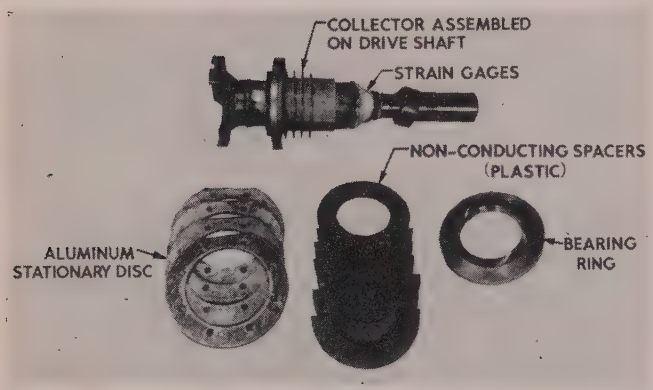


Fig. 2 – View of mercury-bath collector parts

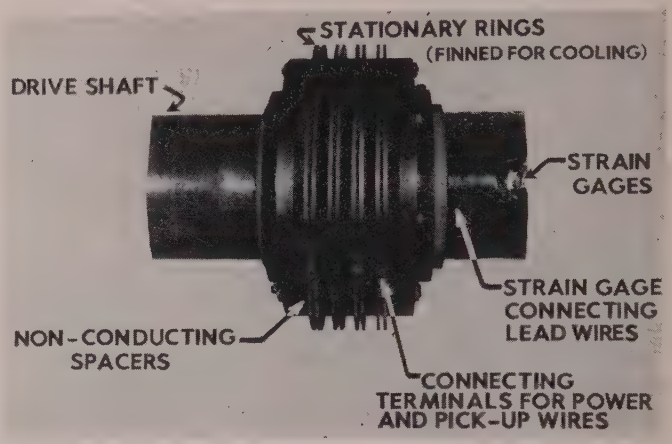


Fig. 3 – Mercury-bath collector ready for truck installation

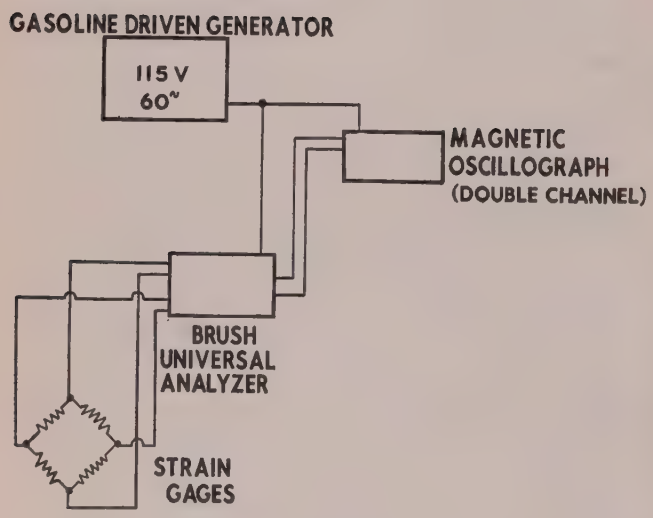


Fig. 4 – Schematic wiring diagram used in field test

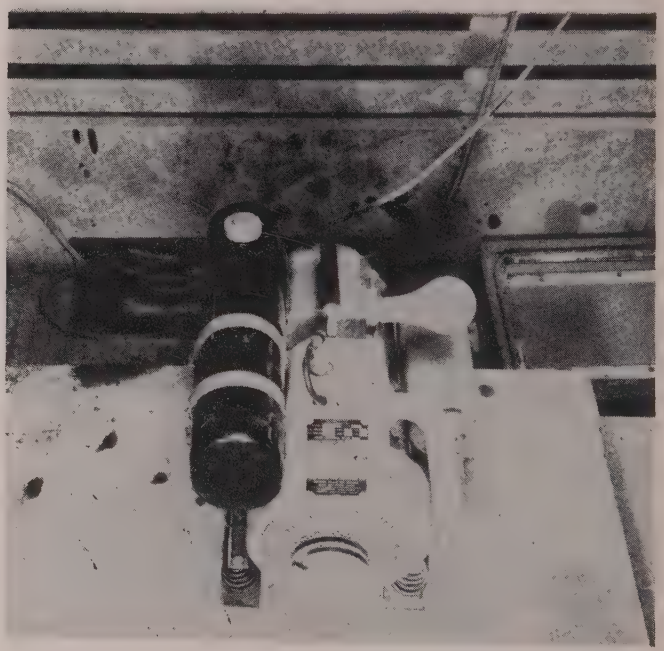


Fig. 5 – Gasoline-engine-driven generator mounted on truck

Fig. 6 – Installation of stationary collector in truck driveline

be used in the equipment indicated above were 3½, 4, and 4½ in. in diameter and naturally the pick-up would have to be large enough to fit these shafts.

When the first commutators, built for this program, were tested, they would not stand up because of high-temperature problems resulting from the spinning of large discs in mercury. So it was necessary to redesign the original pick-up to incorporate cooling fins on the stationary ring to help reduce the temperatures when in operation. Also, a new kind of high-temperature cement had to be found for assemblying the commutator parts together.

Along with the high-temperature difficulties, the commutator cells had to be sealed to prevent cross-leakage of mercury which resulted in shorting out the electric circuit. A felt seal was developed to overcome this difficulty.

Fig. 1 shows a cross-section of the mercury-bath collector mounted on a 4-in. diameter shaft, and the strain-gage bridge located on the projected area of the shaft, with the wiring connections to the collector.

The collector was made with four cells to accommodate the four lead wires from the bridge. Contact was made through a rotating disc and a stationary ring which were in contact with a mercury well. Plastic was used to make the nonconducting portion of the cell, with copper discs and aluminum rings for the conducting elements. The aluminum rings, which were copper-plated, and copper discs, were amalgamated with mercury to give more stable resistance characteristics.

The bridge arrangement was located to measure the compressive and tensile strains due to torsion in the shaft. The strain gages were placed on the shaft at a 45 deg helix with the centerline of the shaft. The effect due to bending can be cancelled by using four active gages on the shaft. This type of arrangement makes the setup four times as sensitive to torsional strain. The gages are connected to form a 4-arm bridge. The two gages lying on the same helix angle and displaced radially by 180 deg form opposite arms of the bridge.

Fig. 2 shows an exploded view of the commutator parts and how they appear when assembled to the shaft. Fig. 3 is a close-up of the commutator assembly ready for truck installation.

A schematic wiring diagram of the truck installation appears in Fig. 4. The bridge is connected through the mercury-bath commutator to an amplifier and a direct-inkling oscillograph. Power for operating the various instruments is obtained from a 115-v, 60-cycle gasoline-driven generator which is mounted directly on the truck (Figs. 5 and 6).

The amplifier is a self-contained Wheatstone bridge and voltage amplifier operating with 120-ohm strain gages. The output is designed to operate a penmotor and will record uniformly from d-c up to 100 cps (Fig. 7).

The magnetic direct-inking (double channel) oscillograph permits instantaneous recording of the shaft loading in torque. The recordings are made

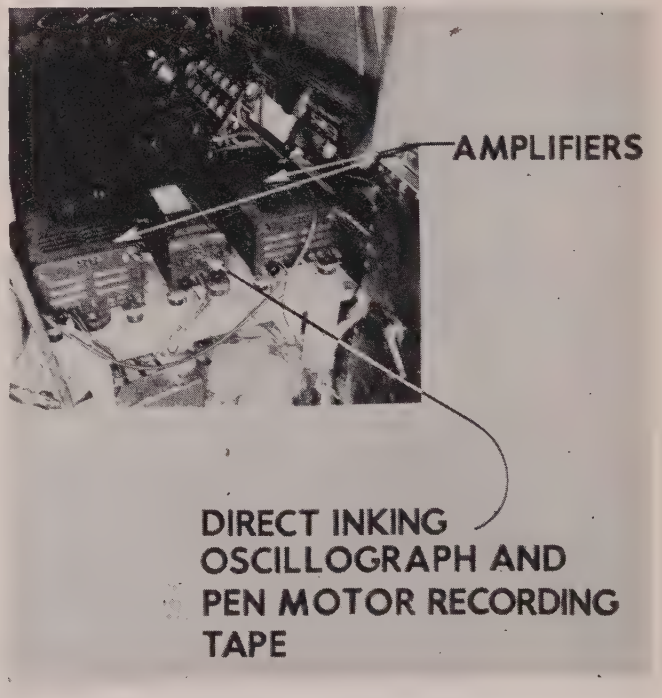


Fig. 7 – Installation of instruments in truck cab

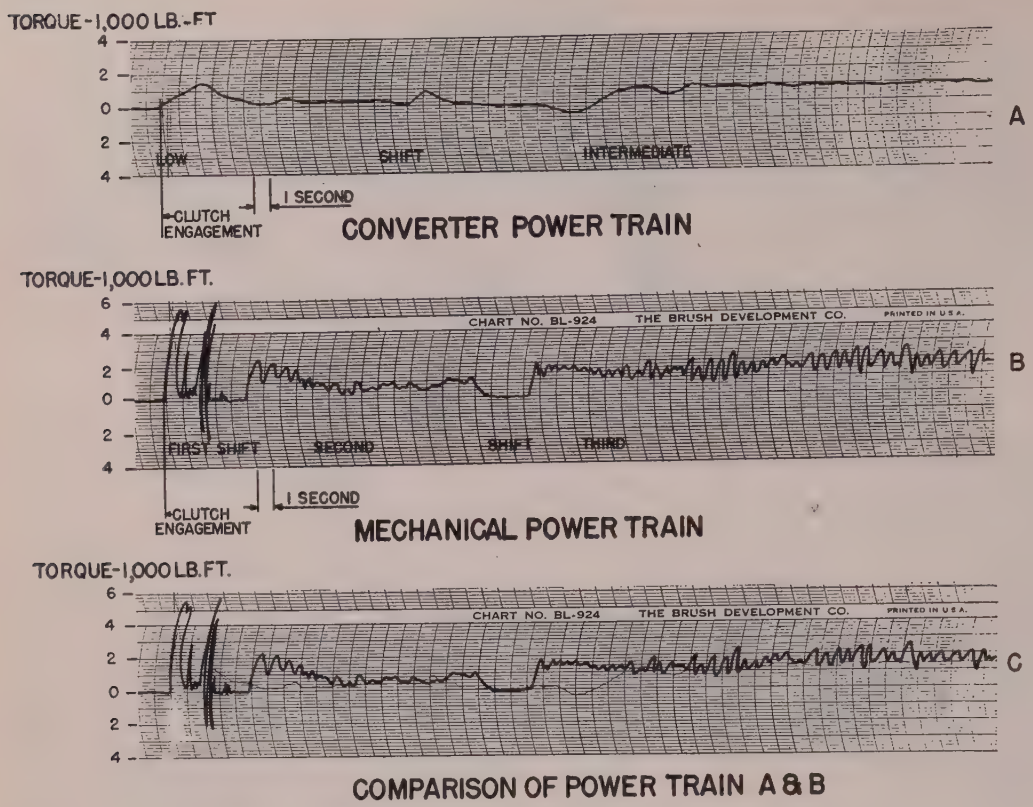


Fig. 8 – Comparison of converter and mechanical power-train reactions when starting a regular run (output shaft-transmission to final drive)

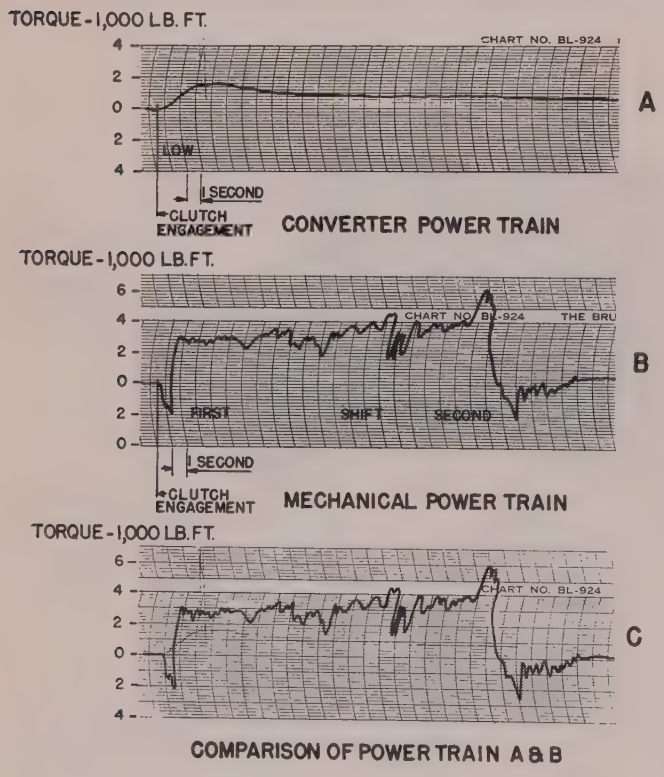


Fig. 9 – Comparison of converter and mechanical power-train reactions when starting in mud and sand (output shaft-transmission to final drive)

on chart paper $3^{15}/_{16}$ in. wide. A synchronous chart drive motor provides paper speeds of 5, 25, and 125 mm per sec (Fig. 7).

Tests

Fig. 8 is a reproduction of the test recordings of a truck equipped with a torque converter, and of another having a mechanical transmission, as they started a regular run over the same terrain. The trucks were identical in capacity and power except for their power trains. The torque converter recording (A) represents the clutch engagement as the truck moves away from the shovel. The next recording (B) shows what happens when the mechanical unit starts away from the shovel. As a comparison, C recording is a record of the torque converter data superimposed on the mechanical data.

Fig. 9 represents another test run of a truck equipped with a mechanical unit as the driver engages its clutch to start the load moving away from the shovel. This start is in mud and sand, and represents severe operating conditions for any vehicle.

Fig. 10 is a recording of a mechanical unit illustrating resonance during a shift from low to second gear. It also shows the stress variation in the power train as the truck moves away from the shovel to the dump.

Referring to Fig. 11, recording A shows the

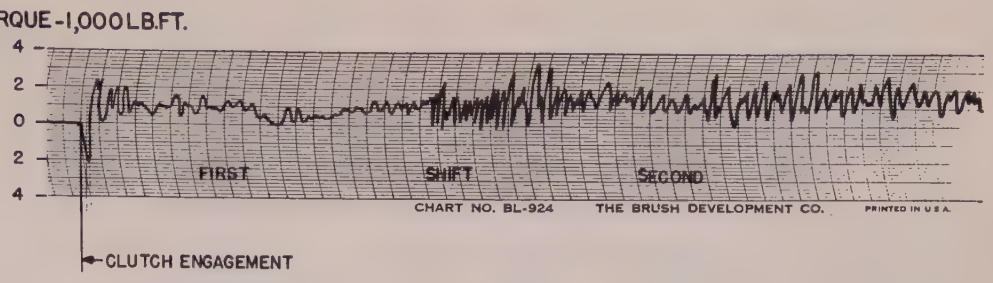


Fig. 10 – Resonance in mechanical power train when shifting (output shaft-transmission to final drive)

torque distribution and the characteristics of the torque converter while operating over a cross-country run. Recording B, which represents the mechanical unit, shows the torque distribution and stress variations while traveling over the same ground. Recording C is again a comparison of recordings A and B. The first half of the recordings, A, B, and C represents the beginning of the run while the last half indicates the finish.

Load variation of a torque converter unit is demonstrated in Figs. 12, 13, and 14. Here the recordings are identified with the test run in order to illustrate how the converter torque distribution reacts to the changing terrain. These recordings show pulling, coasting, and shifting, as the vehicle moves along over its course.

An examination of the tests of the mechanical- and converter-equipped trucks indicates that the results can be reduced to the following kinds of loads:

1. Starting – When the conventional trucks were started with heavy loads, the clutch engagement shock load was approximately 4.28 times the converter starting torque in a similar power train. In Fig. 15, A illustrates the comparison of converter and mechanical units when the clutches were engaged to start the vehicles. It was also observed that the shock loads included load reversals during the start.

2. Shifting – B (Fig. 15) shows what happened during a quick shift after starting a heavily loaded truck equipped with a clutch and multispeed transmission. The shock loads were approximately 4.25 times the smoothly applied torque of the converter

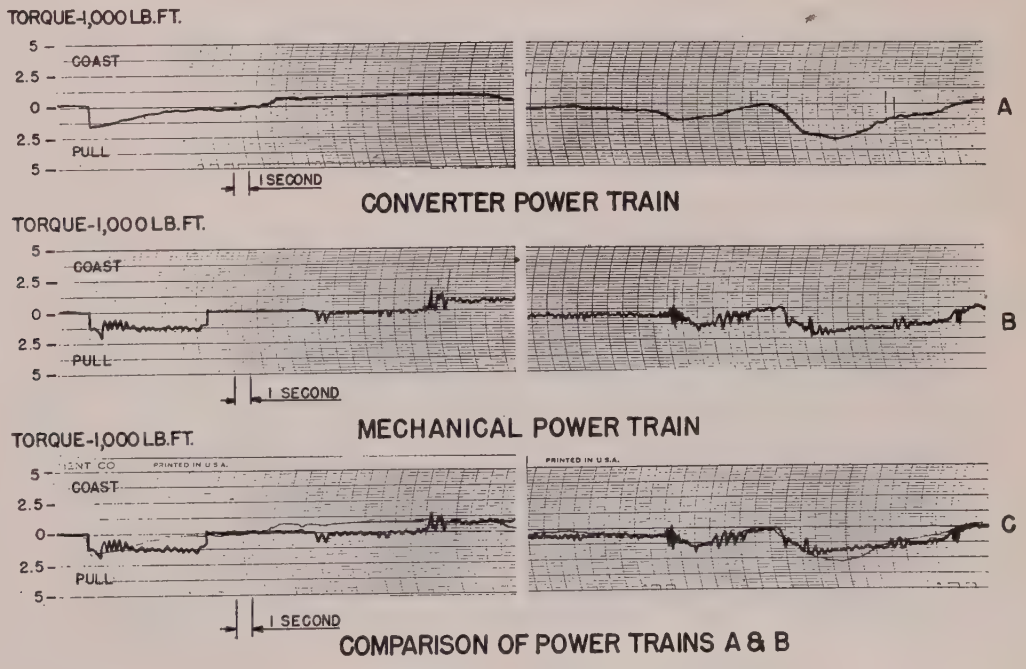


Fig. 11 – Comparison of converter and mechanical power-train reactions during cross-country run (output shaft-transmission to final drive)

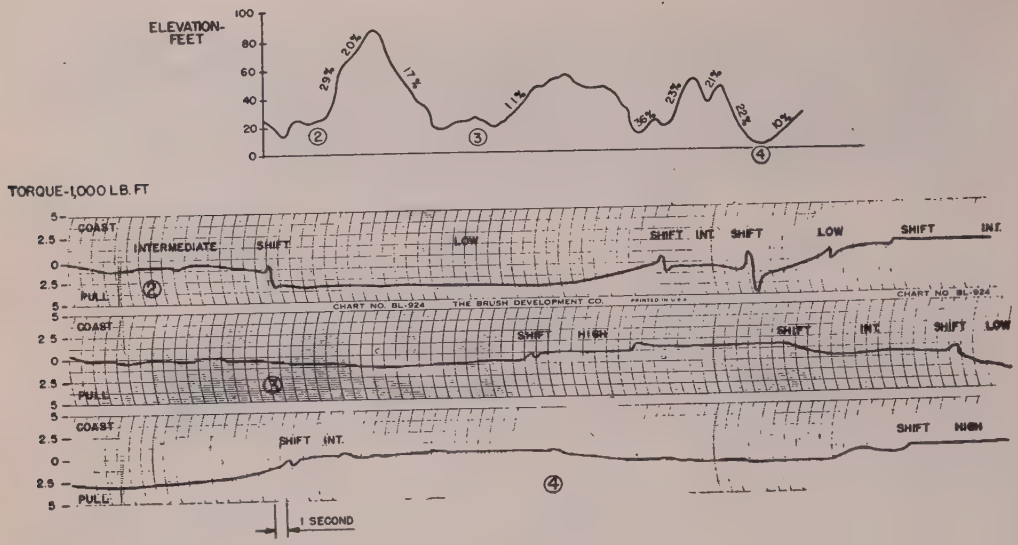


Fig. 12 – Torque distribution of converter-equipped truck operating over changing terrain (output shaft-transmission to final drive)

after the shift from first to second gear.

3. Load Variation – These reactions were due to the torque variation under normal operation and shifting of the trucks. C (Fig. 15) shows a comparison between mechanical and converter units during vehicle operation over a given haul cycle. Based on an average mean load, the mechanical unit indicated about a ± 28% stress variation during its normal operating cycle while the converter unit showed a ± 14% stress variation. And during average shifting operations for both units, the load variation based on average torque of the mechanical unit was approximately ± 67% compared to the ± 25% of converter load variation. (See D, Fig. 15.)

Transmission Design Stress

It has been mentioned previously that an investigation into the working stresses of various transmissions currently used in off-highway trucks, was planned for the purpose of determining a relationship of design stresses between transmissions used

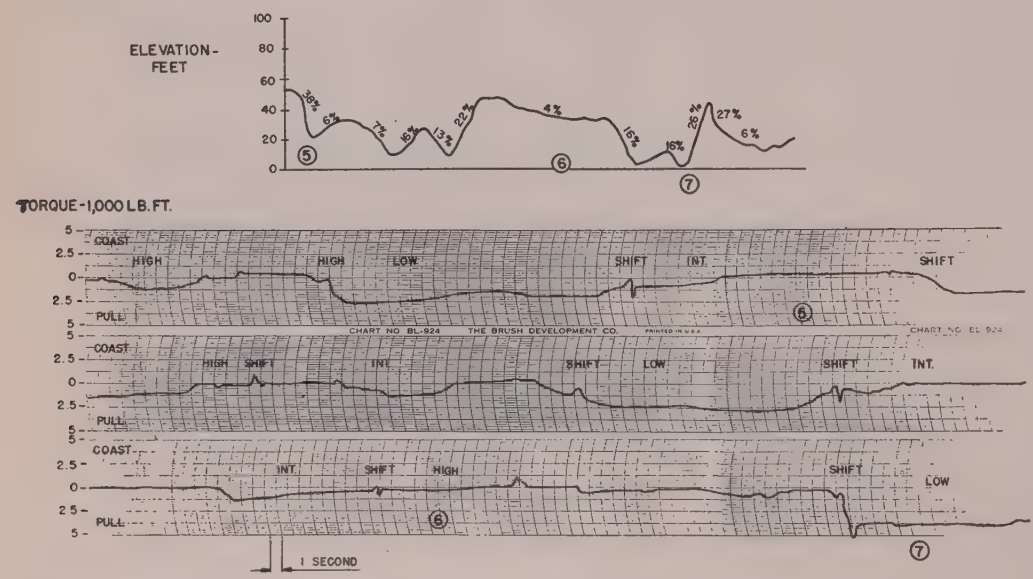


Fig. 13 – Torque distribution of converter-equipped truck operating over changing terrain (output shaft-transmission to final drive)

with mechanical drives and those used with torque converters.

The data used for mechanical transmissions were based on the information we could obtain in the field, and unfortunately, due to lack of time, has not been correlated with their manufacturers.

The relationship between the two types of transmissions is shown in Fig. 16. The average design stress is represented for two types of transmissions, mechanical and hydraulic. The hydraulic design stress is slightly higher than mechanical because it is based on 1.25 times engine torque which we find produces normal life expectancy data for converter operation.

Using both design stress figures as a base, the effects of the load variation of each type of power train are illustrated by the load variation stress figures indicated by the chart. Also shown (Fig. 16) is the relationship of the observed mechanical shock loads when compared with the mechanical design stress, and also when compared with converter stress based on the stall torque of the converter. The converter stall stress is also shown compared with the converter design stress.

The observed mechanical shock-load stress is approximately the same as that shown for the hydraulic stall stress. However, the observed operating load for the hydraulic is lower, due to the smooth applied torque and continuous drive from zero at starting, to the average observed operating load.

According to our figures, the transmissions used in this investigation operated under the stress conditions indicated in Table 1.

Table 1 – Stress Conditions

	Hydraulic	Mechanical
Basic Average Stress, psi	31,200	25,000
Load Variations, psi	51,000	55,000
Shock Load Peaks, psi	—	108,000
Observed Peak Loads, psi	88,000	—

It is seen from these figures that the relationship for the various observed operating stresses of hydraulic and mechanical transmissions may be shown as follows:

Basic average stress is 25% greater for the hydraulic unit.

Load variation stress is 7.5% lower for the hydraulic unit.

Observed peak stress is 18.5% lower for the hydraulic unit.

It is commonly thought that transmissions used with torque converter drives must be designed for greater loads than those used with mechanical drives because of the engine torque multiplication through the converter. But considering all the factors in this particular investigation, the hydraulic unit has shown a lower overall operating stress when compared with the mechanical unit.

It is an outstanding fact that the service life of transmissions is, more or less, governed by stress variations in the power train, and that these variations affect and determine the actual life of transmission parts.

If the unit is operating at an average stress of 30,000 psi, but during the actual operation the stress varied from 20,000 to 40,000 psi and assuming 50% time of operation at both stress levels, the

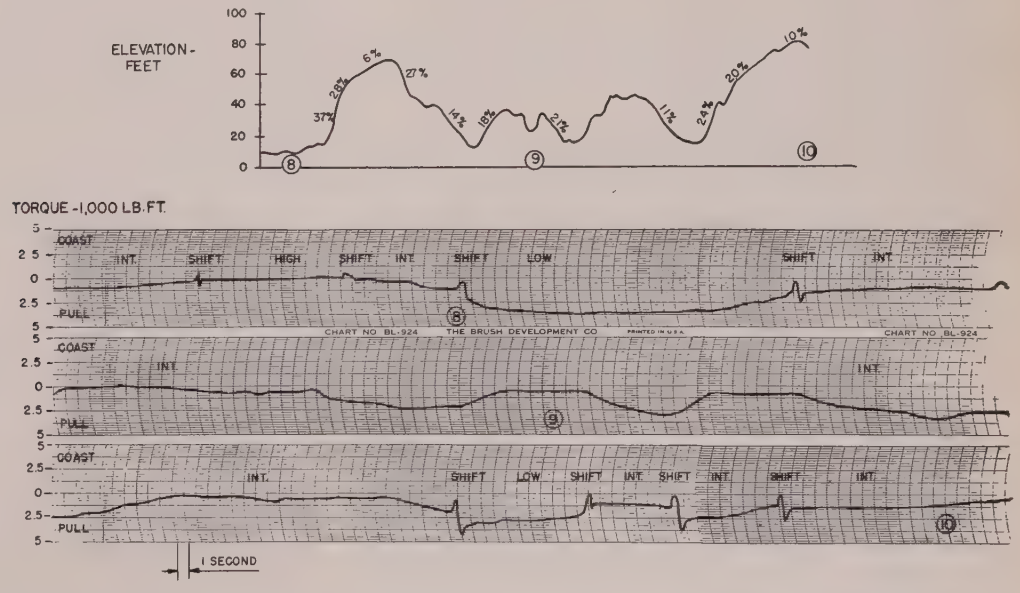


Fig. 14 – Torque distribution of converter-equipped truck operating over changing terrain (output shaft-transmission to final drive)

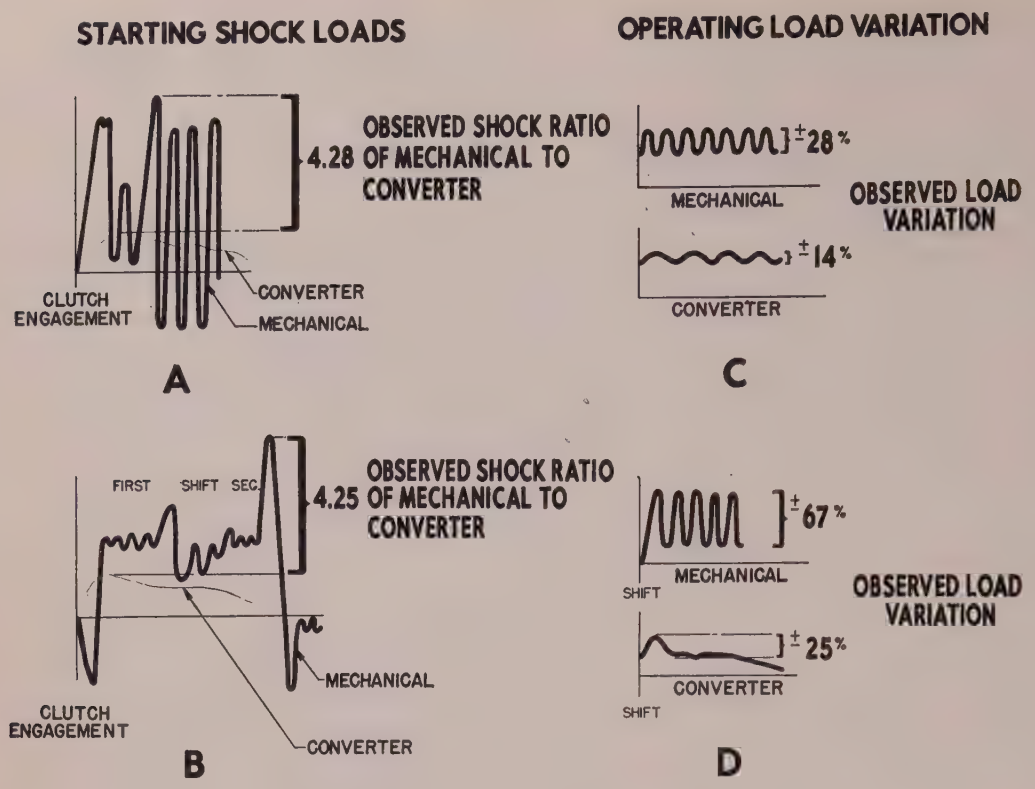


Fig. 15 – Results of field tests

life would be reduced approximately 56%, compared with the life expectancy of the theoretical average. This illustrates the fundamental factors in establishing the operating stresses on both mechanical and hydraulic transmissions as compared to the actual design stresses that may be used (Fig. 17).

As a matter of fact if the stress varied from 10,000 to 50,000 psi the life expectancy, compared with the average operating stress of 30,000 psi, would be reduced 86%.

Using the average operating stress results in Fig. 16 and the effects of stress variation as demonstrated in Fig. 17, it is possible to show how shock loads and load variation affect transmission life expectancy.

First taking the observed stress variation for the mechanical unit from Fig. 16, the total cycles are shown in Fig. 18 at A and B. The maximum stress is represented by A and minimum stress by B. This constitutes the load variations due to load distribution and the resulting life range is established. The total cycles for a constant mean or rated load is represented by C. By assuming that the operating stress cycles are distributed 50% at maximum and minimum stress during the working cycle, then the life expectancy would be represented by D, which is lower than for the mean applied or average load.

Next using the same line of thought for the hydraulic transmission, the life range is shown between A1 and B1, representing the narrower range of maximum and minimum stress for the observed load variation. The life for the mean or average load is indicated by C1, while the life D1 results from 50% operation at the maximum and minimum stress levels during the operating cycle.

Comparing the total cycles for each load variation D and D1, the result would be 60% more life expectancy for the hydraulic transmission. It is important at this point to keep in mind that the difference in the load variation in the two power trains is responsible for the increase in life in the cushioned hydraulic drive.

It also was shown in Fig. 16 that there were shock and observed operating stresses in addition to the normal stress variation encountered in the operating power train. If the two power trains were operated all the time at the extreme shock and observed operating stresses, the life expectancy would be indicated at A and A1 (Fig. 19). However, from the charts we see that both transmissions do not operate full time at these points, but only a small percentage of the total time of the haul cycle. Investigation showed that, for various haul cycles studied, the best average time to use was approximately 2% of the total time cycle. Taking these two

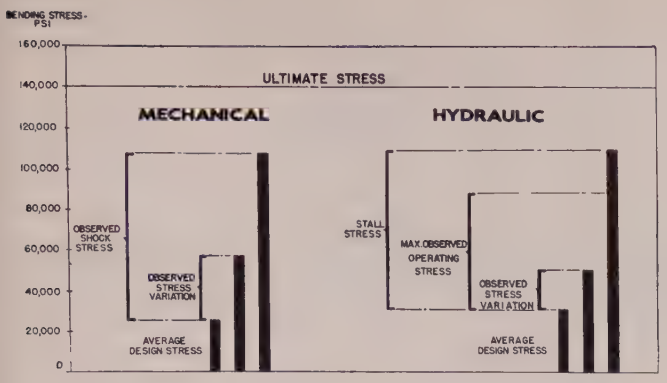


Fig. 16 – Relationship of transmission design and operating stresses

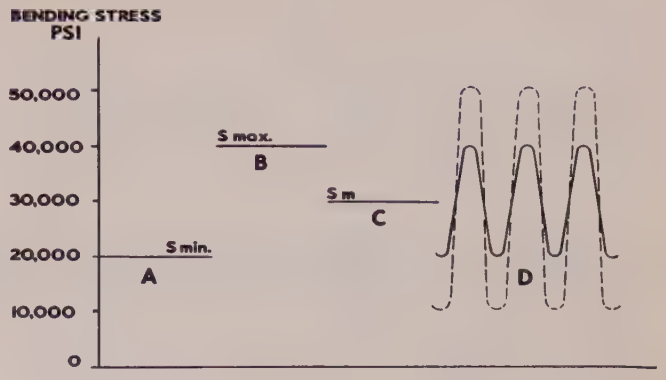


Fig. 17 – Stress variation

classes of loads and basing their operation on this time cycle, the life expectancy is shown at B and B1 (Fig. 19) and comparing them with each other results in approximately 125% more service life for the hydraulic transmission.

It is observed from these two classes of loads – load variation, and shock and observed maximum operating loads – that the hydraulic transmission life, when compared to the mechanical unit, is greatly influenced by the smooth load application and the lower load variation as shown by the tests. When only the load variation factors are considered the hydraulic transmission life was increased 60% over the mechanical unit, but when the shock loads and maximum observed operating loads are combined, the hydraulic transmission shows 125% more life over the mechanical.

Operators using both mechanical- and converter-drive trucks, not only cooperated with us in conducting the tests to obtain data for this paper, but also furnished maintenance records of engine, transmission, and differential overhaul life of these

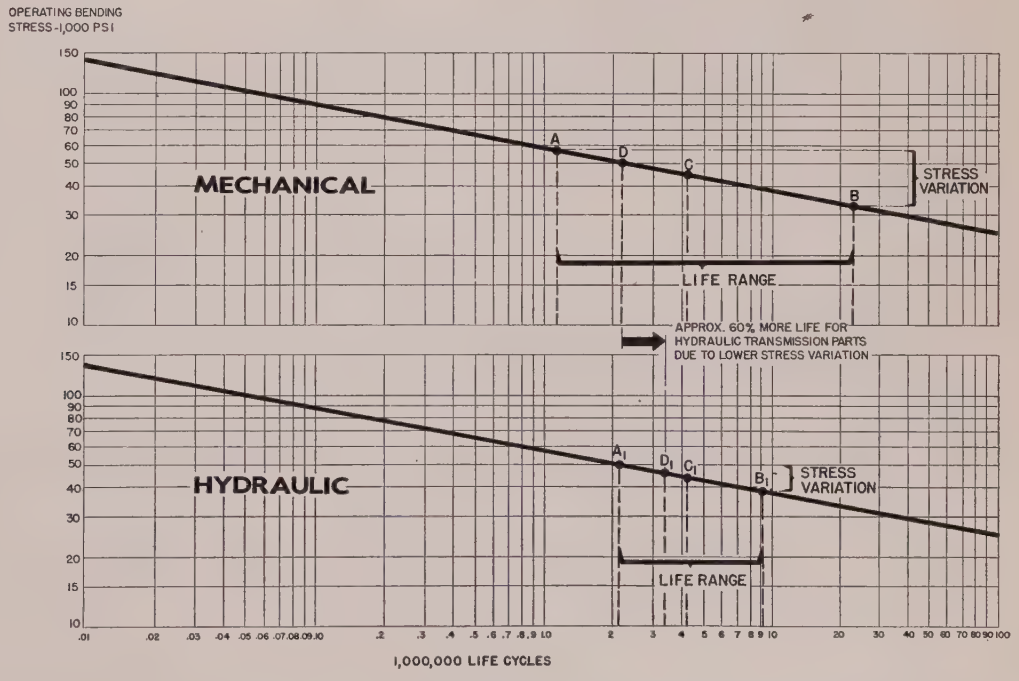


Fig. 18 – Life comparison of mechanical and hydraulic transmission parts (based on stress variation)

components, to demonstrate the influence, if any, of the converter on the life of its power-train components. The information gathered from several sources was not conclusive but did show a trend of better life for power-train components when using torque converters.

For both types of trucks, the frequency of overhaul of power-train components based on one engine overhaul is illustrated in Fig. 20. These data were taken from all records that were available and averaged to obtain the summation presented here. The mechanical power train showed 3.4 clutch, 3.18 transmission, and 0.76 differential overhauls based on one engine overhaul. These figures were based on summer operating conditions. For winter operation the clutch overhauls went up to 10.2 per engine overhaul. The converter records showed that the transmission and differential overhauls were 0.62 and 0.575 respectively. However, the converter-operated power train showed longer engine life than the mechanical unit, the ratio between the engines of the mechanical and converter units being 0.68. In other words, when the mechanical engine has reached overhaul time the engine of the converter units has gone only two-thirds of its overhaul period. Thus the transmission and differential overhaul ratios of the converter unit will change to 0.42 and 0.39 respectively.

The field information presented in Fig. 20 leads directly to a summation and comparison of the life of both mechanical and converter power-train components. Fig. 21 shows the life of the engine, transmission, and differential, for both units and comparing both types of power trains the results may be listed as follows:

The converter power train showed approximately:

47% more engine life.

400% more transmission life.

93% more differential life.

It has been pointed out before that these figures show a definite trend towards greater life for the power-train components when using torque-converter-equipped trucks.

Judging from the test data, there are three classes of loads imposed on the power-train components that affect their life:

1. Starting – First are the starting loads in both converter and mechanical units. The mechanical unit experiences shock loads when starting the vehicle, and their magnitude is greater than the maximum observed stall load of the converter unit.

2. Shifting – The second type of load is that encountered in shifting the mechanical unit with resulting resonance. This kind of load is not experienced in the converter power train to any degree.

3. Load Variation – The third class is the normal operating load and is represented by the torsionals in the drive. This kind of loading condition is indicated by the magnitude of the load variation and

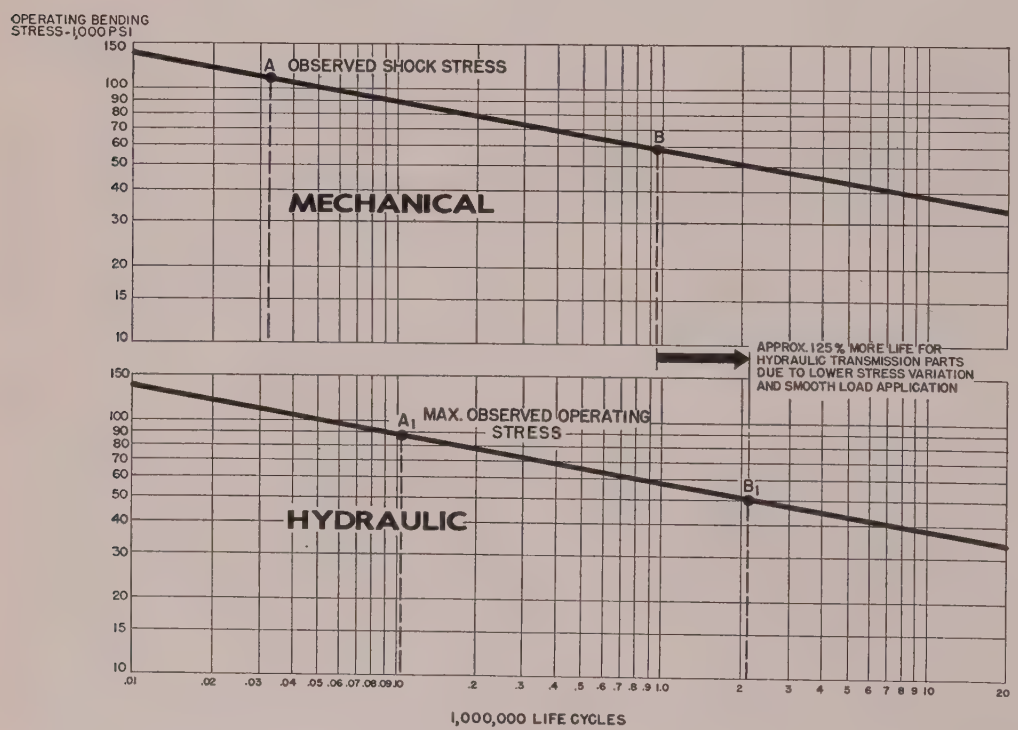


Fig. 19 – Life comparison of mechanical and hydraulic transmission parts (combining stress variation and shock and operating stress)

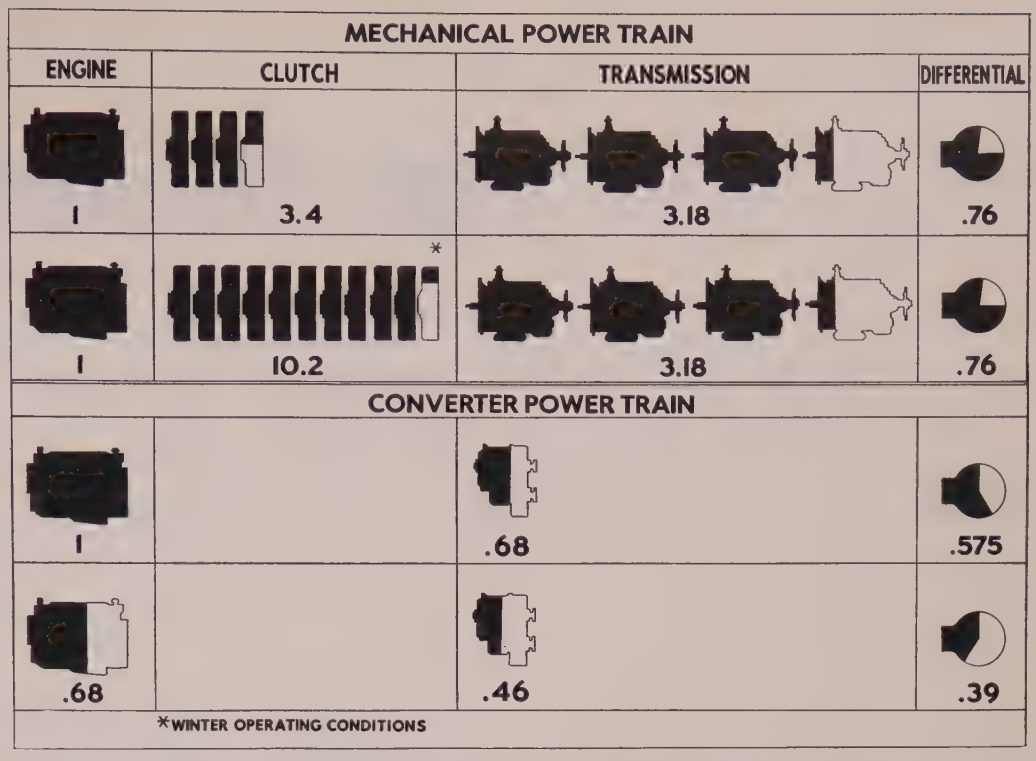


Fig. 20 – Summation of power-train components

it frequency during the truck-operating cycle. Due to the absorbing ability of the hydraulic drive, smaller load variation and practically no cyclic frequency exist in the converter power train.

In conclusion, since the torque converter eliminates shock loads when starting and stress variation caused by torsionals and resonance, we find the following things to be true in hydraulic transmission parts:

Higher design stresses may be used.

For equal life, it is possible to use higher loading.

For equal loading considerable more life will be obtained.

Operating data previously collected in the field, and presented at the SAE Earthmoving Industry Conference, April 1950,[1] indicated that with a converter the equipment can be operated at a higher load factor in order to get more work done. (See Fig. 22.) From the facts presented above, it is indicated that with a converter the equipment also has a greater life factor. Since the equipment can be operated at a higher load factor with longer life factor, the converter drive will result in reduced operating costs to the owner.

[1] "Going to Work with Hydraulic Transmissions," by R. M. Schaefer. Presented at SAE Central Illinois Section Meeting, Earthmoving Industry Conference, Peoria, Ill., April 11, 1950.

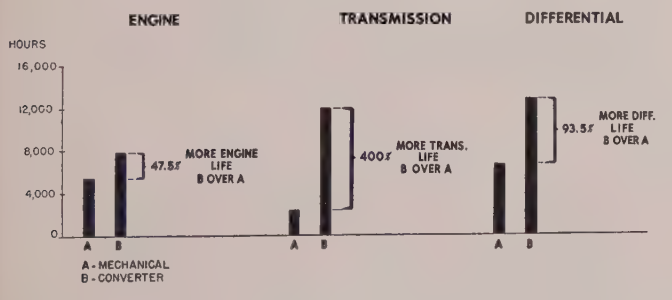


Fig. 21 – Average life of power-train components

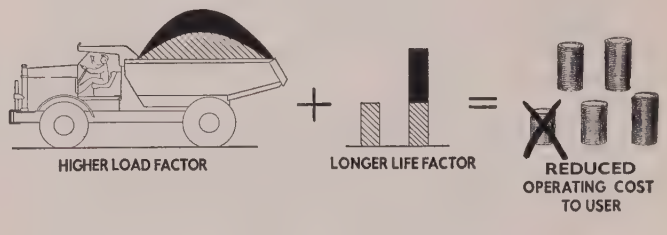


Fig. 22 – Higher load factor plus longer life factor equals reduced operating cost to user

Can All Engine Wear

TODAY'S progress in the development of the automotive engine is marked by frequent announcements of new engine designs with lower frictional losses, higher compression ratios and horsepower ratings, offering greater economy and thrilling new performance in driving; but we cannot afford to forget that the consumer will ultimately measure our progress in terms of engine life. Much of our effort has been directed to the study of engine life by investigating wear in typical service, and trying to learn the most effective ways of controlling it.

In general, engine life is recognized as the number of miles of operation the engine affords before, (1) wear in the cylinder bores and of piston rings increases clearances to the point where oil consumption becomes excessive, and (2) bearing shell and journal wear enlarge passages for increased oil flow with a resulting loss of oil pressure. When the lubricated surfaces of engines wear out in this manner, solid particles of contaminant in the film of lubricant have augmented the loss of metal by increasing friction and abrasion between the surfaces. These contaminants have several sources, among which are (1) dirt, metal chips, and core sand "built into" the engine, (2) dust particles entering through the induction and ventilation systems, (3) metal worn off the piston rings and cylinder walls and carried throughout the system by the lubricating oil, (4) carbon particles and varnish caused by oxidation of the oil in elevated tempera-

THE study of engine life, carried out by investigating engine wear in typical service, and by then striving to find the most effective ways of controlling it, forms the basis of this paper on contaminants in lubricants. The investigation involved a study of engine wear in 20,000 miles of operation typical of the average driver.

The average driver was selected by using test cars from an employee transportation car pool. At the conclusion of the tests it was found that the use of the full-flow oil filter proved to be the best method for restricting engine wear caused by contaminants that get inside the engine.

It was also shown that after successfully eliminating large, solid particles, further restriction of engine wear would depend upon the ability of the oil to lubricate, and upon the engine design to provide the oil supply in a manner suitable for lubrication of each part of the engine.

The Author

R. J. POCOCK is supervisor of the Lubrication and Filtration Laboratory, Engineering Research Department, Ford Motor Co. He organized this unit in 1945. Mr. Pocock joined Ford in 1939 after studying mechanical engineering at the University of Detroit. Since that time he has worked at Ford in various engineering capacities.

Be Trapped in a Can?

R. J. Pocock, Ford Motor Co.

This paper was presented at SAE National West Coast Meeting, San Francisco, Aug. 11, 1952.

ture zones in the engine, and (5) lead salts from octane improvers in the fuel.

This investigation involved a study of engine wear in 20,000 miles of operation relative to the effects of engine oil conditioning by periodic oil changes and oil filtration. Separate studies were made to determine to what extent engine wear is affected by operation with fuels containing octane improvers, inhibited oils containing appreciable quantities of detergent additives, and air cleaners of different efficiencies.

Scope of Investigation

Primarily, the data in this investigation was to be representative of the magnitude of wear occurring in operation typical of the average driver. Who do we feel is an average driver? He may be the person who peers back at some of us from a mirror; or our neighbor; or someone we associate with in our work day: a fellow we all know but may fail to recognize. To make sure that we would not neglect to include him in our test program, the tests were conducted with the use of vehicles from the Ford transportation car pool operating in the Detroit area. These cars are driven daily by company personnel, in connection with the dispatch of employee and courtesy transportation between plant operations throughout greater Detroit, and to vendor's plants or other places of business in this locality. In this way we avoided the personal driving habits of one operator and obtained a thorough cross-section of drivers.

Although the actual operation of these engines was not directly controlled, the many factors that influenced the test results were controlled by the thorough follow-up procedure of the engineering research field test group. Included among these controls were all mechanical servicing of the engines and test components, the dispensing and tabulation of the amount and proper type of fuel and oil, and the removal of periodic samples of fuel and lubricating oil for analysis in the laboratory.

In this study of engine wear results, the basic Ford V-8, 239 cu in. engine was used. However, since the program to date has been carried on over a span of five years, there were three models of this engine used: the 59A, 0BA, and 1BA. The latter two engines are basically similar, but one distinct difference exists in the type of connecting-rod bearing used in the 59A engine. The 59A engine used a single floating bearing for each pair of rods, whereas the 0BA and 1BA engines used the individual locked-in bearing.

The engines are normally equipped with the bypass type of oil filter. However, some of the engines were modified to use the full-flow oil filter system for testing purposes. In the presentation of this data on engine wear, a comparison of wear criteria between the different models of engines has been avoided, except where our studies conclusively showed that the changes in design did not prohibit comparison.

The engines were operated in Ford sedans of model years 1947, 1948, 1951, and 1952. And, depending on the duration of the test, the engines were sometimes transferred from cars of one model year to another. The 59A engines were operated in cars equipped with standard transmissions. The 0BA and 1BA engines were coupled to either a standard transmission, overdrive, automatic transmission, or, as in several cases, two types of transmissions were used with the same engine. A study of the data has not indicated that the type of transmission affects the wear. A tabulation of the

Table 1 – Tabulation of Operating Data for Type 0BA and 1BA Engines

Group No.	Engine Summary: Engine No.	Total Engine Miles	Average Miles per Month	Type Car: Fordor	Type Car: Tudor	Transmission Summary: No. of Transmissions Used: Standard	Fordomatic Drive	Overdrive	Miles Operated: Standard	Fordomatic Drive	Overdrive
I	R-0BA-298	20,013	1650	4	—	3	1	—	12,272	7,741	—
	337	20,002	1560	4	1	—	5	—	—	20,002	—
II	R-0BA-300	20,291	1524	3	—	1	2	—	4,267	16,024	—
	338	20,079	1512	3	—	—	3	—	—	20,079	—
III	R-0BA-287	20,004	1631	1	2	2	1	—	13,848	6,156	—
	288	20,030	1665	2	1	2	—	1	10,975	—	9055
IV	R-0BA-290	20,019	1765	—	3	2	—	1	11,330	—	8689
	291	20,004	1753	—	3	1	2	—	9,720	10,284	—
V	R-0BA-297	20,067	1632	3	—	1	2	—	6,895	13,172	—
	299	20,065	1545	6	—	3	3	—	10,526	9,539	—
VI	R-0BA-282	20,048	1568	4	1	3	2	—	9,964	10,084	—
	283	20,099	1793	3	—	2	1	—	13,250	6,849	—
VII	R-0BA-284	20,018	1598	2	1	3	—	—	20,018	—	—
	285	20,047	1837	3	—	1	2	—	9,231	10,816	—
VIII	R-1BA-384	20,019	1971	2	—	2	—	—	20,019	—	—
	385	20,005	1580	3	—	3	—	—	20,005	—	—
IX	R-1BA-388	20,011	1654	3	—	3	—	—	20,011	—	—
	389	20,008	1754	1	1	2	—	—	20,008	—	—

conditions of operation of the 0BA and 1BA engines appears in Table 1.

The bypass oil filter elements were of the replaceable type, operating in standard Ford oil filter bodies. The filter elements were of the extended area, surface-type, paper filter construction, with a diameter of 4.0 in. and an overall length of 4.875 in. with a 9/16-in. center opening in each end cap to which was added a rubber sealing washer, when installed in the filter body. In operation, the oil flow was from outside to inside through the resin-impregnated paper which acted as the filter medium.

The full-flow filter elements were of the replaceable element type in bracket-mounted filter bodies, with $\frac{3}{8}$-in. pipe inlet and outlet connections and a built-in pressure relief valve, set to open when the differential pressure exceeded 8 psi. These special filter bodies were made by modifying the standard Ford bypass filter body to incorporate the larger passage and the built-in relief valve. The inlet and outlet oil line connections to the engine were through $\frac{3}{8}$-in. flexible hose lines. The full-flow filter elements were of the extended area, surface-type, paper construction, with an outside diameter of 4.0 in. and an overall length of 4.75 in. with a center opening in each end of $\frac{3}{4}$-in. diameter. The elements were similar in construction to the bypass type. The main difference between the bypass and full-flow filter element types was in the density of the filter paper used, which in the full-flow element was less restrictive to the passage of oil in order to take care of the additional oil flow requirements.

The carburetor air cleaners were classified in the oil-wetted and oil bath groups. Both cleaners were of the hat-type construction, designed to conserve space.

The oil-wetted cleaner had an element of copper wire mesh with an operating efficiency of from 40 to 70%.

The oil bath cleaner utilized the washing action of a 1-pt bath of oil to make the cattle tail hair-constructed element self-cleaning to a degree. This type of cleaner had an efficiency of 95 to 99%.

The base oil was solvent-refined from a Mid-continent base stock and had a viscosity index of 100 minimum. The oil was prepared in a SAE 10W grade and was furnished in sealed quart cans for test purposes.

The additive oil was especially prepared for us in the 10W grade by adding to the base oil the same measure of inhibitors and detergent additives used in the SAE 30 grade oil which had qualified under U. S. Army Ordnance Specification 2104-B, Supplement No. 1. This oil was also furnished in sealed quart cans for test purposes.

The test fuels used were catalytically cracked and were basically the same with respect to volatility, vapor pressure, and gum content; but differed in the use of octane improvers and octane rating number.

The base fuel was a nonleaded fuel with an octane rating of 83.0 by the Research method.

The leaded fuel was made from the same base stock as the base fuel, with the addition of approximately 2.44 cc of tel per gal and had an octane rating of 91.3 by the Research method.

The engine measurements were made on the new engines before test and after the completion of 20,000 miles of operation. The wear measurements included the break-in wear. It was found that the following were the most reliable criteria for determining the wear of engines in these tests:

1. Increase in cylinder-bore diameters.
2. Increase in piston-ring gap.
3. Increase in side clearance of top piston ring.
4. Main and connecting-rod bearing weight loss.

In all, a total of fifteen different measurements were made on each engine. Other measurements were found to be in general agreement with the above, but since the magnitude of wear and rela-

tive significance were less, they have been omitted in order to simplify the presentation of results.

The samples of oil removed periodically from the engines were checked for contamination, fuel, and water dilution, viscosity change, and neutralization number. The samples of fuel removed from some of the cars were checked for tel content as a control measure for those cars operating on leaded and nonleaded fuel.

Throughout the test program, a pair of engines was operated for each condition under investigation. Because of the many different types of bypass oil filters available, we found it necessary to allot a greater number of engines to this category. For each of the different bypass-type filters tested, a pair of engines was assigned. Consistent results could not be expected, since the efficiencies of the different filters vary considerably. However, in all instances, engines equipped with bypass-type filters showed a marked reduction in wear. For our presentation of data on bypass filters, we have selected the data from three engines which were equipped with extended area, surface-type filter elements, similar in construction to the full-flow filter elements used in our study of that system.

In servicing the engines, Ford service recommendations were followed, except where test conditions prohibited. Service of the air cleaner, breather cap filter, oil filter, oil changes, and engine tuneup were included among these operations.

Test Results

As mentioned previously, engine wear occurring from operation typical of the average driver was sought. A survey was conducted to classify the different types of operation available among vehicles in the company fleet, using the passenger cars and station wagons assigned to different types of service. The survey established that the cars used for general dispatch service represent a happy medium of the short trip and long trip drivers, as shown in Table 2. It is interesting to note in the tabulation that a correlation exists between the frequency of engine operation and the relative percentage of wear in the upper cylinder bore at the top ring travel limit.

Table 2 – Comparison of Top Cylinder-Bore Wear from Survey of Ford Fleet

Group	At 6000 Miles			At 10,000 Miles		Average Mileage per Month
	Wear, in.	Relative Per Cent	M[a]	Wear, in.	Relative Per Cent	
Executive Assigned	0.00261	100	100	0.00372	100	750
General Dispatched	0.00163	62.8	62.2	0.00244	65.6	2115
Scheduled Route	0.00137	52.5	50.8	0.00181	48.6	2395

[a]M is the index of frequency of engine operation. The higher the index value the greater will be the number of starts for the same mileage.

The question arises, "What other factors affect engine wear?" In operation, an engine breathes air and burns fuel, and the various rubbing surfaces are lubricated with oil to reduce friction and prevent sticking. Consequently, the determination of variations in wear that would be introduced by these factors was considered.

Effect of Tel in Fuel on Engine Wear – First, let us see what variables are introduced by the fuel used in the engine. Each condition of operation represents the average of measurements from a pair of engines. The engines were equipped with oil bath air cleaners and full-flow oil filters, and were lubricated with the base oil.

Fig. 1 is a bar graph showing the percentage of increase in wear that was observed in the en-

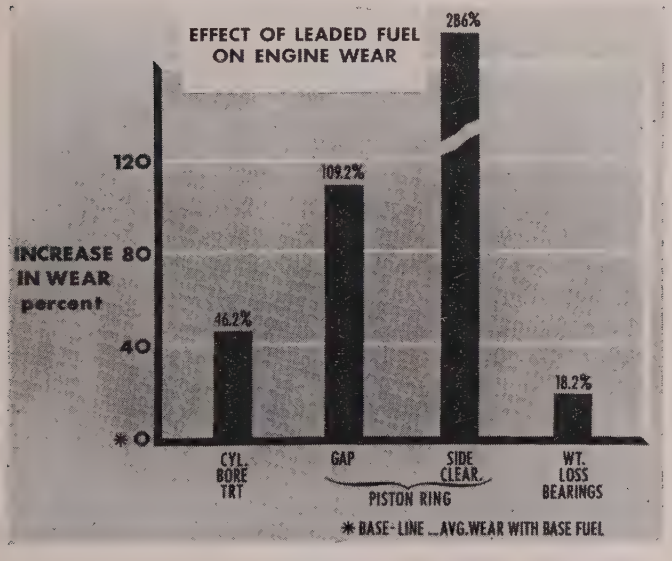


Fig. 1

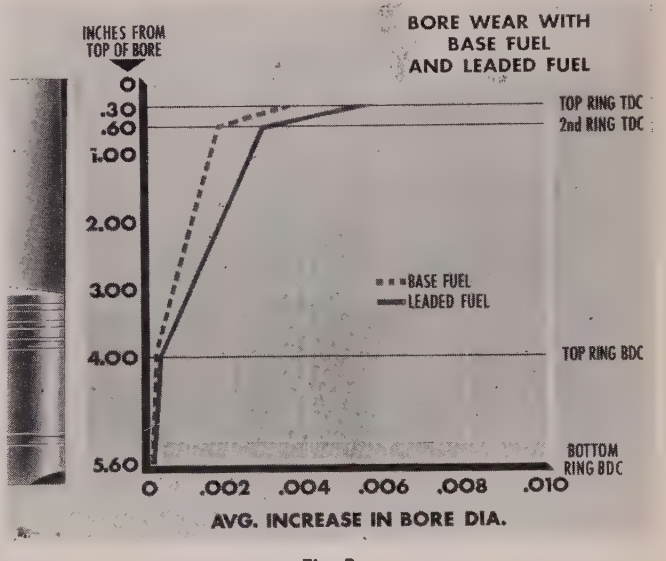


Fig. 2

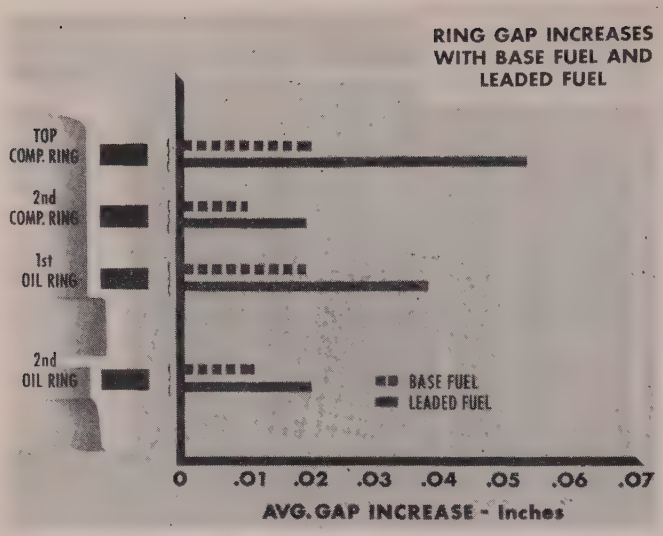


Fig. 3

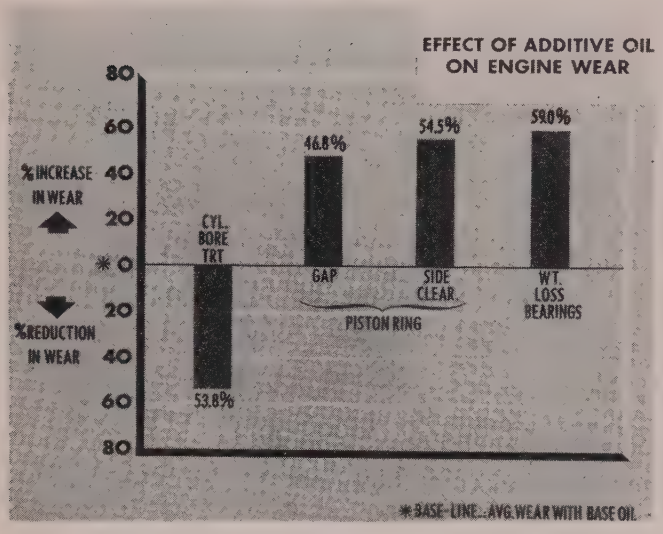


Fig. 4

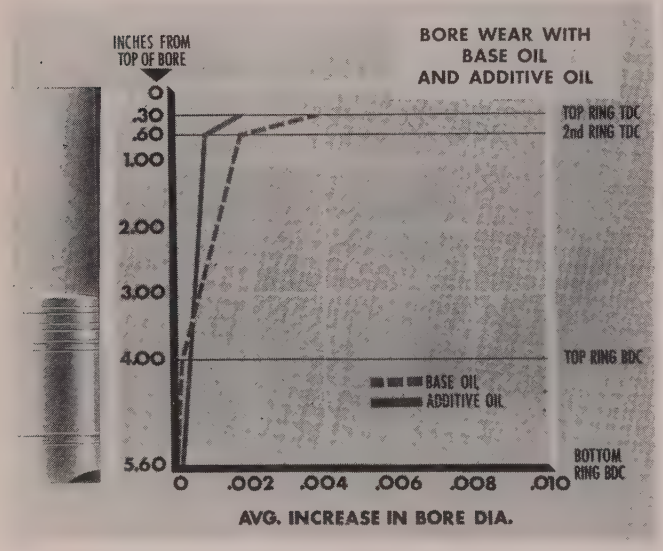


Fig. 5

gines operated with the leaded fuel. The distribution of wear of the cylinder bores is shown in Fig. 2. The total change in bore diameter is plotted on the horizontal axis, increasing in diameter from left to right. The depth of the bore is plotted on the vertical axis. Wear measurements were taken at the upper limits of top ring travel and second ring travel; and at the lower limits of top ring travel, and bottom ring travel. Measurements were made of four diameters, in planes parallel and perpendicular to the crankshaft and bisecting planes. The data represents the average wear of the four diameters for each position in the bore. The curves may exaggerate the wear differential near the center of travel of the piston. The locations of measurements taken in the bore leave a relatively large section without measurement, and it is possible that the wear in this area would be less than that shown. The wear in the cylinder bore shows the effect of the lead compounds carried by the rings throughout the ring travel.

In Fig. 3 is shown the increase in piston ring gaps. The average gap increase of the top compression ring appears at the extreme top of the bar graph, the second compression ring below it, and the last two positions show the first and second oil rings. The scraping action of the top ring as the piston travels back up the bore is a normal function. When the leaded fuel was used in the engines, the additional work of removing the unburned lead deposits from the cylinder walls caused a sharp rise in the wear of the top compression ring. The top oil ring shows a similar effect. With the exception of the bearing wear, the other measurements did not indicate that the wear patterns were affected by the presence of tel in the fuel.

The quantities of lead compounds that passed into the oil had a pronounced effect on the life of the oil filter elements. A total of 19 full-flow filter elements were used for the pair of engines operated on the leaded fuel, whereas only 11 elements were required for the pair of engines operated on the base fuel. This, in itself, is a substantial indication of the amounts of lead compounds that are not carried out through the exhaust system.

Effect of Oil Additives on Engine Wear – Next, we shall see what effect the addition of inhibitors and detergent additives have on engine wear. In Fig. 4 is shown a bar graph with comparisons of the average wear in the engines operated with the base oil, and those operated with the additive oil. The wear of the engines operated with the base oil is used as the base line, and the increase or decrease in wear effected by the use of the additive oil is shown as a percentage of the base line wear. These engines were equipped with full-flow oil filters and oil bath air cleaners, and were operated on the base fuel. As a rule, the cylinder bores and piston rings wear similarly in an engine, but, in this example, the wear at the top of the cylinder bore was considerably less with the additive oil,

even though the ring wear was greater for the same engines. The next two charts will throw more light on what happened.

The wear of the cylinder bores is shown in Fig. 5, in which it should be noted that the wear in the upper part of the cylinder bore is considerably less with the high-additive oil, while the lower portion of the bore shows a reversal of these conditions with a somewhat higher bore wear. One explanation for this wear pattern would be that, in the upper portion of the cylinder, the action of the detergent in the oil reduces the amount of varnish and carbon deposits on the side of the piston above the ring and carries a good part of them away in the oil. This action would reduce the abrasive effect of these formations on the cylinder walls. In the lower portion of the bore, the influence of contaminants carried in suspension in the oil by the detergent resulted in increased bore wear.

Fig. 6 shows the average gap increase of the piston rings in both pairs of engines. Notice that there is a greater differential in the gap increase of the bottom rings, decreasing with the higher ring positions. The top oil ring has the greatest total gap increase, probably because the greatest volume of oil flows through this ring.

Fig. 7 shows the weight loss of the main and connecting-rod bearings. The average weight loss in milligrams of the main bearings appears at the top of the chart, and the average weight loss of the connecting-rod bearings is shown at the bottom. The increased wear occurring in the engines operated with additive oil is a reflection on the ability of an oil filter to remove all harmful suspended particles of contaminant from detergent oil. Later in our presentation, we will deal with the contamination level in the oil and its relation to engine wear.

Effect of Oil Additives on Engine Wear With Tel in Fuel – The effect of additives in the base oil on the wear level in the engines when lead compounds from the fuel were present is shown in Fig. 8. Here, we see again the reduction in wear at the top of the cylinder bore; and, in addition, we find a reduction in the wear of the piston rings, as indicated by the gap and side clearance of the rings. However, increased bearing wear is again evident in the 85.6% increase in weight loss of the bearings. This represents an increase of 26.6% over the increase in wear previously shown for the additive oil, and is believed to be the effect of the presence of lead compounds in the oil. This differential in wear compares with an increase of 18.2% that was shown for the leaded fuel when the engines were lubricated with the base oil.

The wear pattern in the cylinder bores for this condition is shown graphically in Fig. 9. Again, the reduction in upper cylinder-bore wear and the increased wear of the lower cylinder bore are typical of the additive oil. The data previously shown in Fig. 5 for engines operated on the base

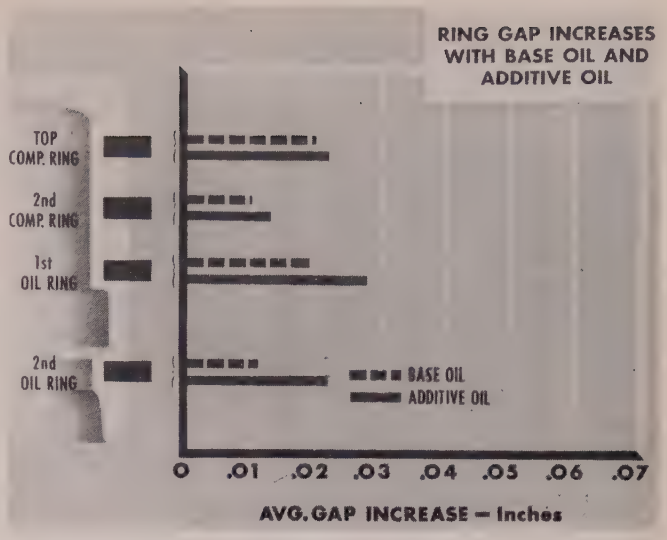


Fig. 6

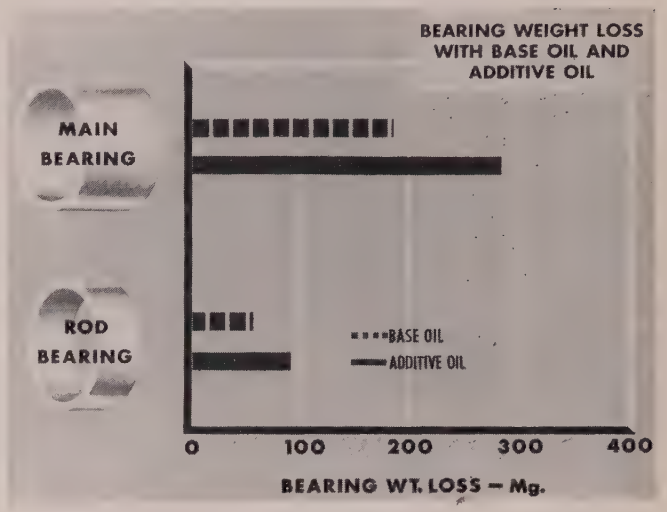


Fig. 7

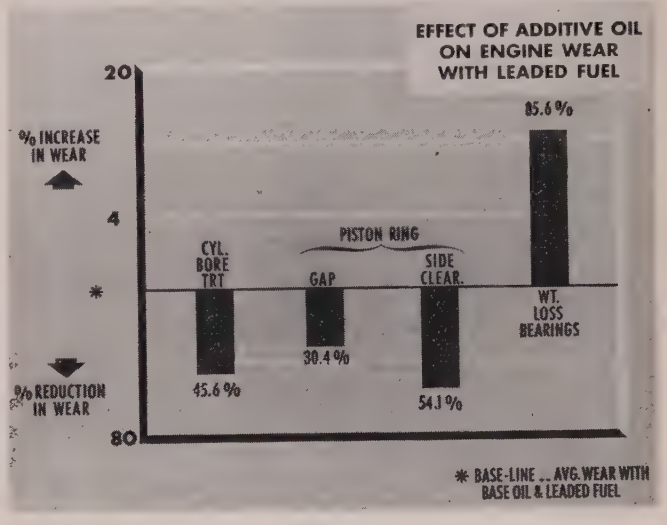


Fig. 8

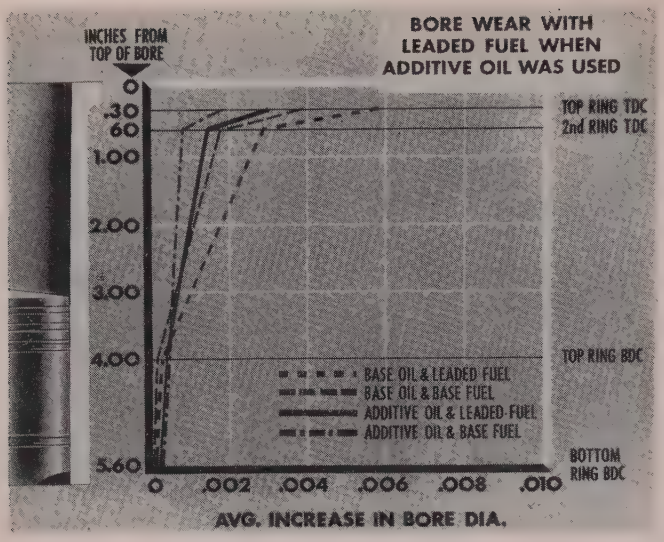


Fig. 9

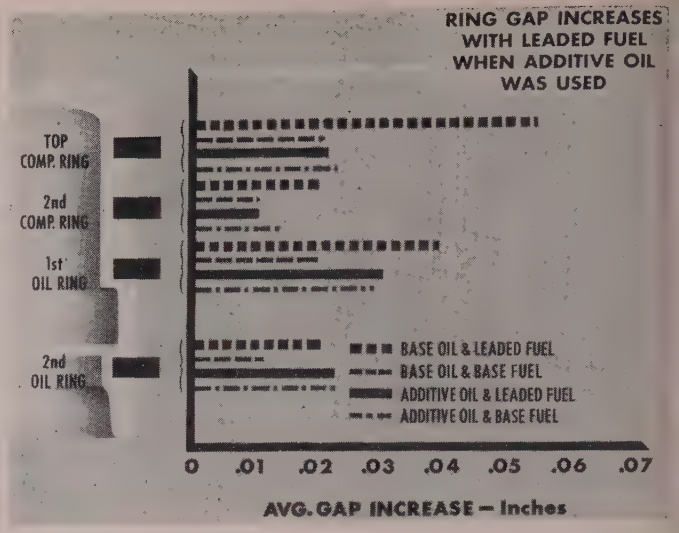


Fig. 10

fuel is also included to point up the increase in the wear level when the leaded fuel is used.

In Fig. 10 is shown the increase in piston ring gap for the two pairs of engines of this test condition as compared with the corresponding data using the base fuel. Notice that the bars showing the wear when the additive oil was used are practically the same, regardless of which fuel was used. This indicates that the lead compounds are held in suspension by the detergent and carried away in the oil, permitting the rings to move freely.

Fig. 11 shows the weight loss of the bearings to be characteristic of the increase in wear when the high-additive oil is used. There appears to be a slight increase in the weight loss of the main bearings and a reduction in weight loss of the connecting-rod bearings when the leaded fuel was used.

Effect of Different Air Cleaner Efficiencies on Engine Wear – The third major factor that influences the level of wear in an engine is the amount of abrasive contaminant particles that are passed by the carburetor air cleaner, and permitted to enter the engine through the induction system.

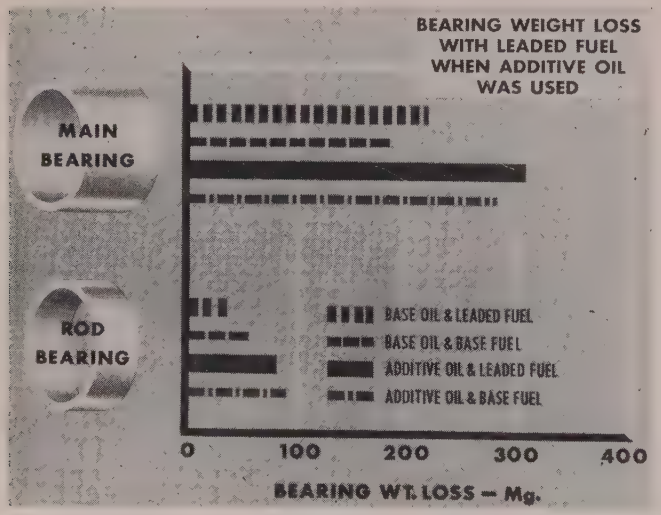


Fig. 11

In Fig. 12 is shown the increase in engine wear brought about by the amount of additional contaminant entering through the air cleaner when an oil-wetted cleaner of 40 to 70% efficiency is substituted for an oil bath cleaner of 95 to 99% efficiency. No evidence of the effect on bearing wear was noted for this test condition. These engines were equipped with bypass oil filters.

In Fig. 13, the relative magnitude of cylinder-bore wear at different levels is shown. Notice that the effects of these particles are carried down the entire length of piston-ring travel.

Fig. 14 shows the piston-ring wear, in which the familiar pattern is evident, but the magnitude of gap increase resulting when a greater number of dirt particles enter the induction system is very pronounced. The large increase in the gap of the top compression ring reflects the higher gas-loading and greater rubbing intensity of this ring. The gap increase of the first oil rings, as mentioned before, is probably a result of the greater flow of oil through this ring. Not all of the increase in wear is a result of the first passage of dirt between the piston rings and cylinder bores. A discussion of how a portion of this wear can be controlled will be presented later.

To summarize briefly what we have covered so far, we see that the amount and pattern of wear in an engine is affected by (1) tel added to the base fuel, (2) the additives in the base oil, and (3) the introduction of larger quantities of dust particles with the air entering the induction system. We point out that one element in the operation of these engines has been active in assisting each of these variables to penetrate throughout the engine, and that has been the lubricating oil which

acts as the carrier medium. In each case, the element that promoted wear was some form of contaminant particle which reduced the efficiency of the lubrication film, or retarded the normal movement of the part, as in the case of the piston rings. Therefore, the next logical step is to find the most effective means to rid the oil of these particles which cause wear in an engine.

Effect of Different Methods of Oil Conditioning on Engine Wear – To evaluate the effects of different methods of oil conditioning, that is, (1) by replenishment of consumed oil; (2) by periodically changing the oil; or, (3) by oil filtration; a separate program, entailing the use of fifteen cars, was set up, and operated in the following manner:

Case I: Not equipped with oil filter – no oil change.

Case II: Not equipped with oil filter – periodic oil change (approximately 1000 miles).

Case III: Equipped with partial-flow oil filter – no oil change.

Case IV: Equipped with full-flow oil filter – no oil change.

From the test results, the base-line wear of the engines not equipped with oil filters and without oil changes (Case I) for the 20,000 mile test was as follows:

Cylinder-bore wear (top of piston ring travel): 0.0060 in.

Piston-ring gap increase (average of all rings): 0.031 in.

Piston-ring side clearance (top ring only): 0.0052 in.

Main bearing weight loss: 473 mg

The effectiveness of the three methods of oil conditioning as shown in Fig. 15, establishing the wear of the engines in Case I above as 100%, and

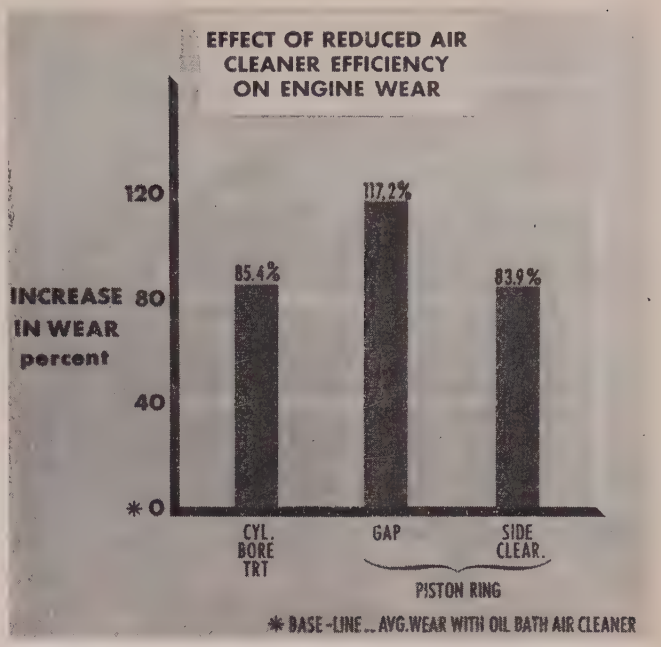


Fig. 12

showing the relative wear of Case II, Case III, and Case IV. The real effectiveness of oil filtration is clearly evident in these observed results. The full-flow type of oil filter, through which all of the oil must pass before being delivered to the lubricated points of the engine, offers the greatest protection.

Effect of Oil Filtration on Engine Wear Resulting from Contaminant Passed by Air Cleaner – We have shown previously the effect of air cleaner efficiency on the level of wear in an engine using a bypass oil filter. Now, let us take a look at what happens to the engine wear for the two types of air cleaners when a different kind of oil filter is

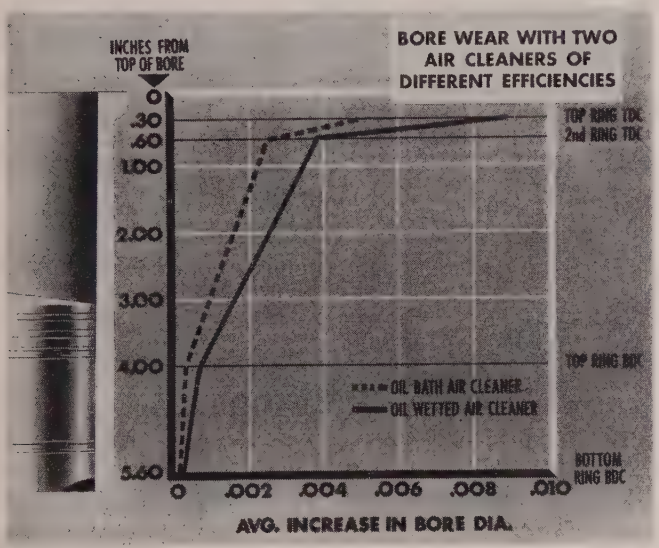


Fig. 13

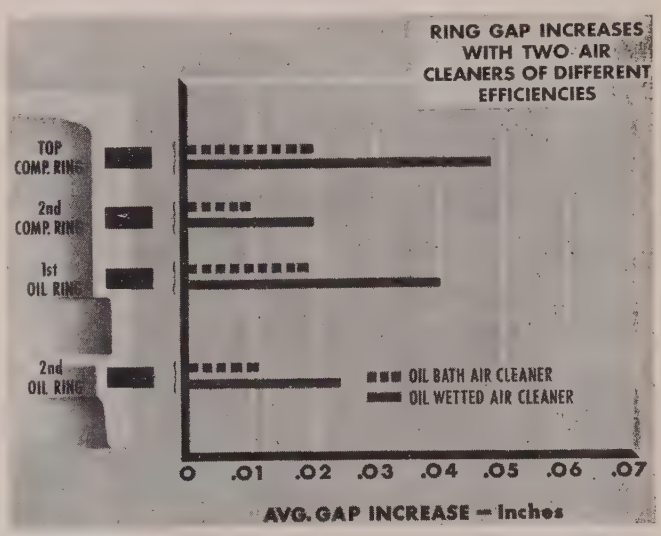


Fig. 14

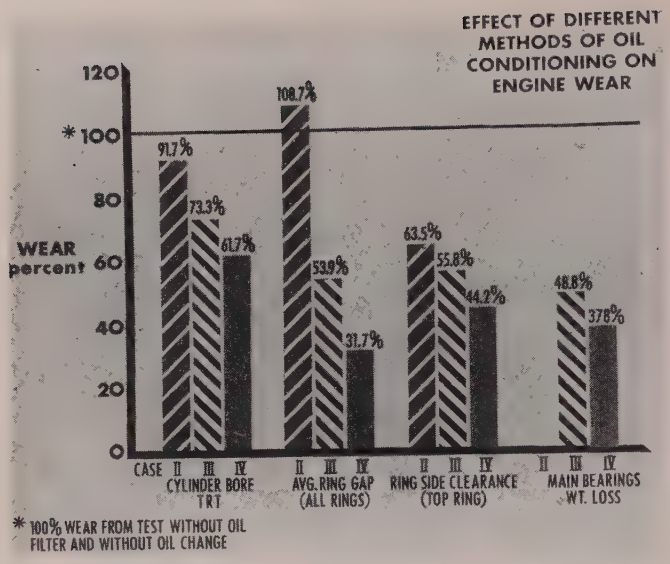


Fig. 15

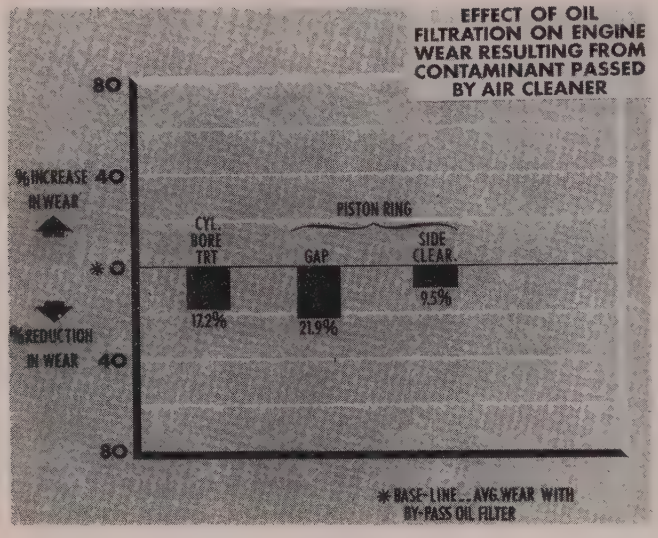


Fig. 16

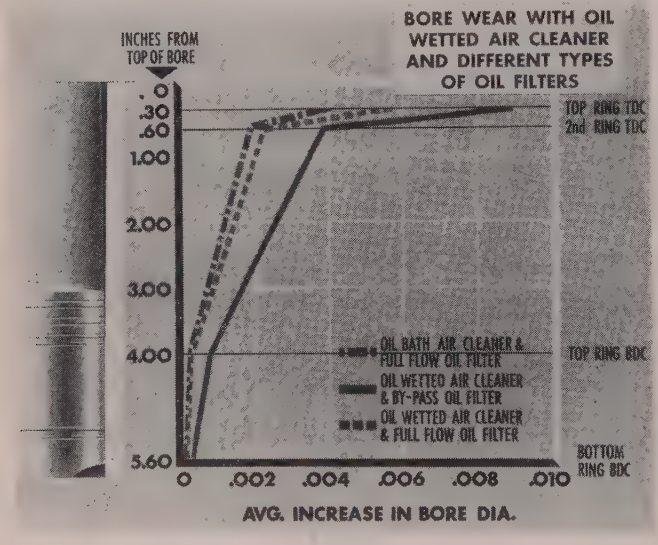


Fig. 17

used. The base fuel and base oil were used in each pair of engines.

In Fig. 16 is shown the net reduction in wear effected by the full-flow filter over the level of wear observed with the bypass oil filter, when both were operated with an oil-wetted air cleaner. These figures have been adjusted to remove the improvement that has previously been shown to exist in favor of a full-flow oil filter system, so that they show that some of the wearing action of these abrasive contaminants that enter through the induction system can be controlled by an oil filter. We believe that the bypass filter was effective in reducing wear that would result from this source of contaminant, but for maximum effectiveness, the full-flow oil filter is favored.

The next chart, Fig. 17, shows the cylinder-bore wear patterns of the engines equipped with oil-wetted air cleaners, and shows the reduction afforded by the full-flow oil filter over that of the bypass filter. Also, an additional curve illustrating the bore wear observed for the full-flow filter with the more efficient oil bath air cleaner is included, showing that there is a margin of wear remaining that the full-flow filter could not control. This margin may represent the wear of the initial entry of the contaminant before it gets mixed into the oil stream and is carried to the oil filter. Fig. 18 shows the ring wear that occurred under the above conditions, and again, the same advantages are evident.

Effect of Different Bypass Oil Filters on Engine Wear – There are many kinds of oil filters available for application on an engine of both the full-flow and bypass types. We have not tested all of them, but our test program has included many of the most widely known and commonly used brands. Laboratory tests have shown that the efficiencies of different oil filters of both types vary considerably. It is difficult to correlate laboratory tests of filter performance, with road tests meant to determine the ability of a filter to limit engine wear. It is apparent that a variable exists which prevents correlation in some cases, and we think that our next charts will throw some light on it.

Fig. 19 shows percentage of wear increase observed in two pairs of engines equipped with different types of bypass filters made by the same manufacturer. Tests in the laboratory have shown that one of these filters has a low effluent flow of oil through the filter medium as compared to a relatively higher flow through the other type. The low-flow filter consistently showed an increase of 15% in contaminant removal efficiency over the high-flow filter. The contradictory results shown here (in terms of wear) are evidence that the degree of cleanliness a filter can maintain in the engine oil is not all important, and that a greater amount of oil filtered to a somewhat lesser degree is more effective in protection against wear. The

elimination of this variable is undoubtedly responsible for the rapid trend toward full-flow filters.

Effect of Full-Flow Oil Filter Efficiency on Engine Wear – When an engine is designed to provide for all of the oil supplied to lubricate the engine to be passed through a filter medium, then the efficiency of the medium to remove harmful contaminants from the oil has a direct bearing on the effectiveness of the filter. The wear observed in a pair of engines operated with full-flow filters that were found in the laboratory to differ in contaminant removal efficiency by approximately 22% is shown in Fig. 20. By some coincidence, the numerical difference in efficiencies of the filters is almost the same as the per cent reduction in wear shown in this chart for the cylinder-bore wear and piston-ring gaps. Whether or not this is also significant is not known.

Oil Analysis Results Compared to Piston-Ring Gap Increase on Several Engines – We have dealt to some extent with data relating to laboratory findings of the contaminant removal efficiency of various filters. Before leaving this subject, let us examine briefly the data we obtained from analysis of oil samples taken from the engines during our test programs to see if a correlation with wear in the engines exists.

In Fig. 21 is shown the average increase in the gap of the piston rings for seven pairs of engines, and the corresponding average petroleum ether insolubles and chloroform insolubles by weight found in the oil for each pair of engines. The groups of engines were arranged so that the respective average gap increase of the piston rings for each group would be increasing from left to right. The corresponding oil analysis results are plotted in a scale shown on the right side on the vertical axis. It is evident that the present methods of determining the amount of contaminant in the oil give results that are lacking in correlation with the wear in an engine. The example shown here was typical of the lack of correlation that was observed with other measurements in the engines.

Effect of Lead Compounds and Oil Additives on Full-Flow Oil Filter Life – The amount of contaminant load that each filter element can retain has been found to vary only slightly in both laboratory and field test experience. In the case of the full-flow oil filter element, the rate at which the contaminant was introduced into the oil had a direct bearing on the full-flow filter life. The full-flow filter life is determined when the pressure drop across the filter element exceeds 8 psi and the relief valve in the filter body opens to bypass oil.

In Fig. 22 is shown a plot of the typical filter life of full-flow filter elements that were operated under combinations of base oil and base fuel conditions. It can be seen that the additive-type oil increases the filter life when either the base fuel or leaded fuel is used. The suspension of dirt by the detergent additive in the oil prevented the filter

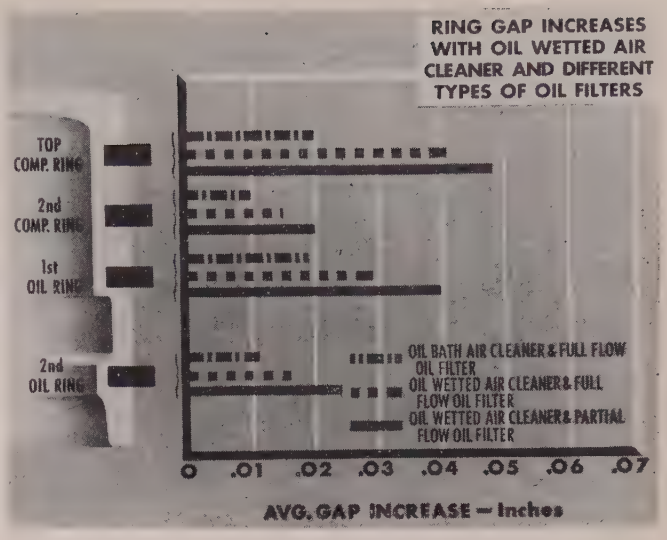


Fig. 18

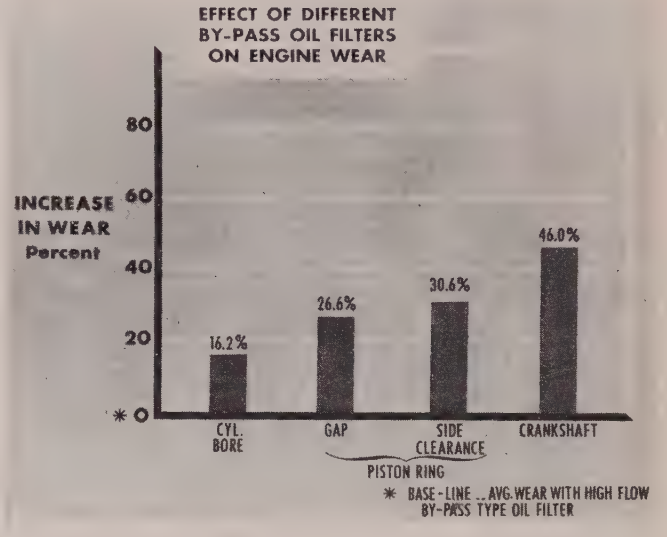


Fig. 19

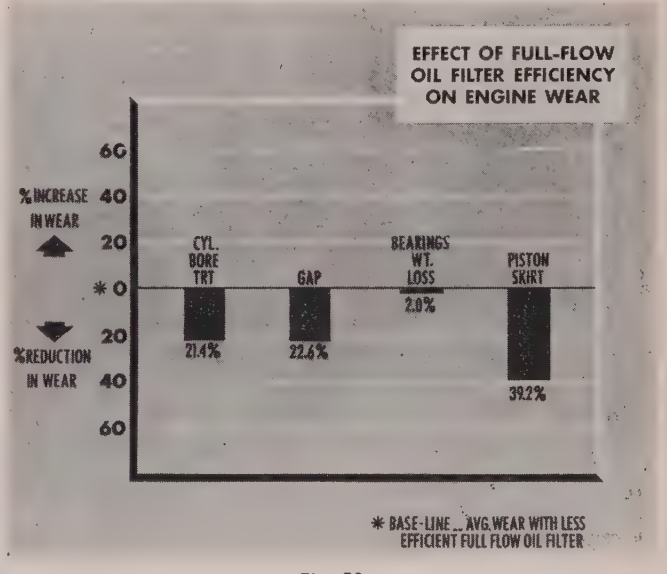


Fig. 20

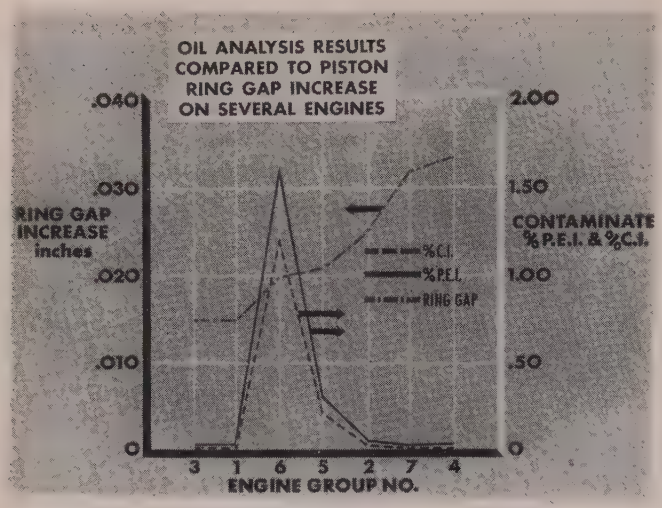


Fig. 21

from removing all of the fine, suspended particles of contaminant. The effect of lead compounds entering the lubricating oil when either the base oil or additive oil was used can be seen by the reduced life of the oil filter elements.

Summation

In an investigation of the internal wear in an engine, we have seen that the lubricating oil plays a very important role. The oil film picks up foreign contaminant particles that are introduced into the engine through manufactured dirt, compounds formed in the process of burning fuel, air entering the engine as part of the combustion process and for ventilation purposes, and products formed from the deterioration of the oil itself, when exposed to high operating temperatures. Whether or not there would be engine wear when we have successfully isolated all of these sources of contaminant, would depend a good deal on the ability of the oil to lubricate, and on the engine design to provide this oil supply in a manner suitable for the lubrication of each part of the engine. The data we have presented shows some of the factors that tend to increase wear in an engine. Since the lubricating oil has played such an important role, we have shown data on the effective control that the use of the different systems of oil filtration can accomplish.

The full-flow oil filter of this era is by far the best method we have seen for restricting wear in an engine emanating from contaminants that get inside the engine. There are limitations on how much the full-flow filter can do today. Some of these limitations are a result of the amount of oil it must pass through the filter medium. Careful attention to the utilization of the oil supply by improved metering methods would lessen this load. A reduction in engine octane requirements, until better and cleaner fuels can be produced in quantity to suit our needs, would lessen the contaminant load and improve piston-ring operating conditions. As these improvements are made, the need for such high concentrations of detergent additives in the oil will disappear, so that more attention can be directed to essential lubrication qualities.

Meanwhile, the effective life of an engine, before it is said to be worn out, will be greatly lengthened, as "engine wear is trapped in a can."

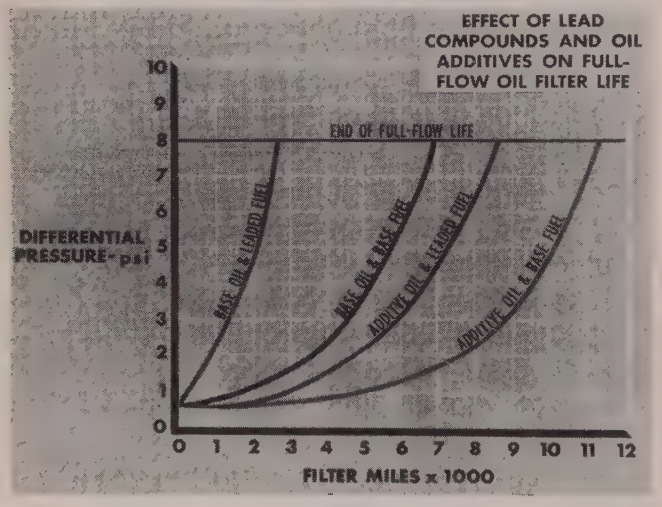


Fig. 22

ORAL DISCUSSION

V. E. Weiss, Standard Oil Co. of Cal.: How was the need for a filter change determined in the comparative fuel-filter studies?

Answer: Filter changes were made at the pressure differential where the relief would normally open.

U. B. Bray, U. B. Bray Co.: Did the additive oil used contain a calcium or a barium detergent?

Answer: The oil contained a barium-type additive with a phosphorous-containing inhibitor.

J. H. Shandorf, City of Los Angeles, Department of Water and Power: What was the physical size of the filter elements used?

Answer: All elements were conventional for the engines used.

L. Ketchan, Seaboard Motors: Was any effort made to establish which size of particle is the wear agent?

Answer: Some information is available in that less wear was obtained with the full-flow filter which removed relatively large particles from a large quantity of oil, as compared to the removal of both large and small particles by the bypass filter which filtered only a small quantity of oil.

B. E. Boswell, Shell Oil Co.: How were the wear measurements made?

Answer: Wear was determined by physical measurements of ring gaps, cylinder bores, bearing journals, and so forth.

Afterburners and Variable Nozzles

Ralph Kress, Solar Aircraft Co.

This paper was presented at SAE National Aeronautic Meeting, Los Angeles, Oct. 3, 1952.

MUCH has been written about the potentialities, performance, and theoretical design of afterburners, but very little can be found on the actual detailed construction and design of this important device. This paper, therefore, although limited in scope by government secrecy provisions, attempts to set forth some of the problems which beset the design engineer when he endeavors to make a practical working machine from the paper calculations.

An afterburner is essentially another complete engine attached to the exhaust of a turbojet engine to provide additional thrust boost. It is used primarily to develop short bursts of increased speed when needed during take-off, climb, and combat maneuvering. In principle, an afterburner is able to accomplish this by raising the temperature of the jet gases after they leave the turbine by burning additional fuel in a specially designed tailpipe. Because an afterburner must perform its function without affecting the normal operation of the main

THIS paper discusses the modern concept of afterburner design, breaking down the various major components and treating, as fully as allowable under secrecy imposition, the problems confronting the design engineer.

Because an afterburner must provide short bursts of speed during take-off, climb, and combat maneuvering, without affecting the normal operation of the main turbojet engine, it is necessarily very complex. Essentially, it is a complete engine attached to the exhaust of the turbojet engine. By burning additional fuel in a specially designed tailpipe, the afterburner raises the temperature of the jet gases after they leave the turbine.

Early designs are evaluated, with advantages and disadvantages pointed out. Actuator power sources are dealt with and suggestions given for future development.

Comparisons are drawn between the nonafterburning jet engine and that incorporating an afterburner.

Where secrecy imposition makes specific discussion impossible, a general introduction to the subject is attempted.

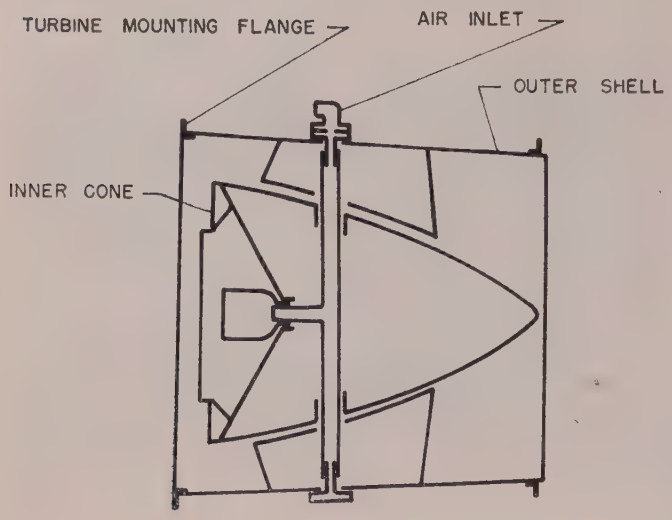


Fig. 1 – Floating inner cone tailcone

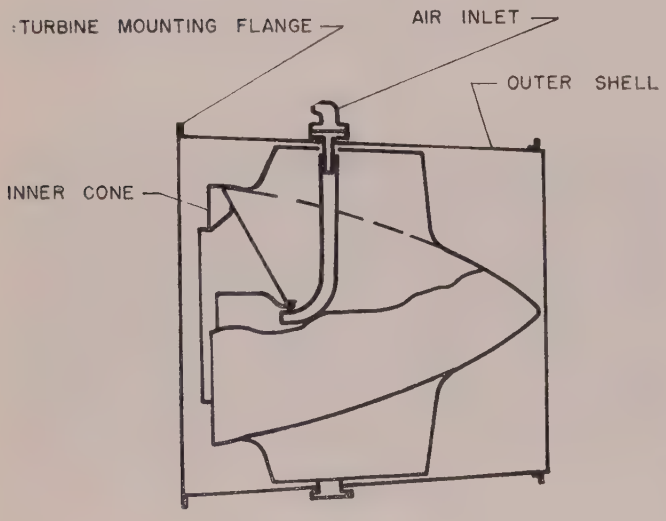


Fig. 2 – Integral tailcone design

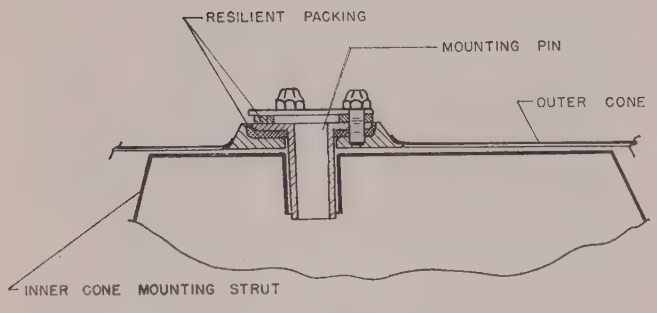


Fig. 3 – Resilient pin inner cone mount

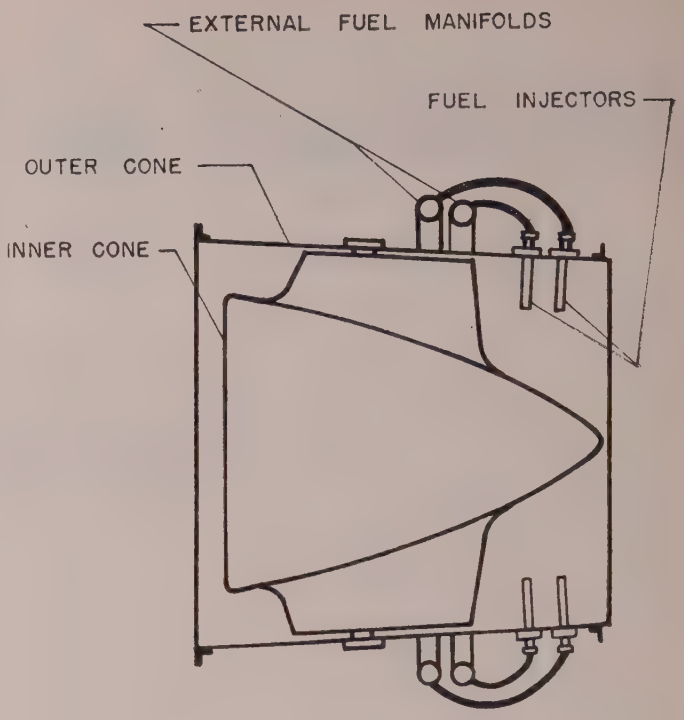


Fig. 4 – Externally manifolded fuel injectors

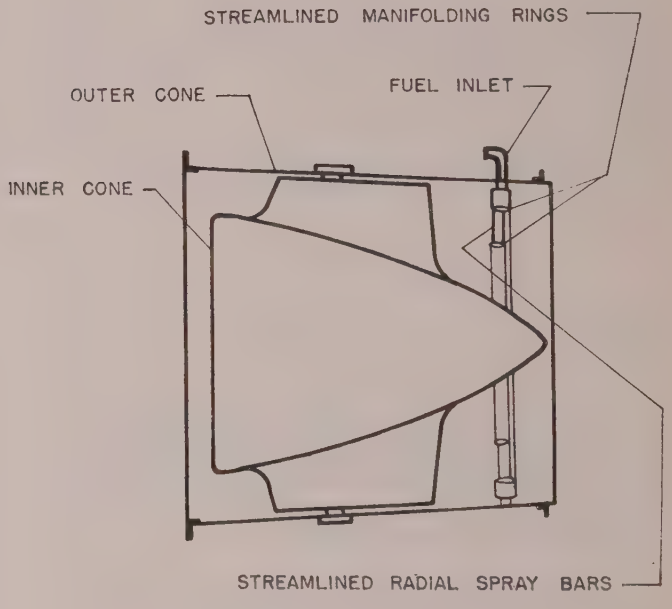


Fig. 5 – Internally manifolded spray bars

turbojet engine, it is much more complex than a simple tailpipe.

An afterburner presents special design considerations. To produce substantial thrust gains, the gases in the tailpipe, after combustion, exceed 3000 F, which is more than 500 F over the melting point of the best alloys. The tailpipe and components create special problems in high temperature engineering, as an afterburner must have complete serviceability and reliability. For example, the variable nozzle must be capable of adjusting the jet area without leakage and distortion while subjected to high temperatures. Careful attention must be paid to detail design to assure cooling of all hot areas because normally the high temperature gases will rapidly melt uncooled surfaces. Smooth, efficient combustion of large quantities of fuel in an afterburner is an achievement in itself, because, in the path of gas velocities of 300 to 400 mph, combustion instability would produce a force which would rapidly cause serious service difficulties.

Comprehensive Development Program

For the past six years, Solar Aircraft Co. has been engaged in a comprehensive development program on afterburners which must be designed for the particular type of engine on which they are to be used. During these six years, the company has successfully designed and manufactured complete afterburning units for virtually all U. S. turbojet engines made to date, accumulating thousands of hours of actual test and flight experience.

Early work dealing with afterburners has been primarily concerned with obtaining stable combustion and performance. In this regard, the development engineer was primarily interested in the burner problem, and, because of this, the early afterburners were made as simple as possible to alleviate the high temperature problems associated with near-stoichiometric burning.

Afterburners that could produce acceptable thrust boost and still have a useful working life have been demonstrated by extensive test flying and are now in service. They have demonstrated to the aircraft industry that they are indispensable for modern high-speed aircraft of certain types.

Major Improvements Made

With the advent of afterburner-equipped aircraft, it was soon evident that major improvements in an afterburner were desirable to improve its installed performance. Steps could be taken toward refinement of the basic afterburner design, improving performance, increasing efficiency, and so forth. A problem of primary importance was that of the variable-area jet nozzle. Airplane requirements dictated a smooth variation of power from normal thrust to full augmented thrust, rather than the surge given by the early 2-position nozzle afterburners. Also, advanced engine designs would give better fuel efficiency by using the afterburner nozzle for no-burning operation. These factors made the fully-variable area jet nozzle a highly desirable objective.

Every detail of the afterburner has been subjected to its share of investigation and it has been shown many times over by sad experience that no part of an afterburner is too small to be considered unimportant.

The afterburner, as a whole, can be conveniently divided into four major components. This not only allows a better insight into the workings of the afterburner, but permits a more systematic analysis of the afterburner design. The major components are as follows:

1. Diffuser.
2. Burner or combustor.
3. Afterburner controls.
4. Variable nozzle.

Diffusers

The gases leaving the turbine wheel in the ordinary jet engine are directed through the shortest possible distance to the engine jet nozzle. This is usually accomplished by means of a streamlined tailcone attached to the turbine flange. The gases leave the turbine wheel usually at near-sonic velocities, which is much too high for afterburning. In order to reduce the turbine exit velocities to their proper values, a diffuser section is required before the burner. General practice is to diffuse gas to a velocity of 400 to 500 fps at the combustion section. This diffuser is usually made integral with the tailcone, in order to minimize the afterburner tailpipe length.

Fig. 1 is a cross-section typical of the many tailcone designs produced today. The tailcone consists of an outer shell with flanges suitable for attachment to the turbine flange and the balance of the afterburner, an inner cone, support struts, and fairings, sometimes an air-ducting system for cooling the turbine wheel, and supplementary bosses and pads required for engine instrumentation and controls. Notice that in this design, the inner cone is mounted off the crossbars which are tied into the

The Author

RALPH KRESS is a design engineer in the Development Engineering Division of Solar Aircraft Co. Prior to joining Solar in 1949, he held positions with Consolidated Vultee Aircraft Corp. and Curtiss-Wright Corp. Mr. Kress is a mechanical engineering graduate of Ohio State University.

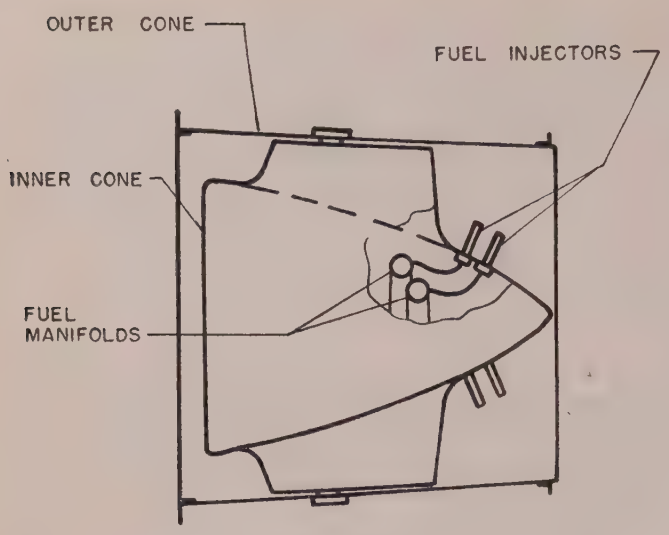


Fig. 6 – Tailcone-mounted fuel injectors

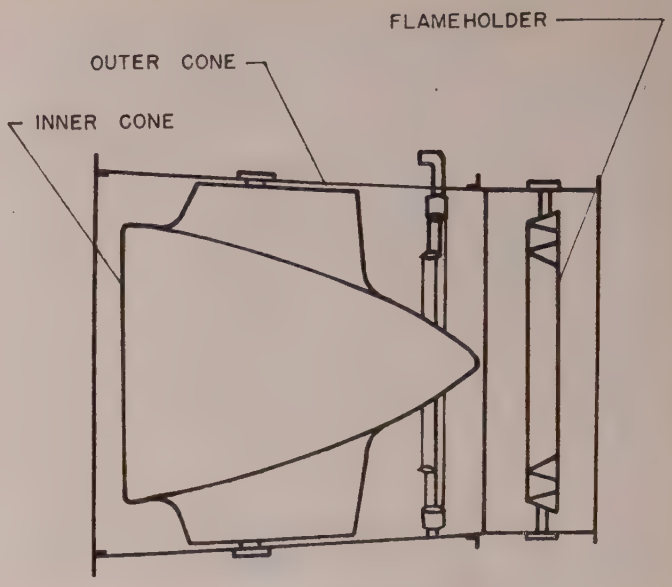


Fig. 7 – Flameholder

outer shell, transferring the cone loads to the shell. The cone is not rigidly connected to the crossbars, but floats more or less to allow differential expansion. The crossbars are covered with strut-like streamlined fairings, which tend to smooth the gas flow.

Integral Tailcone Design

Fig. 2 is a cross-section of an integral tailcone design that has solved many of the operating and production difficulties found in other type cones.

In this design, the tailcone itself is made integral with the cone struts which now become structural members of the cone, instead of fairings, and allow the structural crossbars to be eliminated.

The inner cone is usually retained in the outer shell by means of pins inserted from the outside of the assembly. Liberal clearance is allowed between the pin and inner cone, in order to permit full thermal expansion of the various parts.

The method of attachment of the inner cone to the outer shell has always been a problem, both in no-burning and afterburning tailcones. The actual gas loading involved, due to pressure differential within the assembly, is generally not very high, indicating a relatively light attaching member is required. Experience has shown this is not the case, however. Field and test cell data have given a long history of shell and cone fractures. Thorough investigation and analysis have shown that these fractures are probably caused by the severe pounding and buffeting given to the outer shell by the relatively large and heavy inner cone, acted on by variations in pressure and vibrations during operation. This condition is much more severe during afterburning, and failures have been known to occur after a very short length of time.

Accordingly, a cone attachment was designed that would isolate the inner cone from the outer shell by means of resilient pads of high temperature materials. Fig. 3 is typical of this general design.

Shell Fracture Prevented

The inner cone is retained in the outer shell by means of the usual pins; however, the pin connections to the outer shell are such that no part of the pin actually contacts the outer shell, except by way of the resilient pads. This method has proved very successful and has eliminated the tendency to fracture the outer shell, particularly in afterburner tailcones.

Most modern high performance jet engines have provisions for some type of turbine wheel cooling on the discharge side. This is usually accomplished by ducting compressor discharge air through the tailcone and over the rear face of the turbine wheel. In either of the aforementioned designs, the air is brought in through the pins or tailcone support structure, which is made hollow and fitted with an air connection, then through an internal tube to the face of the turbine wheel. Usually a pan is fitted to the forward end of the inner cone to allow the air to diffuse properly over the face of the turbine wheel.

Tailcones are welded assemblies usually made of AISI type 321 stabilized, corrosion-resistant steel. However, the afterburner design sometimes dic-

tates the longitudinal or beam stresses in the outer shell, especially in cantilevered designs. In this case, the longitudinal stress (factored for g loading requirements of the airframe) indicates a higher strength material than 321. Inconel W and N-155 are typical materials used in the outer shell in this case.

Inner cone materials are also of 321 corrosion-resistant steel, except in cases where burner designs enter into the tailcone configurations. The loads taken by the inner cone are usually light and are primarily due to the differential gas pressure existing across the cone. Buffeting and impact loads are alleviated by the use of the resilient cone pin support mentioned before.

Burners

The burner section is that part of the afterburner that takes the relatively high oxygen content exhaust gas from the turbine wheel, injects fuel into it, and burns the mixture in order to raise the energy level of the gas stream. It usually consists of two major components: the fuel injection system and the flameholder.

Often the burner section is incorporated within either the diffuser or the nozzle section in order to save weight and complexity. However, for ease of maintenance and replacement, it is more convenient to build the burner as a separate section.

In general, the fuel injection system consists of a number of spray bars protruding into the gas stream. These spray bars are perforated along their length to allow fuel to be injected into the gas stream; the perforations being adjusted to give a particular fuel pattern throughout the tailpipe. The spray bars are usually supplied with fuel by a tubular manifold. With dual injection systems, more than one manifold is required, manifolding of the fuel injectors being accomplished either

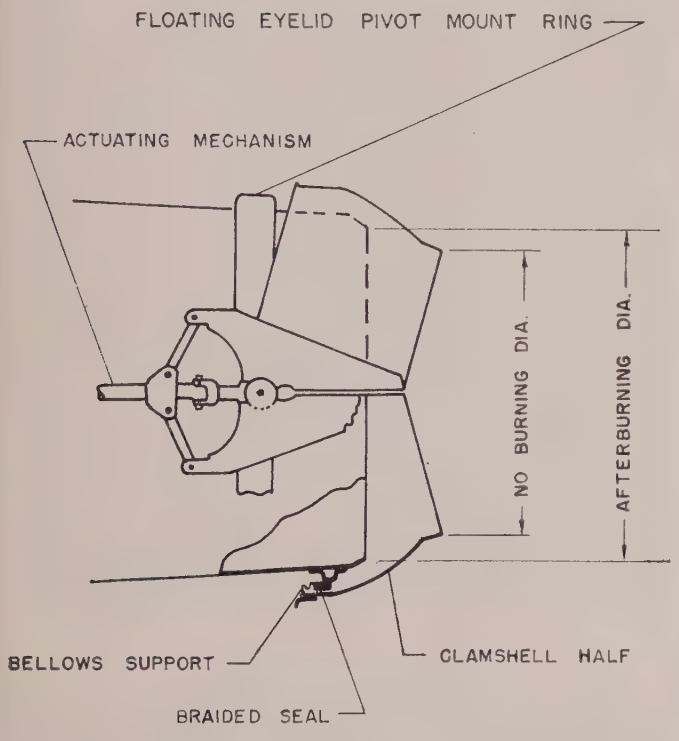


Fig. 8 – Variable nozzle – 2-position clamshell

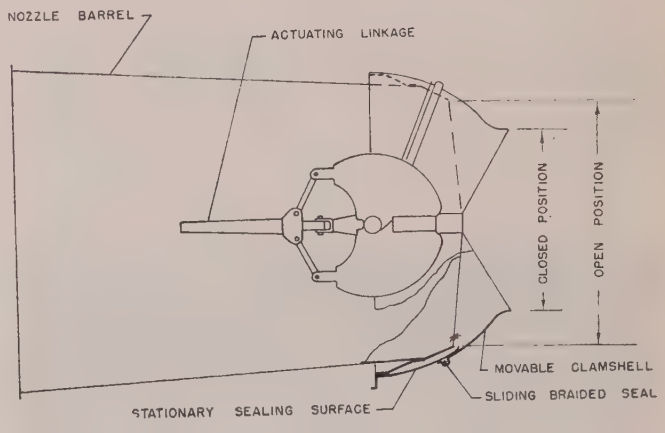


Fig. 9 – Fully variable clamshell nozzle

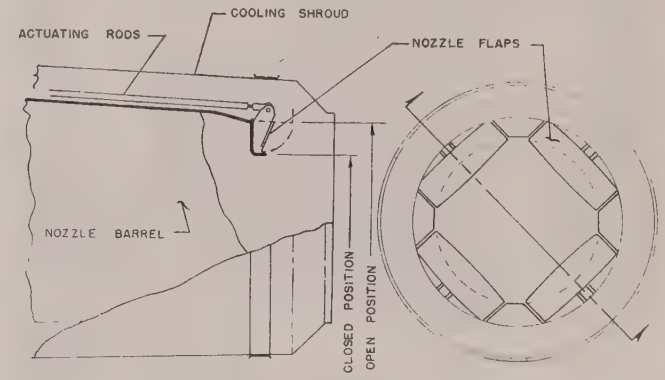


Fig. 10 – Sketch of 4-flap variable nozzle

internally or externally. The external manifolding, Fig. 4, has the advantage of leaving the gas flow area clear, but due to the many connections between spray bars and manifolds, joint leakage is possible, and, of course, a serious fire hazard. Internal manifolding leaves only one or possibly two connections for fuel to the afterburner, thereby reducing the possibility of fuel leakage to a minimum. All joints on the internal manifold are welded, and if cracks do develop, the fuel simply leaks into the gas stream, which of course presents no fire hazard.

The internally manifolded fuel injector shown in Fig. 5 has the problem of differential expansion within itself. This is due among other things to nonhomogeniety of the temperature profile across the gas stream coming out of the turbine, and the temperature differentials existing within the manifold due to fuel flow. In smaller injector rings, this problem is neglected and materials of high enough strength are used to withstand the thermal stresses. In the larger rings, however, this design leads to short life, so provision, such as slip-pin mounted crossbars, must be made for thermal expansion in the manifold itself.

Fuel Leakage Problem

Designs have been made incorporating the fuel injectors in the tailcone, thereby achieving a shorter overall length of the afterburner. In this case, manifolding of the injectors is accomplished inside the tailcone, thus alleviating the fuel leakage problem. Such a system is outlined in Fig. 6. Fuel leakage would not be dangerous from the standpoint of the engine compartment; however, raw fuel within the tailcone constitutes a combustible mixture which may burn and deform the internal cone.

Fuel injector size is selected to pass the fuel at an optimum internal velocity such that tendency to boil and coke will be minimized, and yet not produce excessive pressure drop. Cross-sections of the injectors are usually circular, but resultant gas flow turbulence sometimes causes attachment of flame to the injector, giving rise to serious combustion disturbances with subsequent failure of the injector and other parts of the afterburner. Aerodynamically streamlined injectors have been developed which almost entirely eliminate this problem. Aspect ratios leading to low drag sections are practical, but care must be exercised to see that the sections will withstand the internal fuel pressure without permanent deformation since pressures may reach 500 to 600 psi.

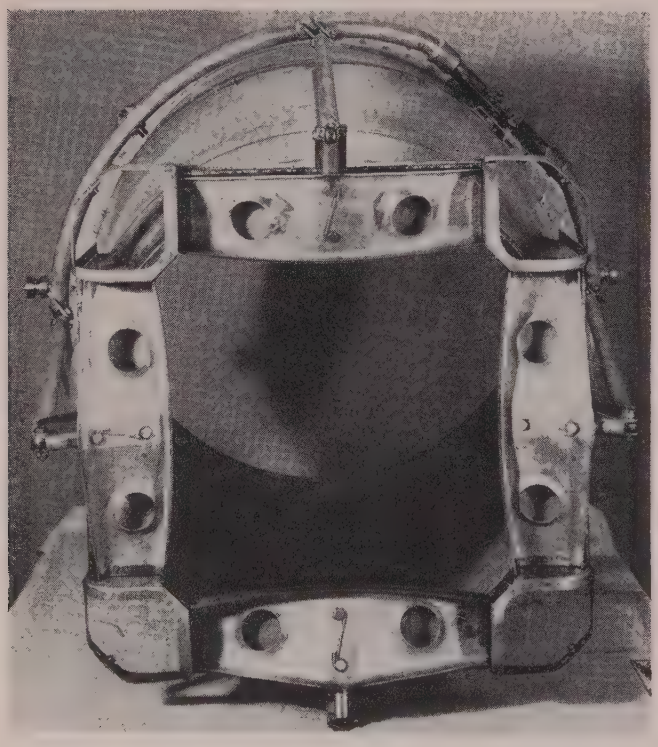


Fig. 11 – Photograph of 4-flap variable nozzle

Flameholder

The flameholder accepts the fuel-air mixture from the fuel injector, and, by providing an area conducive to ignition and flame propagation, maintains a stable flame front across the tailpipe. By virtue of this requirement, the flameholder is necessarily a turbulence-producing device which results in pressure losses from gas stream drag; therefore, extreme care in the design is necessary lest flow losses cause substantial thrust loss when the afterburner is not operating and may seriously reduce thrust gain when it is operating.

The flameholder, as shown in Fig. 7, is mounted downstream of the fuel injector at the distance required for proper fuel dispersion. It may be supported from the afterburner skin, in which case allowances must be made for differential expansion between the flameholder, which reaches very high temperatures, and the outer shell, which remains relatively cool. This is usually done by pins, installed externally, through threaded bosses, to a loose slip fit on fittings mounted off the flameholder.

Avoidance of Hot Streaks

The struts connecting the flameholder to the outer shell must be carefully streamlined in order to reduce turbulence which tends to propagate flame, causing hot streaks on the skin of the tailpipe. In all cases, any connections or protuberances into the gas stream must be examined with care in order to eliminate hot tailpipe streaks. Hot streaks in skin temperatures deform the tailpipe, deteriorate the nozzle, and shorten the afterburner life.

In order to alleviate the hot streaks due to

struts, flameholders have been mounted off internally manifolded fuel injectors, and also off the inner cone. In either case, the subject parts must be strengthened to take the heavier loads induced by the violent combustion forces acting on the flameholder.

Loads on the fuel injectors and flameholders are imposed chiefly from velocity of gases and are relatively light. The fuel injectors are usually made up of AISI 321 corrosion-resistant steel. Flameholders, because they are exposed to flame and operate at high temperatures, are made of N-155 or other high temperature alloys. Because of the burning problem, rather heavy gages are indicated for the flameholder. Flameholders are usually fabricated sections, although cast sections have been used.

Nozzles

The nozzle is that part of the jet engine which converts the available pressure energy in the tailpipe to velocity energy, thereby achieving a momentum change resulting in propulsion force. In a jet engine without afterburner, the nozzle diameter is chosen to give the best operating efficiency over a wide range of flight conditions. In an afterburning jet engine, the diametral span required of the nozzle by the afterburner is so great that a variable nozzle is used.

Early afterburner nozzles were of the 2-position type, that is, either the afterburner was fully on or off. Only one afterburner nozzle area was available which was usually for the maximum possible thrust.

Fig. 8 is a diagram of an early 2-position clamshell-type nozzle. Two stainless steel assemblies similar in shape to clamshells are hinged at the end of the tailpipe. These clamshells are so arranged that they restrict the flow in the closed position, but swing out of the way of the gas stream in the open position, thereby achieving the change in nozzle area required for afterburning operation.

Fig. 9 is an adaptation of the clamshell nozzle to fully variable operation. Very high actuating friction and imperfect sealing, because of distorted parts, resulted in poor performance and rapid deterioration.

Nozzle Requirements Analyzed

After preliminary testing and analyses of the fully variable clamshell nozzle, an investigation was initiated on the design of a fully variable, so-called flap-type nozzle. Preliminary studies showed good possibilities for a design that would be relatively free of distortion and heat problems, and could also be adaptable to production processes. Aerodynamically, the designs were sound, as the nozzles showed excellent orifice coefficients with round planar, jet orifices.

Fig. 12 – Variable bulb nozzle (open)

As a preliminary study, a nozzle was designed that incorporated four large flaps. These flaps were designed so that no seals were necessary and clearances were adequate at the running temperatures. Load and stress studies were made showing that the loading would not be excessive, though too high for the simple air-cylinder actuator then in use except for test cell operation.

Fig. 10 is a cross-section of this design and shows the type of orifice achieved in the various positions. The actual orifice varies from approximately a square shape in the closed position, to a circular shape in the open position. The flaps were actuated by means of push-rods connected to synchronizing yokes on the forward end of the nozzle section. In the closed position the flaps projected into the gas stream at right angles. In order to make the orifice losses as low as possible, the gas stream end of the flap was generously rounded.

Fig. 11 is a photograph of the actual, full-scale working model of this nozzle design. The entire unit was of sheet metal construction, arc and spot-welded together.

Nozzle Test Results

All parts for the nozzle barrel, including the hinges, were of 321 corrosion-resistant steel. The flaps were spotwelded assemblies made of N-155. The hinge design was of the familiar piano-hinge type, each half-section being spotwelded in place. Since all leakage from the hinges and flaps was parallel to the gas stream, it was felt that no seals would be required.

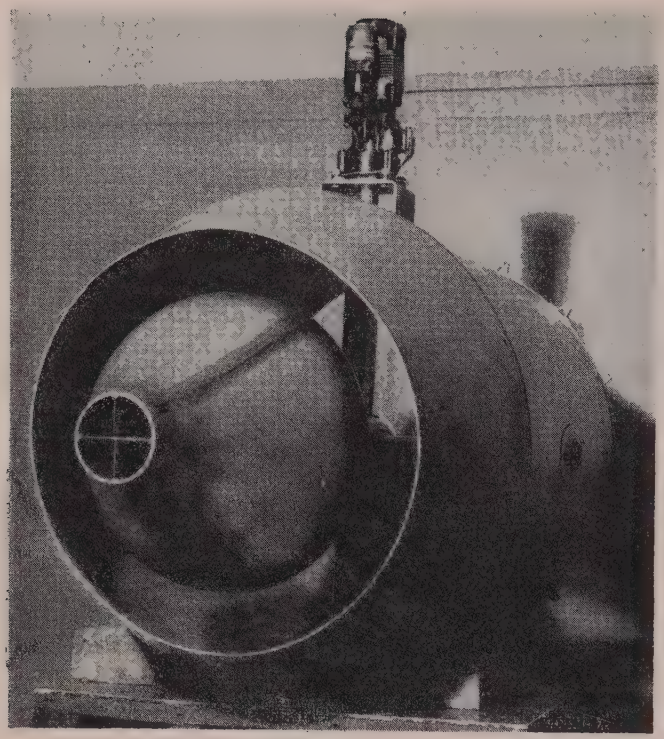


Fig. 13 – Variable bulb nozzle (closed)

The nozzle was run on the test stand for quite a few afterburning and no-burning hours without showing any great degree of deterioration. Test results showed low nozzle losses with slightly better efficiency in the more open position.

The major disadvantage of this particular design was the relatively large clearance diameter required in the airplane at the jet orifice. Since the airplane base drag is a function of the base annulus area between the jet stream and the airframe air stream, it was felt that further work should tend toward making nozzles more dimensionally efficient. The type of nozzle that varies the tailpipe area by means of a bulb moving in and out of the tailpipe exit, has been given much thought. The bulb-type nozzle has been built and has shown a measure of success when used in a nonafterburning engine. However, when used in the intensely hot stream of the afterburner, the actuation problem, together with the structural loads induced in the bulb, presents an extremely difficult problem.

Figs. 12 and 13 are photographs of a working afterburning bulb-type nozzle in the retracted bulb or open position, and the extended bulb or closed position respectively.

Multiflap Nozzle

Further investigation and study on the nozzle problem led to a requirement for a larger number of individual flaps which would give a more circular orifice and be planar in all positions. This would also allow a nozzle flow shape that was conducive to high nozzle coefficients. Various configurations were studied and analyzed as to flap design, flap actuation, flap gas load, and actuator load, with rigorous stress analyses of all structural parts.

Numerous multiflap nozzle designs have been developed over a period of several years. Many of these have been manufactured and have accumulated many hours on the test stand in both afterburning and nonafterburning conditions. They have proved that the multiflap nozzle can be an extremely rugged device and that it can be easily manufactured. It is indeed unfortunate that security impositions make any detailed discussion of these nozzles impossible.

Actuator

An important link that connects the afterburner to the control system is the actuator. Its function is to position the afterburner nozzle to the areas required by the control system.

Early afterburner nozzles of the clamshell type which had only two positions, either fully open or fully closed, required an actuator adjustable to only two positions. The nozzle was aerodynamically balanced, so most of the actuator force was spent in seating the eyelids against the closed position seal, and pressurized air from the compressor usually provided sufficient force for this purpose.

The advent of the variable thrust afterburner with its fully variable-area nozzle made necessary an actuator which was infinitely adjustable from fully open to fully closed, and capable of locking at any point in this range. Greater power was a prime requisite of the fully variable actuator because, unlike the balanced clamshell nozzle, the variable flap-type nozzles are aerodynamically unbalanced, and in operating them, the actuator must overcome both friction and the internal gas pressures acting on the nozzle segments.

Actuators in general can use any of a number of sources of power. In an airplane and on a jet engine, actuators are limited to four practical sources of energy. These are electricity, hydraulic pressure, lubricating oil pressure, and pneumatic pressure. Each has its advantages and disadvantages, none of them being ideal.

Electrical Actuator

An electrical actuator is a natural choice for an afterburner. It has been made for many years in all shapes and sizes, and for a variety of operating conditions. Electrical energy is readily available on all aircraft and is easy to control. Electrical actuators are easily designed into electronic control systems. By necessity, however, the actuator must be placed near the variable nozzle, and because of the high ambient temperature found in this area, the electrical actuator requires a com-

plicated cooling means. The power requirements of flap-type nozzles are from about 3 hp and up. This is probably the most serious drawback of the electrical actuators. Actuators of this power are not known to be flyweights. Because aircraft designers are extremely weight-conscious, the electrical actuator has been limited more or less to the balanced clamshell nozzle.

Most modern aircraft are equipped with hydraulic systems and require a hydraulic pump on the engine. A few of the jet engines themselves are equipped with integral hydraulic systems in addition to the aircraft system for air inlet screen and no-burning nozzle actuators. All jet engines are equipped with pressure lubricating systems. Any of these are sources of energy for afterburner nozzle actuation. Hydraulic systems are high pressure systems and therefore can provide high actuating forces with small actuators of light weight. Actuators using lubricating oil pressure can produce limited forces because of the relatively low pressure available. The light weight advantages of hydraulic systems make them very attractive for nozzle actuation; however, they do have certain disadvantages. Particular care must be used in all joints and packing design because of the fire hazard. All lines and devices subject to leakage must be kept clear of the hot afterburner surfaces. The high ambient operating temperature of the jet engine is a serious problem in the operation of many types of control valves used in conjunction with the hydraulic actuator. The hydraulic fluid itself could boil or decompose under high temperatures. With the new high temperature hydraulic fluids, this problem could be alleviated.

Auxiliary Power

All gas turbine engines are equipped with a source of auxiliary power that is rather unique as a prime mover. Relatively high pressure air may be tapped off of the compressor section of the engine and used as a source of power for accessories. Quite a bit of air may be used before serious loss of power is noted in the jet engine. Compressor discharge air is attractive as a power source for actuators for many reasons:

1. The air is always available for use as long as the engine is running.
2. The exhaust air may be dumped almost anywhere and not do any harm.
3. Air constitutes a one-line system; that is, no reservoir or fluid return lines are necessary.
4. It is unaffected to any great extent by high temperature.
5. The design of packings and seals could be relatively simple as leakage would introduce no safety problems.
6. Air, being a compressible fluid, is a source of potential energy. It therefore, not only can be used to transmit energy, but also to store it.

The disadvantages, although serious, can be circumvented:

1. The available compressor discharge pressure on most jet engines is relatively low in comparison to a standard hydraulic system. This available pressure is also a function of altitude and air speed.
2. The ability of air to compress and store energy is also a disadvantage. The fact that air compresses, means that it acts similar to a spring and when used in actuating cylinders, results in springy or bouncing action of the piston. Positioning of the piston, as known in hydraulic servo cylinders, is very difficult to do.

Unique Challenge

A method of stabilizing an air cylinder that makes it fully positionable and in all respects makes it act like a hydraulic cylinder, has been

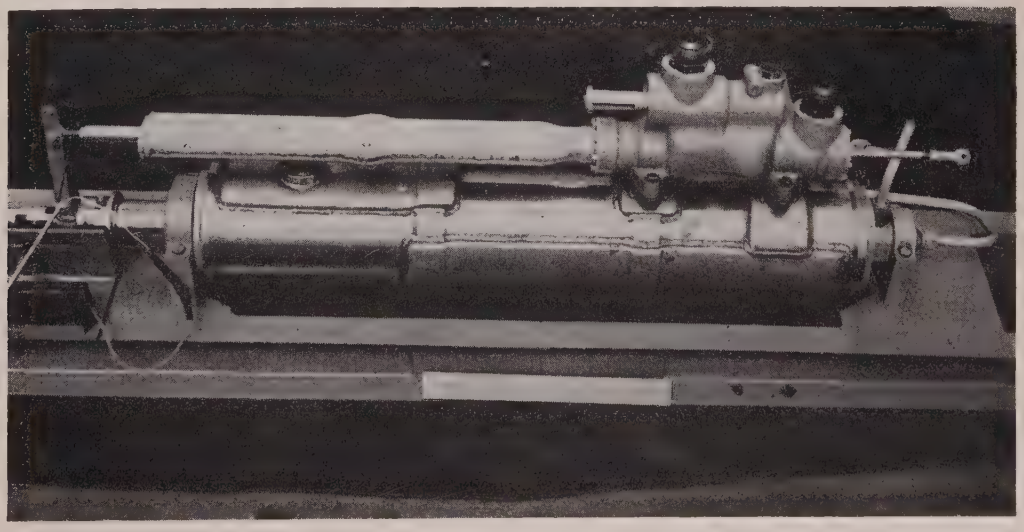


Fig. 14 – Afterburner nozzle actuator

worked out. This design incorporates an integral sealed self-contained hydraulic system within the actuator. An actuator of this type has been designed by Solar Aircraft Co. and proved practicable.

Fig. 14 is a photograph of the finished actuator. This actuator is of the servo type; that is, it takes a linear signal on an input rod, requiring a negligible force and amplifies it on the output side to the required actuating force. The actuator was designed to operate at an ambient temperature of 400 F. The hydraulic system, including fluid and seals, was composed of materials capable of withstanding this temperature. The pneumatic section of the actuator was fabricated of stainless steel and aluminum with special piston rings designed for lubricationless operation. In its entirety, the design stressed aircraft quality with as low a weight-output ratio as possible.

The afterburner as a whole offers the designer a challenge that is unique among machines. Very few devices proffer such extreme conditions that practically all available sources of engineering information and research are touched before solution to the problem is seen.

It is hoped that this attempted clarification of the present afterburner design conception will spur others to ingenious solutions of this most vexing problem.

ORAL DISCUSSION

Reported by J. B. Pitkin,
Lockheed Aircraft Corp.

H. W. Withington, Boeing Airplane Co.: Another problem connected with the use of the afterburner is the noise and vibration resulting from the high energy level of the afterburner. For a given pressure ratio it has been noted that the nozzle shape markedly affects jet pattern. Convergent nozzles produce normal shocks when the exhaust velocity is above sonic; convergent-divergent nozzles produce weak shocks. Schlieren photographs of the normal shocks of a convergent nozzle indicated a disturbance which when analyzed was found to include a frequency coinciding with the noise frequencies.

Supersonic applications require variable convergent-divergent nozzles. Two-position nozzles designed for one point have poor off-design performance. A variable convergent nozzle may give better performance.

Mr. Kress: It has been noted that "screech" occurring in the combustion process improves combustion efficiency. However, the noise is intolerable. The problems of the variable convergent-divergent nozzle have been examined. However, no additional comments could be made at this time.

Frank B. Lary, Wright Aeronautical Corp.: The performance of afterburners needs to be examined from the standpoint of the troubles that may be encountered with the type of configuration shown. First, it should be expected that because fuel flow varies approximately as the square root of the pressure available, proper combustion in the afterburner would be improbable over the complete altitude range. Second, it should be expected that having the fuel manifold inside the afterburner would result in a coking problem. Third, it should be expected that a plug-type nozzle would improve the base drag of the airplane. Has the plug-type nozzle been tested with the plug cantilevered to the engine?

Mr. Kress: First, the fuel-flow problem at altitude is accompanied by other engine altitude problems. One of the satisfactory solutions to this fuel-flow problem is the use of the duplex nozzle which closes off one fuel discharge orifice under low fuel-flow conditions. Another satisfactory solution is the use of group injectors used in the afterburner. At altitude, some groups may be turned off so that remaining groups can provide an adequate pattern.

Second, the fuel manifold inside the afterburner has been scavenged with engine bleed air after afterburning to stop coking. It has also been found that adjusting the fuel-injector size usually stops coking.

Third, the plug nozzle, when supported from the tail cone of the engine, was found to run satisfactorily in a non-afterburning condition. When run in afterburning the plug went down the pipe. Although less airplane base drag would be encountered with a plug-type nozzle, it is considered that the effect of making a plug-type installation on the engine would actually be detrimental to the engine, to the extent of merely transferring the drag loss associated with the present clamshell nozzle from the airplane to the engine.

A New Look at the Scoring Phenomena of Gears

B. W. Kelley, Caterpillar Tractor Co.

This paper was presented at the SAE National Tractor Meeting, Milwaukee, Sept. 11, 1952.

UP to recent years occasional gear-scoring problems were subdued without much technical difficulty, although always with costly production changes, by modifying gear geometry in a trial and error manner or by changing oil characteristics. However, scoring is becoming a real problem now that the engineer prefers a lighter nonadditive oil. He prefers this type of oil because:

(1) It facilitates cold starting and gives protection before normal operating temperatures are reached.

(2) It is always convenient, and sometimes necessary, to use a common light lubricant in connected mechanisms such as a transmission-torque converter combination.

(3) High operating temperatures require oils that have maximum stability, a property which is often not a characteristic of gear oils having maximum ability to prevent scoring.

(4) Requirements for decreasing gear and bearing wear normally include a filter which, at the present time, restricts the use of gear oils containing certain additives. More accurate formulas are needed to cope with the lesser ability of light oils to provide scoring protection.

A scored gear will not put a machine immediately out of operation but the damage caused by the apparently superficial failure is threefold. First, it produces a noisy pair of gears and no one dislikes noise as much as a driver. Second, scoring

THIS paper points out that the advantages of using light, nonadditive oils are often sacrificed because of lack of fundamental knowledge about gear-scoring problems. Most formulas that have been developed to determine the scoring resistance have been totally empirical and have proved inadequate for stringent design requirements.

The author discusses the excellent correlation that exists between scoring test results and a hypothesis on the failure of straight mineral oils. This correlation also encompasses the test results of ball and roller scoring test machines, showing the probable universal application of the hypothesis.

The method of approach to the scoring problem of gears as discussed in this paper is a fundamental one, which combines the factors affecting the conversion of frictional energy into surface temperature with gear tooth geometry, stiffness, and surface finish, and points a way to design gears of higher scoring resistance.

B. W. KELLEY has been with the Caterpillar Tractor Co. since 1946. He was first with the Field Test Division of the Research Department and then with the Research Laboratory, where he has specialized in gear research for the past four years. Mr. Kelley is a civil engineering graduate of the University of Illinois. During World War II he spent two years overseas as a heavy equipment officer with the Seabees.

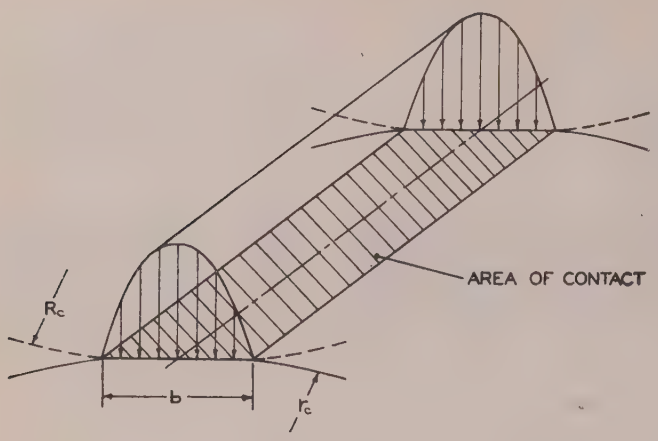


Fig. 1 – Elliptical load distribution and band contact pattern for theoretical development of Blok's temperature formula

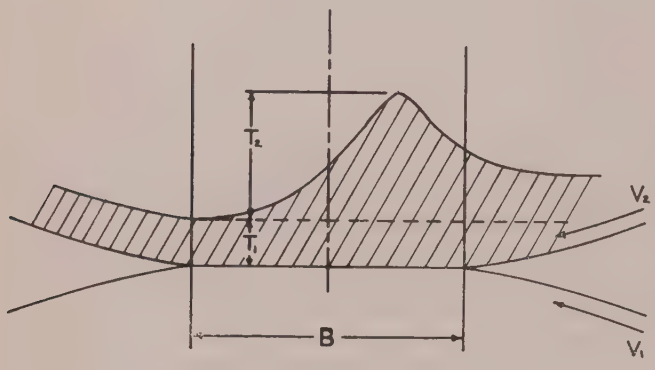


Fig. 2 – Surface temperature pattern in band of contact

releases small particles of metal that attack oil systems and aggravate the wear of bearings and other gears in the same oil supply. Third, scoring destroys the gear-tooth profile, increasing its sensitivity to pitting and tooth breakage.

Characteristics of Scoring

The phenomena have been called a variety of terms such as roping, burning, scratching, abrasion, galling, and abnormal wear. These terms are often misused but they commonly indicate the appearance of the scored area. The different appearances are the result of the characteristics of the material, the loads at which scoring occurred, and the properties of the oil. Many times they result from a single cause, however – failure of the lubricant.

It is necessary, before continuing, to define the scoring that we are concerned with here. Two of the terms mentioned, scratching and burning, are very mild cases of the failure. Scratching may be described as a few light lines, indicating mild seizure at random locations on the tooth face. Naturally, the direction of these lines is in the direction of sliding. This is not a form of scoring which has been recognized as being detrimental. The high points on the surface that cause this scratching are eventually leveled off and the score marks will tend to heal over with no apparent ill effects. Burning is indicated by a darkly discolored area, normally in the addendum of the tooth where no gross amount of material has been removed. This discoloration generally precedes the actual welding and true scoring of the material. It is a danger sign, but although there is some evidence of its causing somewhat more than normal wear, it will not be considered serious for our purposes.

Scoring, as it is considered in this paper, is a welding and tearing action resulting from metal to metal contact, which removes material rapidly and continuously as long as the loads, speeds, and other operating conditions remain the same. This is the type of scoring that causes noise, aggravates pitting, and eventually leads to the complete destruction of the gear. Naturally, this is the form of scoring we should design to avoid.

The most important reason why scoring phenomena have not been adequately expressed in a formula is due to our lack of specific knowledge of lubrication. If the physicochemistry of lubrication were fully understood the approach to the problem would be more obvious. Instead, the designer has been forced to rely largely upon purely empirical formulas. None of these has been universally accepted by gear designers because of the lack of precise test data. Supporting data for previous formulas has been meager, or has not covered a wide enough set of variables. This makes purely empirical criteria acceptable only within narrow limits of gear design.

Most of these formulas include some relation between pressure and sliding velocity. The use of two variables simplifies calculation but causes inaccuracy. The seven factors listed below are considered as having significant effect on scoring resistance.

1. Pressure.
2. Absolute surface velocities and the resultant relative sliding velocity.
3. Viscosity and composition of the oil.
4. Temperature of the oil bath.
5. Properties of the material.
6. Surface finish.
7. Surface treatment.

Since these factors are all common variables of modern gearing practice, our formula must take proper cognizance of them to be an effective design tool.

Scoring may be looked upon as a combination of two separate and distinct phenomena. The first one is the failure of the oil film in the contact area. The second, which is visible evidence of the first, is the metallic welding and tearing that occurs.

There is no contention here that full, thick film lubrication exists between gear teeth, for in most cases it does not. Nevertheless, a supporting layer of oil in some form, such as an adsorbed film, must exist, and its destruction in the area of contact is almost instantaneous when final scoring occurs.

In an analysis of scoring, a pair of gear teeth may be likened to two rollers pressing against each other at some area of contact on their periphery, rotating in such a manner as to simulate the combined rolling and sliding of gear teeth. Assuming that the rollers are of a known material, having a known surface condition and finish, and that the oil which is lubricating them is of known viscosity and quality, three variables remain which will score the rollers – namely, contact pressure, surface velocity, and oil supply temperature.

Although surface temperature, as a result of sliding and pressure, has been implied in most formulas, it is not adequately treated, and the effect of bulk oil temperature has been largely ignored. It will be shown in this paper that temperature alone may be the ultimate basis for the failure of straight mineral oils.

Surface Temperature

In 1937, H. Blok (at that time research engineer with the Shell laboratories in Holland, and now professor of mechanical engineering at Technical University at Delft) developed a theoretical formula for instantaneous surface temperature that he called a "temperature flash." It is based upon the conversion of friction energy to heat. In the Delft publication[1] which presented this interesting work, Blok postulated that straight mineral oils have a critical temperature at which they fail, dependent only upon their viscosity grade. His postulate included the premise that such critical temperatures would limit the use of mineral oils on gears.

A brief summary of Blok's work follows:

A band-shaped contact pattern is assumed, having a parabolic load distribution such as in Fig. 1. The parabolic distribution is chosen in place of the actual elliptic one because of ease of calculation. The error is not significant for our purpose. A surface temperature pattern is formed in the contact area due to sliding, as in Fig. 2. Notice that this instantaneous surface temperature change T_2 is added to the bulk temperature of the material T_1 to form a final surface temperature T_t. It is interesting to note that the peak of the temperature pattern lags the center line of the contact band width b, the amount being dependent upon the surface velocities. This heat penetrates only a small distance into the material, and the temperature quickly returns to the blank temperature as contact passes. Blok's original formula appears below. For consistency throughout the paper the liberty has been taken of changing some of the nomenclature to fit commonly used terminology in the United States.

$$T_2 = K \frac{f W_n (V_1 - V_2)}{(C_1 \sqrt{V_1} + C_2 \sqrt{V_2}) \sqrt{\frac{b}{2}}} \qquad (1)$$

where:

K = Constant[1]

f = Coefficient of friction

W_n = Normal load per unit length

V_1 and V_2 = Surface velocities

C_1 and C_2 = Constants of the materials which include their thermal conductivities, specific heats, and densities.

b = Width of the band of contact as calculated by the Hertzian elasticity formula for cylindrical objects.

It is important to note that the formula includes three of the seven variables mentioned as having an effect on scoring: load, velocity, and character-

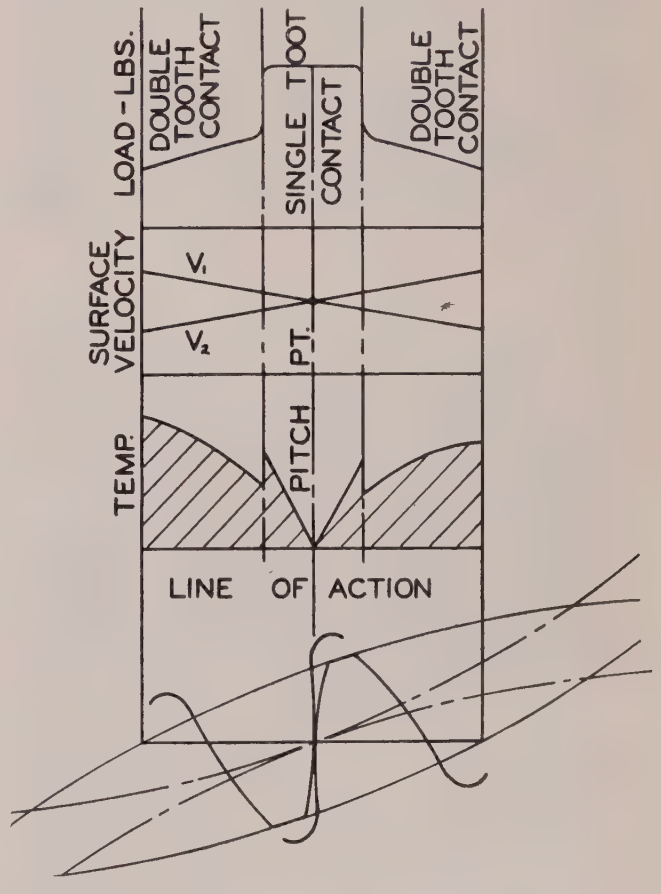


Fig. 3 – Load, surface velocity, and surface temperatures of gear teeth having perfect involute profiles and perfect spacing

[1] See "Les Temperatures de Surface dans les Conditions de Graissage sous Pression Extreme," by H. Blok. Second World Petroleum Congress, Paris, June, 1937.

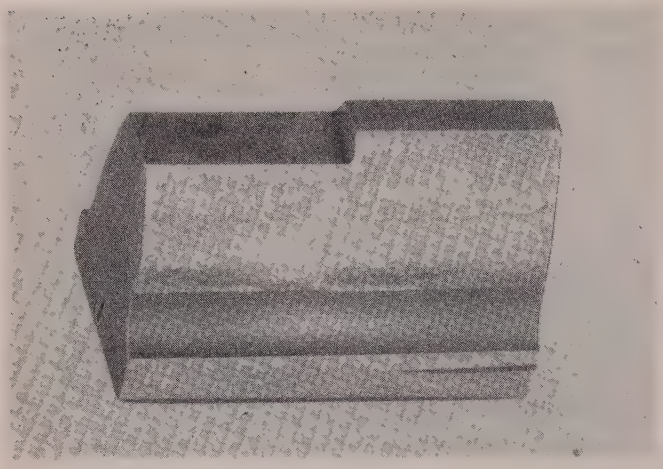


Fig. 4 – Scored gear tooth face – test run with gears in half-mesh

istics of the material. If we include in our calculations the bulk temperature T_1 then the formula appears as below:

$$T_t = T_1 + T_2 \qquad (2)$$

Where T_t is the final temperature at the surface; thus we have included four of the seven factors of scoring.

Characteristics of the material included in the formula are not meant to give a complete picture of the load-carrying ability of mechanisms such as worm gears. The success of such gears is largely dependent upon physical properties of the material, such as the ability to conform to a mating surface, and possibly some factor such as compatibility of materials.

By making thermoelectric temperature measurements on spur gears Blok was able to prove the proportionality of the instantaneous temperature formula.[2] In 1939, Blok, on an analysis of the well-known Boerlage 4-ball extreme pressure lubricant tester, justified the application of the critical temperature hypothesis to that machine.[3] In 1950, T. B. Lane showed that the critical temperature postulate worked on a simple 2-ball scoring test machine.[4]

Attempted use of the hypothesis on gears was not very successful, although Blok showed that it could be generally applied to data published by J. O. Almen in 1935.[5]

Application of Formula to Gears

A certain basic error has been evident in previous investigations of scoring on gears. Scoring occurs in either or both the addendum and dedendum of the gear. Investigation, therefore, is somewhat confined to a period in the mesh of a gear tooth when two pairs of teeth are sharing the load, or the range of double tooth contact at the beginning and end of mesh. When considering the application of any formula to gears there is little justification in assuming that the pressure applied at the tips of the teeth depends only upon the applied tangential load or torque on the gears. Empirical formulas will almost always ignore load sharing in this range of double tooth contact. Thus, such formulas cannot be expected to work satisfactorily for sets of gears which do not have exactly the same errors and geometrical characteristics.

A pair of gears having perfect involute profiles, perfect spacing, and perfectly uniform load distribution across the face of the teeth may be analyzed as follows:

Considering the flexibility of the teeth, a load curve is obtained on a single pair of teeth as they go through mesh similar to that shown in the top diagram in Fig. 3. Notice that the smallest load exists at the start and end of contact since the combined deflection is greatest at the tips of the teeth.

The chart just below this on the same figure shows the surface velocities of the two teeth as they pass through mesh. These velocities are instantaneous velocities of points tangent to the profiles at the points of contact. The difference of the two velocities is, of course, the commonly used term – sliding velocity.

With this information, assuming a constant coefficient of friction, the instantaneous surface temperature can be calculated for all points on the profiles of the mating teeth. The resultant temperature pattern is shown in the lower chart. Notice that on each side of the pitch point the temperature increases rapidly to the ends of single tooth contact. A sudden drop in temperature occurs when two pairs of teeth share the load; then a much more gradual temperature rise takes place to the end of mesh. This latter portion of the instantaneous temperature curve almost flattens out near the tips of the teeth.

There are several values in this analysis which are in need of refinement. The first is an assumed constant coefficient of friction. Notice that in formula (1) the surface temperature is directly proportional to this factor. Many charts have been published showing the effect of sliding velocity on the coefficient of friction. Practically all of them are different and are apparently dependent upon the test apparatus on which the data was obtained. Our data, taken on the coefficient of friction on actual gear teeth and on rollers, show the value to be approximately constant. This work is certainly not conclusive and, because of dynamic problems

---

[2] See General Discussion on Lubrication and Lubricants, Vol. 2, 1937, pp. 14-20, Group III: Industrial Applications — "Measurement of Temperature Flashes on Gear Teeth Under Extreme Pressure Conditions," by H. Blok. Published by Institution of Mechanical Engineers, London.

[3] See *SAE Journal* (Transactions), Vol. 44, May, 1939, pp. 193-200: "Seizure-Delay Method for Determining the Seizure Protection of EP Lubricants," by H. Blok.

[4] See *British Journal of Applied Physics*, Supplement on Physics of Lubrication, No. 1, January, 1951, p. 35: "Scuffing Temperatures of Boundary Lubricant Films," by T. B. Lane.

[5] See *Automotive Industries*, Vol. 73, Nov. 16, 1935, pp. 662-668 and Nov. 23, 1935, pp. 696-701: "Factors Influencing the Durability of Spiral Bevel Gears for Automobiles," by J. O. Almen.

of the test equipment above 360 fpm pitch line velocity, friction at speeds that normally occur in gearing could not be investigated. More work on this very difficult problem is needed.

The second possible source of error is the lack of dependable values on actual stiffness of the teeth. These values are very important when calculating the load distribution in the range of double tooth contact. Several investigators have published work on this subject, including S. Timoshenko and R. V. Band[6] in 1926, H. Walker[7] in 1938, and C. Weber and K. Banaschek[8] in 1949. Each gives somewhat different results, and an analysis based upon them at the present time is questionable. The temperature calculations for this paper were made by using stiffness data from our tests, which again are not in agreement with the above mentioned investigations.

A third possible source of error exists in calculations of the width of the contact band, particularly near the tip of the tooth. Formula (1) shows that the temperature is inversely proportional to the square root of this band width. A wider band means a lower temperature flash. The calculations are based on the well-known Hertz formula for this value, which was developed on rollers. Gear teeth are not rollers, and considerable error may exist near the tip of the tooth due to the lack of supporting material beyond the tip.

Test Results

These possible discrepancies have been discussed for the very pertinent reason that mathematical analysis does not agree exactly with test results. According to the lower chart in Fig. 3 the temperature appears to be higher at the tip of the tooth than at the end of single tooth contact. The calculated difference between these peaks is rarely more than 30 F. A very important observation of our tests on 20-deg pressure angle gears is that the scoring failure is sensitive to the temperature at the highest point of single tooth contact if no involute modification such as tip relief is used.

Scoring covers an area of the tooth face such as that shown on the tooth in Fig. 4. As is explained in Appendix I, our scoring tests are run normally with the gears in half-mesh. This permits a total of four runs per set of gears. Notice that a fairly definite line limits the bottom of the scored area. In more than 90% of the gear scoring tests this line was found to correspond to the highest point of single tooth contact, or just short of the point where two pairs of teeth commence to share the load. Notice also in Fig. 4 that on one half of the face width of the gear the outside diameter has been cut down. It was found that removal of the portion of the tip of this tooth had no effect on the scoring resistance. Scoring started at the highest point of single tooth contact and was apparently dependent on conditions in that area.

To further confirm this observation the highest point of single tooth contact was moved by grinding the outside diameter of the mating gear down on half the face width. Fig. 5 shows the results. The right half of the tooth shows scoring started at the highest point of single tooth contact in the normal location. The left half of the tooth face, when tested, had an end of single tooth contact raised on the profile due to a reduced OD of the mating gear. The difference in the starting point of scoring and the calculated results depended upon the conditions at the new location. Scoring resistance in this case was reduced, due to the increase in instantaneous surface temperature. This will be more carefully analyzed later.

Assuming that this highest point of single tooth contact was normally the critical point of scoring for our 20-deg pressure angle test gears, values for total surface temperature were calculated in accordance with formula (2) for this point on all of our tests. Good results were found when considering only one design and batch of test gears, but changes in design and small differences in the machining of a new batch of gears – even with the same cutting tools – seemed to produce a different level of critical temperatures for the same oil.

It was then recognized that wide variations of surface finish are normally obtained in the shaving operation, unless this quality is carefully controlled. The surface finish values of the test gears ranged from approximately 17 rms microin. to

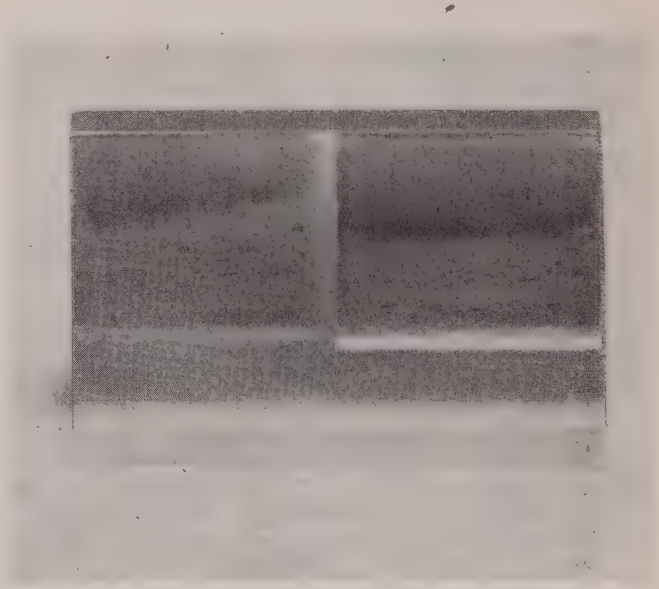


Fig. 5 – Scored gear tooth face – highest point of single tooth contact moved by grinding down outside diameter of mating gear on half of face width

[6] See *Mechanical Engineering*, Vol. 48, December, 1926, pp. 1105-1109: "The Strength of Gear Teeth," by S. Timoshenko and R. V. Band.

[7] See *The Engineer*, Vol. 166, Oct. 14, 1938, pp. 409-412, Oct. 21, 1938, pp. 434-436, and Vol. 170, Aug. 16, 1940 pp. 102-104: "Gear Tooth Deflection and Profile Modification," by H. Walker.

[8] See "The Deformation of Loaded Gears and the Effect on Their Load-Carrying Capacity," by C. Weber and K. Banaschek. Sponsored Research of Germany, 1949, Reports Nos. 3, 4, 5, and 6, Institut fur Maschinenelemente der Technische Hochschule, Brunswick, Germany.

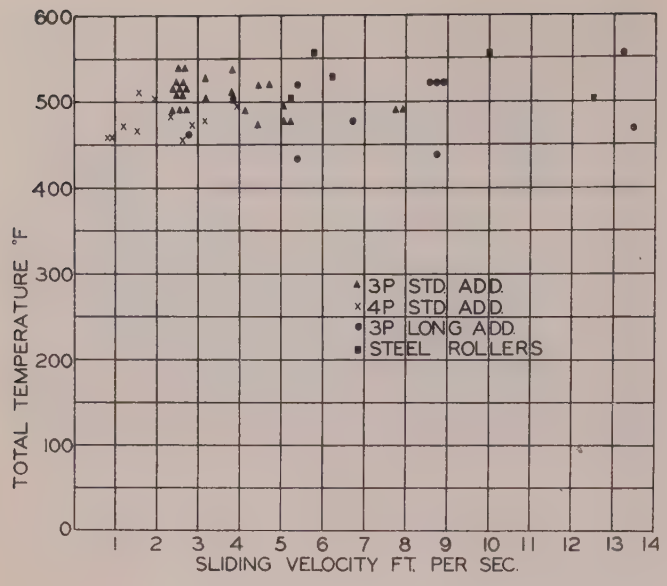


Fig. 6 – Scoring failures on Caterpillar test gears

about 27 rms microin. These values of surface finish were obtained after the gears were tested, in the area immediately below the start of scoring. The measurements therefore included the improvement of the surface due to running-in. Measurements of surface finish before the gears were run were found to be as high as 50 rms microin. This surface finish was reduced to the 27 rms microin. value within a couple of hours of running. The important criterion is the surface finish of the gear after it has been run-in rather than its original finish. No doubt, different machining practices, and use of different materials, resulting in producing or having the same initial surface, will lead to reduction to different levels of roughness after running the gears for only a short time.

The following formula was found to agree very well with our data:

$$T_t = \frac{T_1 + 0.00317 \dfrac{W_n (V_1 - V_2)}{(\sqrt{V_1} + \sqrt{V_2})\sqrt{b/_2}}}{1 - \dfrac{S}{55}} \qquad (3)$$

The constant 0.00317 includes thermal properties of hard steel mating with hard steel, and a constant coefficient of friction of 0.06.

S = Surface finish rms microin.

More detailed information on the construction of this formula may be found in Appendix II.

As is shown in formula (3), surface finish plays a very important part in the scoring resistance of gears. This formula has been proven to work satisfactorily for surface finishes between 7 rms microin. and 27 rms microin. Naturally, the formula becomes questionable as the value of S approaches 55. However, as was explained before, surface finish readings that are taken before the gears are run are worthless. On some of our tests where surfaces were purposely roughened by abrasive blasting a few gears were scored during the run-in. Others which had undergone the same treatment, if able to get by the light run-in load, would withstand loads up to the estimated scoring load without failure. A new gear would either score within a few cycles or smooth over sufficiently in the same length of time to support the load. The first hour of operation is a critical period for scoring resistance.

Our data as calculated per formula (3) appears in the chart of Fig. 6. The chart contains the results from gears of various geometry and pitches, having surface roughness measurements varied as previously noted. It also includes ground rollers. Bulk oil temperatures were controlled at a variety of levels from 125 F to 250 F. Test machine details, and so forth, may be found in Appendix I.

The last of the seven factors affecting scoring is the influence of surface treatment. Some such treatments have shown the ability to permanently increase the scoring resistance of gears. Most surface treatments form a type of surface that assists break-in such as that produced by a manganese phosphate treatment, with an etch on the basic material and a granular coating over this. Measurements of the surfaces after they have been run have indicated a decided improvement in the roughness, much more than is generally gained through the normal running-in procedure used on our scoring tests. Soft metallic coatings, such as those produced by copper or tin plating treatments, no doubt tend to flow under operating temperatures and pressures, thus probably tending to fill the lower surfaces of the roughened area, and making them support a larger than normal portion of the load, thus obtaining a better distribution of pressure over the contact pattern (see Appendix II).

A note of caution should be mentioned about temperatures. The gear blank temperature is not always the same as the bulk oil temperature. Differences close to 30 F are not uncommon, particularly where an externally cooled jet of oil is the only source of lubrication. In such cases the temperature of the gear blank may run significantly higher than the temperature of the oil jet. Our measurements of gear blank temperatures of jet lubricated gears were taken from a thermocouple suspended in the oil stream being thrown off the gear. It was felt that this value was much closer to the true gear blank temperature. When splash lubrication is being used with no external oil cooler, the blank temperature is no doubt very close to the bulk oil temperature.

It is apparent from Fig. 6 that the critical temperature of the SAE 30 grade straight mineral oil used is approximately 500 F. This temperature value cannot be assumed as a real surface temperature. It is still merely a calculated value based on our best effort to calculate the actual. It is ex-

pressed as a temperature for the reason that the background of its formulation was a theoretical consideration of temperature and the answer cannot be justly expressed in any other term.

Analysis of Scoring

Many scoring problems have been brought to the writer's attention that indicate the critical scoring point is not always the highest point of single tooth contact. The effect of profile modification, pressure angle and other factors of gear geometry can readily change the scoring picture. It is necessary that more than a superficial attempt be made to understand these variations, for the danger of rejecting a new method of approach is very great if a single personal experience does not seem to agree with it.

A good example of scoring starting higher than the highest point of single tooth contact is found with high-pressure angle gears.

The higher the pressure angle the greater the stiffness of the teeth. This increase in stiffness produces a temperature at the tip of the tooth that is much higher than the temperature at the highest point of single tooth contact. When this difference becomes sufficiently great, scoring will then start at some point during double tooth contact. If profiles are unmodified and scoring does not extend down to the highest point of single tooth contact the probability of running into serious continual removal of material is very small. Fig. 7 shows what normally happens in this case. If the tip of the tooth has a surface temperature that is above the critical temperature of the oil, scoring extends down the profile only to the point where such temperatures are not obtained. This amount of scoring may pass fairly rigorous inspection without being recognized. The abnormal wear that occurs at the immediate tip of the tooth during this process tends to redistribute the load and a change is made in the temperature pattern, such as is shown in the shaded area. Once this maximum temperature goes below the critical point the scoring will cease and the scored area will tend to heal over. We have observed this healing effect many times in the laboratory, and have seen very clear evidence of it on field test tractors. Small amounts of abrasive, such as dirt, in the transmission seem to make this healing more rapid. It is generally better to design a gear that will not score in the first place than to design one that scores and then repairs itself, and tip scoring can be eliminated by relieving the profiles slightly. The diagram in Fig. 8 shows results that can be obtained with tip relief. To produce exaggerated temperature differences, the abscissa of the diagram has been represented by the line of action of a short addendum gear and a long addendum pinion. By relieving the tip of the pinion as is shown, improvement is obtained on the scoring resistance of the gear set until one of two things occur at higher loads and/or speeds. The first and most probable one is that, with the addition of the correct amount of relief, scoring will extend down to the highest point of single tooth contact. The second form of the failure that will limit further improvements is the occurrence of scoring on the dedendum of the pinion and the addendum of the gear, due to the increased temperature in these areas. In either case no more improvement is to be found in profile modification.

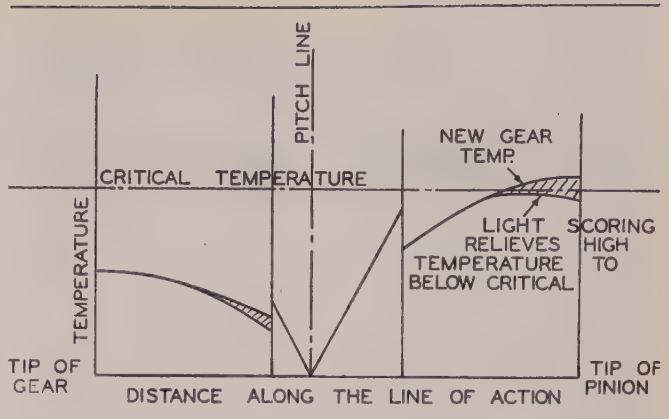


Fig. 7 – Effect of light tip scoring on surface temperature

Modification in the form of tip relief is rather common practice in the gearing industry today. Sometimes scoring troubles are a result of this modification. Suppose that the tip of the mating gear is relieved in the case of the long addendum

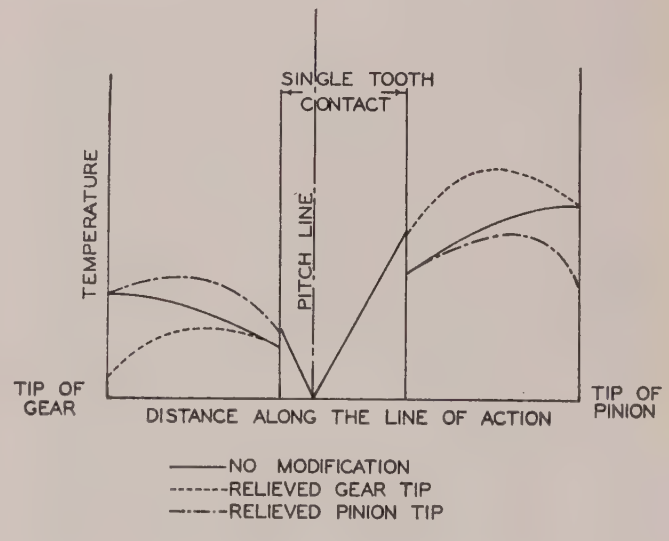


Fig. 8 – Effect of tip relief on surface temperature

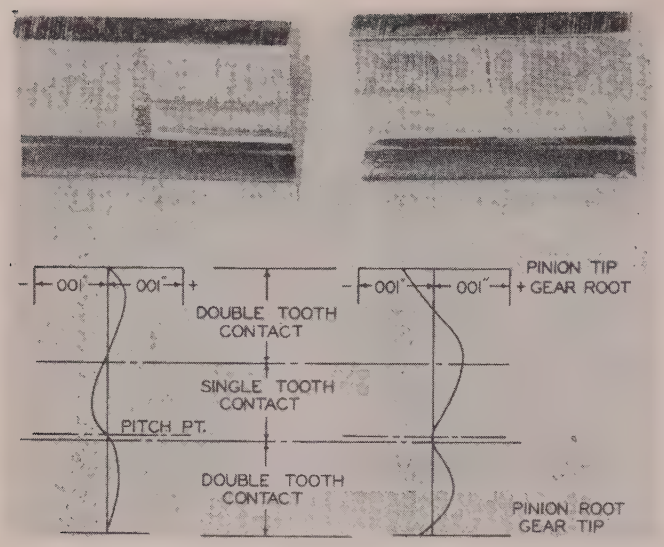


Fig. 9 – Result of combined profile errors on scoring pattern of gears

pinion. Notice, in Fig. 8, that this throws more load on the addendum of the pinion, thus raising the surface temperature in that region, possibly above the critical temperature of the oil. Therefore, tip relief can either help or aggravate the scoring resistance of gears. Improvement caused by tip relief is limited to a certain extent, however; that amount may be calculated by assuming the peak temperature is at the highest point of single tooth contact in normal gearing, as determined in the development of formula (3).

Some indication of the sensitivity of scoring to tip modification is shown in Fig. 9 of two test gears. The one at the left has no significant modification at the tip of the tooth. The other has a combined tip-of-pinion, root-of-gear error amounting to 0.0005 tip relief. Notice that the resultant scoring pattern on the tooth faces bear out the diagrammatic explanation in Fig. 8. Scoring does not continue all of the way to the tip of the modified profile tooth.

Results from Other Published Data

A large amount of data has been published as the result of tests made with the 3¼-in. center distance machine used by the Institution of Automobile Engineers in England. Most of the data have been contributed by H. Mansion of the Motor Industry Research Association.[9, 10] This machine, which is now widely used as a lubricant tester, is a power circulating gear test machine similar in principle to our test machine. The test gears on the machine normally have a face width of 0.2 in. from the tip to the pitch point. From the pitch point the tooth width increases in a radius to 0.5 in. at the root. This is done particularly on scoring and pitting tests in order to obtain maximum beam strength. The changing face width, however, greatly changes the stiffness pattern of the gear teeth, thus not faithfully reproducing the load distribution conditions of conventional gears.

Published gear data which has been obtained on this machine has not been successfully correlated with our own scoring results. Probably the different stiffness factors produced by these gears is the primary reason.

A proper closure to this paper is a word of optimism and a word of caution. Blok's postulate, that a straight mineral oil will fail at a critical temperature dependent only upon its viscosity grade, has been confirmed on machines which are widely divergent in design, on balls, on rollers, and on gears. Considering this correlation, the postulate shows promise of becoming a reliable lubrication design tool. However, the proper use of such a tool is dependent upon our knowledge of the many factors which enter into it and our ability to properly apply them to our problems. Purely empirical formulas, although necessary as a temporary design tool, are rarely expandable and cannot be trusted for advanced designs. The approach which attempts to deal with more fundamental concepts is more desirable because of its probable expandability. Even with only fair initial applicability it will form a base which is generally added to instead of discarded, with the advent of more stringent design requirements.

APPENDIX I

Test Machines and Procedures – The gear test machine used in obtaining this data was one of the single pair gear test machines described fully in the SAE paper by VanZandt and Kelley.[11] They are power circulating test machines using test gears of a size and shape comparable to tractor transmission gears. Oil temperature, loads, and speeds are controlled on all tests.

All test gears were spur gears, carburized and hardened to approximately Rockwell C 60 after finish shaving. We have had very little experience with scoring tests on helical gears, largely because these helical gear tests have been conducted on pitches fine enough to eliminate scoring problems. An analytical solution of load distribution of helical gears, which would allow calculations of loads for any part of the tooth face, has not been attempted. It may be that by proper adjustment, formula (3) can adequately fill the need for determining the scoring resistance of helical gears.

Gears were meshed over only one-half of the face width as was described in the text of this paper. There was some concern about the effect of the stiffness of teeth loaded in such a manner.

[9] See "The Effect of Chemical Surface Treatments on the Scuffing of Gears," by H. D. Mansion. Institution of Automobile Engineers Automobile Research Committee Report No. 1943/15, December, 1934.

[10] See "Some Factors Affecting Gear Scuffing," by H. D. Mansion, Part I, Gear Lubrication Symposium, The Institute of Petroleum, London, February, 1952.

[11] See SAE *Quarterly* Transactions, Vol. 3, April, 1949, pp. 354-368: "Gear Testing Methods for the Development of Heavy-Duty Gearing," by R. P. Van Zandt and B. W. Kelley.

Tests, however, showed the same results when the teeth were split in the middle of the face width, thus actually making two gears out of one. We have also had excellent correlation with designs of gears greatly different from our test gears. Gears that have larger face widths supposedly in contact uniformly across the tooth, produce a larger scatter of results than is seen in Fig. 6. No doubt some means of analyzing this maldistribution of load is an important part of determining the scoring resistance of a pair of gears.

Tests were run by progressive load increase increments. Normally a 1-hr run-in was given the gears at a load sufficiently light to avoid scoring. Each half hour the load was increased until scoring was detected by visual inspection. After it was learned that the surface finish was a critical factor in scoring, the procedure was changed slightly. Low-speed runs requiring a great many load steps were long enough to make the gear on test smoother before scoring, than a gear tested at higher speeds but not requiring as many steps. Once an approximate trend of scoring failure appeared the run-in was made at the same load for all speeds, but certain load steps were omitted so that approximately the same amount of running time was required before scoring took place. This time was normally held to 3 hr plus or minus a half hour. This procedure resulted in approximately the same surface roughness value for all runs on the same batch of gears. It was also found that when the highest point of single tooth contact was raised to produce higher temperatures at this point, failure occasionally took place during the run-in. In these cases scoring occurred almost immediately – probably within a few cycles – due to the roughness of the new gears and the immediate high temperature produced by sliding.

The gear roller test machine used for roller test points plotted in Fig. 6 is shown in Fig. 10. Two rollers of approximately 3 in. in diameter and ½ in. thick are pressed against each other by means of a spring-loaded machine designed on the nutcracker principle. As is described in the paper on our test equipment,[11] the rollers are driven either by eccentric phasing gears mounted on the same shafts, which cause different proportions of sliding and rolling to occur at different points on the peripheries of the rollers, or by phasing gears which produce a constant proportion of rolling and sliding on the peripheries. Scoring results were obtained by both methods. Originally the bottom roller dipped in an oil bath, which was considered to provide sufficient lubrication. Large quantities of oil vapor were produced between the rollers indicating the very high temperatures at the surface, and secondarily, producing an explosion hazard which was necessary to eliminate. An oil jet impinging on the inmesh side of the rollers had no effect; however, a heavy jet on the outgoing side apparently entrapped the mist or barred its formation and no more vapor was seen.

Because of the inconvenience of increasing the loads in steps, the rollers were scored by adjusting the load to approximately the scoring point, and then increasing the speed in steps by means of an infinitely variable belt-drive speed reducer. Scoring was determined on the rollers in several ways; the first and the most reliable was visual inspection. The second was through a magnetic pickup, mounted close to the roller surface, which indicated surface roughness. The third was through a bearing block deflection measuring device employed to determine the coefficient of friction.

APPENDIX II

Limits of Formula – The original formula by Blok contains limiting assumptions since thermal formulas of this nature will tend to change with very large changes in velocity. As is pointed out in the 1937 publication,[1] the limiting criterion for the background of development on the formula is that the values of V_1 and V_2 be greater than $5 \cdot 4a/b/2$ where a is the diffusivity of the material and b is the width of the contact band. This is not particularly important in this paper because the range of gearing speeds in transmissions and gear reducers is normally much greater than this. The error of peak temperature is also quite small for relatively large deviations from this limitation.

Several methods could have been used in handling the surface finish factor. Since it was primarily a straight line function, a formula such as

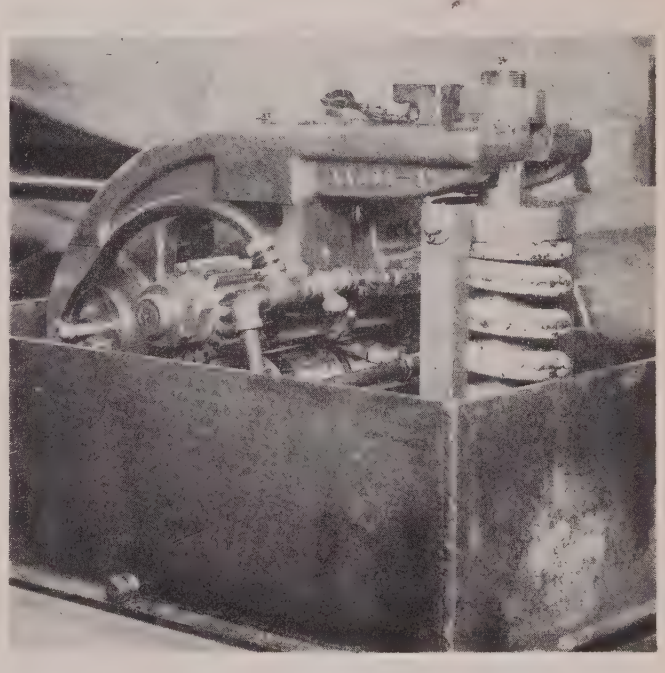


Fig. 10 – Gear roller test machine

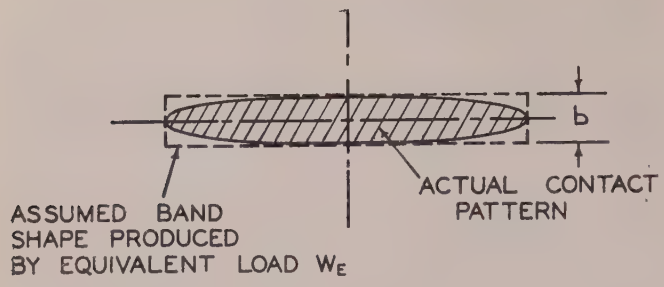


Fig. 11 – Application of scoring formula to elliptical contact pattern

(4) would fit satisfactorily for the basic test lubricant.

$$T_t = T_1 + T_2 + (\text{constant})\ (S) \tag{4}$$

The writer's first thoughts were that the factor should pertain only to the calculations of instantaneous surface temperature T_2. With the surface finish factor inserted in formula (3), as in (4), apparently an instantaneous surface temperature increase occurs even though the surface sliding velocity of $V_1 - V_2 = 0$. There may be some truth to this but it has not been experimentally verified. Tests on gears at varied temperatures of the blank, however, showed that in no way could the factor S be applied to the value of instantaneous surface temperature alone.

It was also felt that the value of S should be used as a multiplying factor indicating an increased percentage of temperature due to roughness. This could be qualitatively justified if surface roughness is considered as disturbing or distorting the shape of the Hertzian distribution of pressure over the contact area. Thus if a perfectly smooth surface is designated as unity[1]; and, according to our curve fitting, a surface having a constant roughness of 55 rms microin. has no scoring resistance, or a surface capacity equal to 0, then the percentage of capacity for a perfectly smooth surface would be an amount equal to:

$$\left(1 - \frac{S}{55}\right) 100 \tag{5}$$

This value then is used as a percentage of ideal critical temperature T_c in a formula which would set the design limit.

$$T_c\left(1 - \frac{S}{55}\right) = T_1 + T_2 \tag{6}$$

$$T_c = \frac{T_1 + T_2}{\left(1 - \frac{S}{55}\right)} \tag{7}$$

It is realized that the variation of surface finish may have a variety of effects. Besides the distortion of the pressure distribution over the area of contact the surface roughness may cause a change in the coefficient of friction. Other equally good conjectures may be found which indicate the need for more research in this field.

APPENDIX III

Proportionalities of Scoring Formula – As has been shown by Professor Blok in his 1937 publication,[1] the instantaneous surface formula (1) can be reduced for a given pair of gears to:

$$T_2 = Cf\sqrt{S_c^3 V} \tag{8}$$

S_c = Compressive stress at the critical point of scoring.

C_1 = Constant dependent upon other constants in the original formulas plus a constant of the gear geometry.

V = Sliding velocity $V_1 - V_2$.

This makes the investigation of a gear set for temperature determination fairly simple, for once the value of C_1 is determined for a single pair of gears, it does not change for those gears regardless of load or speed unless the gear geometry or material is changed. As long as the investigation is being applied to gears having a band-shaped contact pattern, formula (8) can also be stated as:

$$T_2 = C_2 f\, W n^{3/4}\, V^{1/2} \tag{9}$$

In this case scoring is more critical in the lower speeds for the same horsepower transmitted.

The method of approach in the text of this paper would not seem to apply to crown-shaved gears since the hypothesis assumes a band-shaped contact. Bevel gears with the different radii of curvature providing a certain amount of mismatch between the teeth, also fall in the category of having an elliptical contact pattern. The ratio between the minor and major axes of this contact pattern is, however, extremely small in most cases, rarely exceeding 0.03. The error would be insignificant if an equivalent load W_e is assumed to form an equivalent band-shaped contact as shown in Fig. 11. This equivalent load should have the same width of the band of contact as the maximum width of the minor axis of the elliptical contact produced by crown-shaved gears and bevel gears.

The proportionality found in formula (8) is still true for the elliptical pattern of contact; however, formula (9) is not applicable since, in this case, the maximum compressive stress Sc is proportional to the cube root of the total normal load P, where:

$$T_2 = C_3 f\sqrt{W_n V} \tag{10}$$

Therefore, for the case of crown-shaved gears and bevel gears of normal design, where the length of the contact pattern is no greater than the gear tooth face width, the instantaneous temperature increase at the surface will be proportional to horsepower transmitted by the gears. It also follows

that as long as the gear blank temperature can be kept the same at all horsepowers, then scoring will also be proportional to hp. This proportionality has been substantially verified on our bevel gear tests in the laboratory. It is reportedly the experience of the automotive field that bevel gear scoring occurs generally at the higher speeds. This is not necessarily in conflict with the scoring proportionality (10), for at the higher speeds, particularly during periods of high speed acceleration, the bevel gears are generally transmitting more horsepower.

DISCUSSION

Method of Lubrication Important Scoring Factor

– Wells Coleman
Gleason Works

THIS interesting paper on an enigmatic subject relating to gear design should stir up interest among gear engineers to promote further study of gear-tooth scoring.

Two terms, scratching and burning, have been mentioned as very mild cases of failure. Regarding burning, it was stated that it is recognized by a discoloration of the tooth surface. Since some lubricants have the effect of staining the tooth surface, this discoloration may not always indicate burning.

To the list of factors which may be considered important in an analysis of scoring, I would suggest the addition of the method of lubrication – sump, jet, spray, or mist. As mentioned in this paper, the temperature at the point of contact will be somewhat higher than the temperature of the oil bath, and the difference in temperature between the oil bath and the peak temperature on the tooth surface will vary according to the method of lubrication. I believe that the real criterion of scoring is the actual peak temperature at the point of contact, and the items which have been listed as factors are used as a gage for determining this peak temperature.

One point, I believe, needs further clarification. In equation (2) the symbol T_1 is defined as the bulk temperature. Does this refer to the temperature of the oil bath or to the temperature of the gear blank?

Heating Influences on Meshing Tooth Faces

– H. Blok
Technical University, Delft, Holland

IN the commentator's opinion the present paper constitutes a most valuable contribution to the field concerned. Therefore, the comments and suggestions to follow are only presented in the hope that they may eventually result in an even deeper insight into the scoring phenomena of gears.[a, b]

Roughness factor $(1 - S/55)$ in formula (3) being empirical, it introduces an element foreign to the otherwise fundamental nature of the formula. It is suggested that there are two effects of roughness that influence the heating phenomena on meshing tooth faces. Further, it is thought that, by accounting for these two effects, formula (3) could be rewritten in a more fundamental and more universally applicable form. The two effects concerned are:

1. Roughness will disturb Hertz's semielliptic distribution that would be obtained if the tooth faces were perfectly smooth. Accordingly, the distribution of the intensity of the frictional heat generated between the meshing tooth faces will show wrinkles superimposed on Hertz's distribution. When such wrinkles increase maximum heating intensity beyond that occurring with perfectly smooth tooth faces, the maximum flash temperatures could become higher than those calculated according to fundamental formula (1).

2. Greater roughness may well bring about an increase in the coefficient of tooth friction. This will then result in a proportionally increased intensity of frictional heat in the contact area and, therefore, in an increase of maximum temperature.

The second effect is thought to be more promising for the present purpose than the first. This can be argued as follows:

Formula (1) holds good for any distribution of frictional heat, provided that the numerical constant K is properly adjusted. For instance, for a parabolic and for Hertz's semielliptic distribution, K amounts to 0.83 and 0.80, respectively. It should here be noted that, for any given load (or, say, width of contact area), intensity of heating, and speeds V_1 and V_2, K is a proportional measure of maximum flash temperature, T_2. Now, the ratios between the maximum and the mean intensity of frictional heat, for the two distributions considered, amount to 1.50 and $4/\pi = 1.27$, respectively. But, whereas 1.27 is about 15% lower than 1.50, 0.80 is only about 4% lower than 0.83. This may go to show that maximum flash temperature, T_2, is primarily determined by the mean intensity of frictional heat; the distribution of the intensity of frictional heat and, therefore, the maximum intensity of heating with any given mean intensity, is only of secondary importance.

Accordingly, it is thought that the variations of the empirical factor $(1 - S/55)$ within its range of applicability ($S = 7$ to 27 microin. rms) are much greater than can be explained on the basis of any acceptable degree of disturbance of Hertz's semielliptical distribution of heating in-

[a] See "A Practical Application of the Flash-Temperature Hypothesis to Gear Lubrication," by T. B. Lane and J. R. Hughes. Proceedings of Third World Petroleum Congress, The Hague, 1952, Section V88, published by E. J. Brill, Leyden, Holland.

[b] See "From Test Machine to Gear Box," by J. R. Hughes. Preprint, Part 2, pp. 8-14, Gear Lubrication Symposium, Institute of Petroleum, London, 1952.

tensity. A more definite conclusion could be reached only after roughness profiles have been determined for the tooth faces concerned. Then the present data, referring only to the rms – height of the irregularities – could be supplemented by data about the ratio between the spacing of the irregularities and the widths of the contact areas at the different tooth loads; thus more could be said about the degree of disturbance concerned, including the variation of the numerical constant K with the different surface finishes.

For the time being, the second effect, that of roughness on coefficient of friction, and thus on the mean intensity of frictional heat, would appear to be far more promising in regard to explaining the variations suggested by the roughness factor $(1 - S/55)$.

A systematic investigation of the variations of coefficients of tooth friction as a function of surface finish and other operating conditions (such as viscosity of the oil, load, and speed) is indicated also to clarify other questions that are still open. One such question is: to what extent, and in which range of operating conditions does the hydrodynamic formation of pressure in the oil lubricating the meshing tooth faces, play a part in carrying the tooth load?[c]

Finally, in order to render possible the verification of the author's views, as against those propounded previously,[d] or possibly to be propounded in the future, it is suggested that in his closure the author submit a detailed table of his experimental results and operating conditions.

[c] See Fig. 2 in "Gear Lubricant – A Constructional Gear Material," by H. Blok, in *De Ingenieur* (Holland), Vol. 63, Sept. 28, 1951, pp. 53-64. Article in English.

[d] See ASME Transactions, Vol. 73, July, 1951, pp. 687-696: "Effect of Oil Viscosity on Power-Transmitting Capacity of Spur Gears," by V. N. Borsoff, J. B. Accinelli, and A. G. Cattaneo.

Two Formulas Are Compared in Tests

– Darle W. Dudley
General Electric Co.

MR. KELLEY gives us a sound engineering approach to the scoring problem. The work of Almen and Straub[e] solved the scoring problem for a limited range of work. Their PVT formula was empirical in nature and, as might be expected with an empirical formula, worked well in certain ranges of work and failed badly in other ranges. Mr. Kelley's work is based on rational derivations made by Blok. From evidence obtained to date, it appears that his formula will have a wide field of use in designing all types of gears.

In December, 1951, I first obtained a copy of Mr. Kelley's notes on his work. Since then we have had the opportunity to run a large number of scoring tests at the Lynn General Electric plant. Tests have been made on case-hardened gears with ground teeth, and on 350 Bhn gears which were shaved. We have tested spur and helical gears. We have tested steel gears and titanium alloy gears. We have used hydrocarbon oils and diesters. Results have been analyzed by several different methods, including both the PVT method and Mr. Kelley's method. The data shows logical correlation only when the Kelley method is used.

Since much of our work in Lynn is on classified projects, I cannot give you any detailed survey of our scoring test results. However, the same things we have noted in our work can be seen in data already published. A good example is the paper published by Borsoff, Accinelli, and

[e] See *Mechanical Engineering*, Vol. 73, July, 1951, pp. 565-569: "Shot Peening as Factor in Design of Gears," by J. C. Straub.

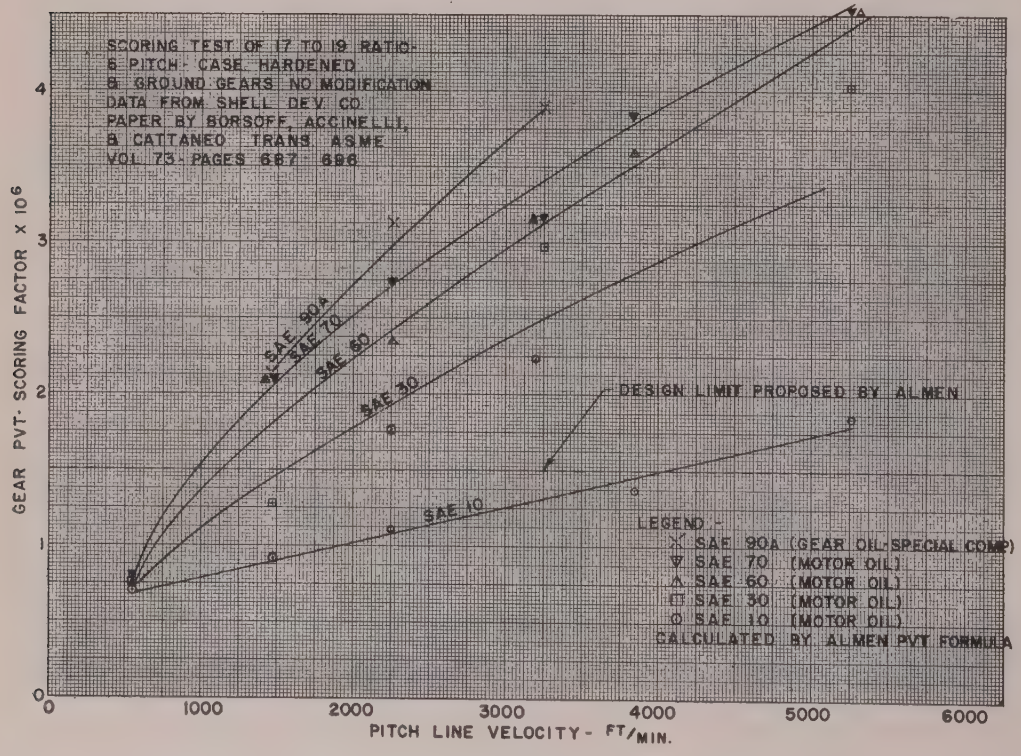


Fig. A – Gear PVT versus pitch line velocity

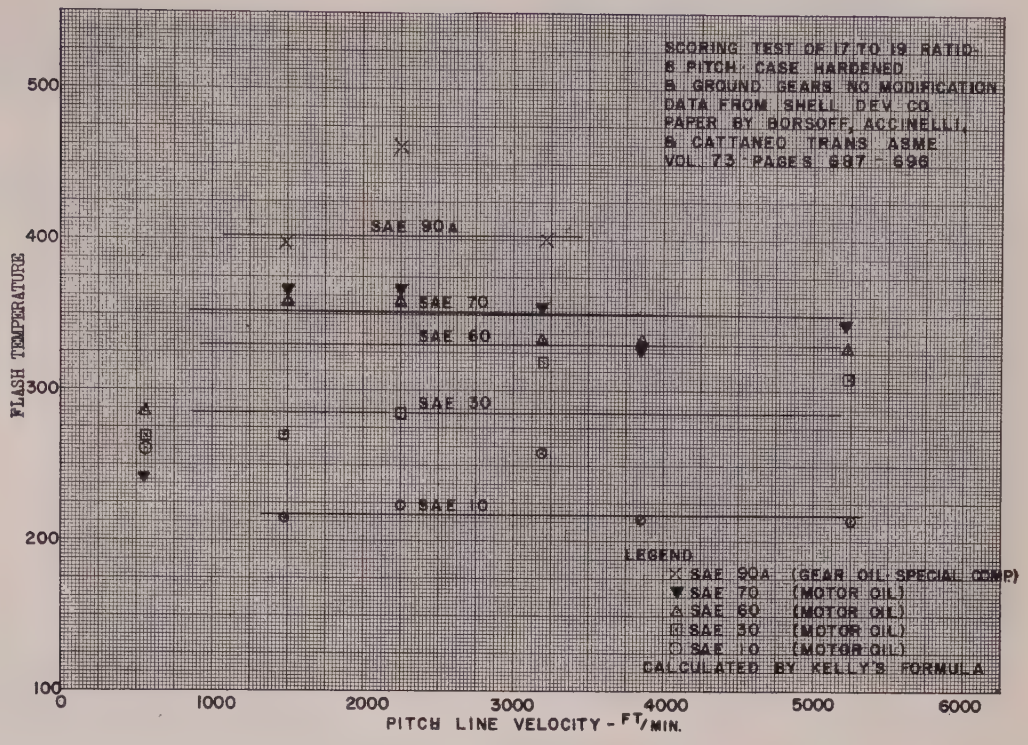


Fig. B – Flash temperature versus pitch line velocity – start single tooth contact

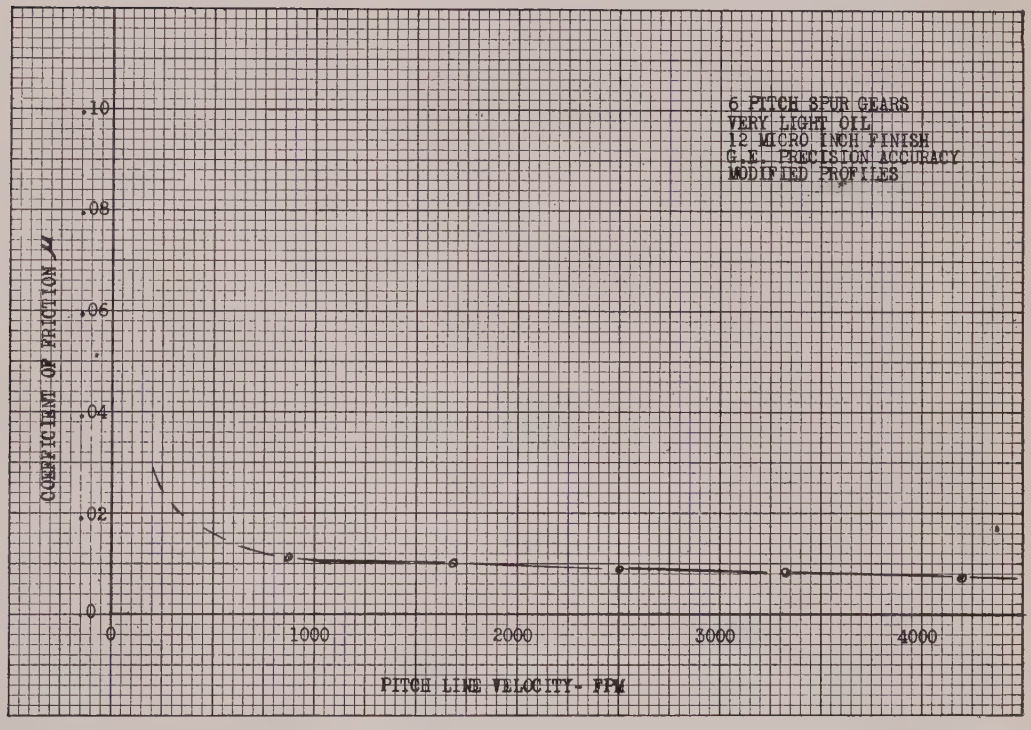


Fig. C – Friction on gear teeth test

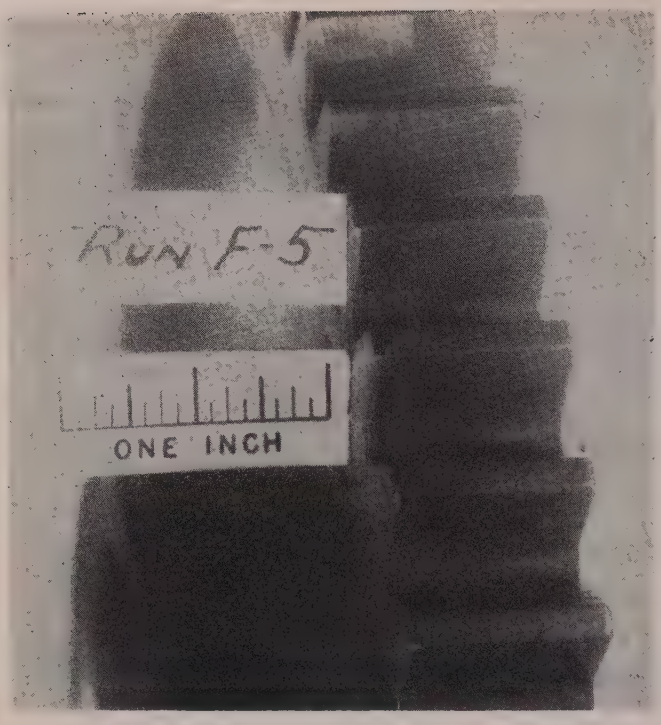


Fig. D – Scoring test of titanium gear

Cattaneo[f] of Shell Development Co. They made a very careful series of scoring tests using different oils and different running speeds. Although they did not have a chance to use the Kelley formula to analyze their data, their paper includes enough detailed information to permit an engineer to analyze it in any way he wishes.

I have worked up their data on both the Almen and Straub PVT formula, and on the Kelley formula. Fig. A shows the PVT approach. Note that the data does not cluster along the suggested design limit of 1.5 million PVT. Both speed and viscosity appear to be out of control. Fig. B shows the same data plotted by the Kelley method. Here the data is under good control. A proper series of flash temperatures would make good design limits.

Our Lynn test results have shown that with thin oils we could carry high loads without scoring. Others have reported that only very light loads could be carried without scoring. We have been using teeth with modified profiles, very close accuracy, and surface finish under 15 microin. rms. I believe we have an answer as to why our gears have carried so much more load without scoring. Fig. C shows a plot of coefficient of friction against pitch line velocity. Note that this curve levels off at 0.01 coefficient of friction. This value is one-sixth of the 0.06 value that Kelley has taken for an average gear. Apparently, with spacing accuracy down to 0.0002 tooth-to-tooth, and with fine finish and precise involute modification, the coefficient of friction can be drastically reduced. This means proportionately less heat available to score the gears.

Mr. Kelley has indicated in his paper that scoring is usually more apt to occur at the lowest point of single tooth loading on the pinion, than at the start of involute contact. Generally, it is impossible to tell where scoring started when you inspect scored gears. Last month, though, we had a case which seems to check the analysis of Mr. Kelley. Fig. D shows a scored titanium alloy gear. The pear-shaped markings clearly show that the scoring started just above the pitch line of the gear. This would mean that the scoring started just below the pitch line on the pinion – which happens to be Mr. Kelley's lowest point of single tooth loading. If scoring had started at the pinion root, the markings on the gear would have had to start at the gear tip.

I feel that Mr. Kelley has given us a very useful tool to work with. A lot of testing yet remains to be done, though. Proper flash temperature limits must be obtained for a wide variety of oils, exact coefficients of friction must be measured, and a variety of gear metals and surface treatments must be explored. The author has taken a big step forward, but we still have some more big steps to take before we can solve all cases where the gear scoring phenomenon occurs.

[f] See ASME Transactions, Vol. 73, July, 1951, pp. 687-696: "Effect of Oil Viscosity on Power-Transmitting Capacity of Spur Gears," by V. N. Borsoff, J. B. Accinelli, and A. G. Cattaneo.

[g] See *Journal of the Institute of Petroleum* (London), Vol. 38, August, 1952, pp. 624-632: "Criteria Governing Scuffing Failure," by F. T. Barwell and A. A. Milne, from Gear Lubrication Symposium, Part I.

Author's Closure To Discussion

I AGREE with Mr. Blok's concern over the empirical nature of the roughness factor $(1 - S/55)$ and appreciate his suggested course of further investigation. We are considering such a program now.

It is realized that the roughness factor should probably be based upon the topography of the surface finish. For the purpose of this paper, however, it is felt that the value of S in microin. may properly be assumed a function of the surface geometry if limited to shaved gear teeth. Such a term as S has the additional advantage of being easily measured.

It must also be realized that on shaved gears the machining marks are very close to being parallel or in line with the direction of motion. This is true also with the ground rollers. It is possible that machining marks across the tooth, such as those produced on most ground gears, will change the roughness factor considerably, or even that a rougher surface, up to a certain point, is beneficial. This effect is quite marked in the work of Barwell and Milne.[g]

Mr. Dudley has shown that the published test data of Borsoff, Accinelli, and Cattaneo in general, seems to agree with the critical temperature hypothesis. It will be noticed, however, that the level of failure is considerably lower than ours. Their tests have been run with a non-reactive mineral oil. Assuming that this non-reactive oil is a very highly refined white mineral oil it would contain a minimum of the polar compounds found in the usual straight mineral oil. This may be the reason for the difference in levels of temperature shown on Mr. Dudley's chart and ours for oil of the same viscosity.

As Mr. Coleman has suggested, the method of applying the lubricant is important, and possibly should be made a part of the seven factors governing scoring. It has a direct relationship to Mr. Coleman's question as to whether T_1 is the bulk oil temperature or the gear blank temperature. The proper answer is that T_1 is the temperature of the gear blank. As was mentioned in the paper, under some conditions the bulk oil and blank temperatures are nearly equal. However, cooling jets, sprays and other forms of lubrication will invalidate this assumption and either the blank temperature must be measured directly for T_1, or another temperature difference factor introduced in the formula for these special cases. It would seem that an investigation would be warranted along these lines for gear sets which are in the critical range of scoring.

Rolling Wheels Gather No Skids

A. C. Gunsaulus, Goodyear Aircraft Corp.

This paper was presented at SAE National Transportation Meeting, Pittsburgh, Oct. 23, 1952.

ON slippery roads, currently available methods of brake control are frequently inadequate and dangerous. A locked wheel, caused by overbraking on that wheel, can permit the vehicle to skid and go out of control. If each wheel needs a brake, then each brake needs its own control. Such separate control can prevent premature wheel stoppage and undoubtedly eliminate this driving hazard.

Before making any more such sweeping statements, it might be advisable to take you through some of the thought processes that preceded these conclusions. Since all wheels on a vehicle do not contact the same spot of the road surface at the same time, and since each brake and each brake drum do not always react in exactly the same manner, we do frequently have inherently different conditions on each wheel. How, but with separate control on each wheel and brake, can these individual differences be adjusted?

The brakes on an airplane are individually applied. This control of each wheel or wheel pair is primarily used for the ground maneuvering of the airplane at relatively slow speeds, such as taxiing to the ramp after landing. At high landing speeds this type of brake application can sometimes result in a locked-wheel forward skid that may not be immediately discernible to the pilot.

As a result of studies, which were made to devise a better control for airplane brakes, a control system was designed that would prevent these locked-wheel skids. It seemed likely that this system would do the same for ground vehicles.

A CONTROL system is described that has been successfully preventing premature rotation stoppage of airplane wheels when too much braking is used. The system has now been applied to a truck in a series of tests conducted on an icy roadway.

It is shown that wheels exert their greatest braking effect when the brakes are applied almost to the point where the wheels lock.

Thus, when used on airplanes, the device (1) detects when a brake is about to lock the wheel, (2) releases the brake pressure to allow the wheel to pick up speed, and (3) again permits brake application. This cycle is then repeated until a stop is attained.

Similarly, when used on ground vehicles, the device functions by sensing the impending wheel lock, and then relays a signal to actuate the brake valve. The tests showed that individual wheel control devices can prevent out-of-control skidding in trucks and buses, and jackknifing of tractor-trailers.

The Author

A. C. GUNSAULUS (M '43) is development manager of the Airplane Wheel & Brake Division of Goodyear Aircraft Corp. He has worked for Goodyear for over thirty years, and for the past twelve years has had his present position. Mr. Gunsaulus received his B. S. and Professional Mechanical Engineer degrees from Iowa State College.

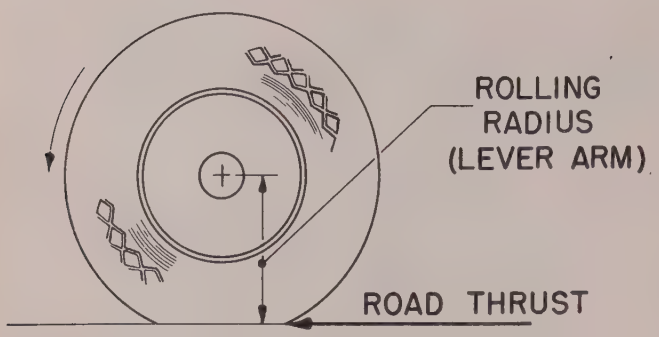


Fig. 1 – Wheel torque—road thrust times lever arm equals wheel torque

Before choosing the particular wheel pair that should be controlled, we turned to the moving pictures that were made by the National Safety Council in the winter of 1951. The Winter Driving Hazards Committee of this council was studying the skidding, jackknifing, and other kindred problems at a test facility located on the frozen surface of Pine Lake near Clintonville, Wis. The entire film taken of these tests is quite lengthy and is most interesting. The National Safety Council has permitted us to copy titles describing some of the significant scenes. The titles are self-explanatory and must substitute for the actual movie for readers of this paper.

The following titles appear before the scene in the movie in every case except the last one, which follows the last scene.

"Testing of tractor-trailers for jackknifing tendencies was performed on the shaved ice surface of Pine Lake, near Clintonville, Wis. Here is shown the straight test course.

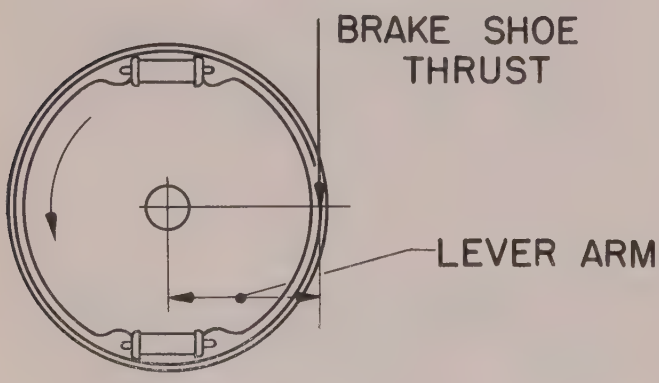


Fig. 2 – Brake torque—brake shoe thrust times lever arm equals brake torque

"With front wheels only locked, the vehicle slides straight, although the steering wheel was turned 20 deg as the brakes were applied. The locked front tires will not steer.

"With tractor rear wheels only locked, violent jackknifes occur. Here the rolling front tires guide the train into a curve, and centrifugal force pushes the sliding tires sideways.

"With the trailer wheels locked, these tires lose their steering ability and the trailer jackknifes.

"Controlled side forces can be produced by driving in a circle at constant speed. Here is a 400-ft diameter test circle on Pine Lake ice.

"Locking the front wheels only on the circle causes the vehicle to go on a tangent because the sliding front tires lose their steering ability needed to maintain the circular course.

"Locking the tractor rear wheels again produces a violent jackknife.

"The sliding trailer tires allow the trailer rear to swing out under the influence of centrifugal force.

"A rolling tire has steering ability which can control the vehicle's direction. A sliding tire loses this property, and the vehicle is then at the mercy of external forces for its behavior."

After reading these titles, it is easily understood why we concluded that one pair of wheels, particularly, produced violent reactions and brought on obvious out-of-control conditions when these wheels were prematurely stopped in their rotation by too much braking. Knowing that the control system we had used to prevent such an occurrence in an airplane wheel was successfully preventing wheel rotation stoppage, it seemed obvious that such a scheme, when applied to the driving wheels of a truck or tractor, would be beneficial in preventing skids.

As a test of this theory, it was decided to equip the rear wheels of a truck with such control devices, and drive it on an all-ice roadway. The purpose of this paper is to describe the control devices used, and to show the results obtained in these tests.

Just what happens when a moving vehicle is stopped with its brakes? (See Fig. 1.) We all know that the vehicle's tires are gripping the road surface, and it is quite evident that this force is dependent upon the tire's adhesion to the road, its contact with the road, and the load on the wheel. A road thrust, acting parallel to the road surface and tangent to the tire's circumference, has a lever arm substantially equal to the tire's rolling radius. If we multiply this force by the rolling radius of the tire, we will have *wheel torque*.

We likewise know that the brake shoes are gripping the brake drum with a force whose magnitude is dependent upon the adhesion of the brake shoe to the brake drum and the amount of pressure in the brake cylinder. (See Fig. 2.) This resultant force acts on a lever arm substantially equal to one-half the drum diameter. When we multiply

force by lever arm, we get *brake torque*. This torque is counteracting the wheel torque. It can readily be varied by the driver changing his foot pressure on the brake pedal. The wheel torque, however, can in no way be controlled by the driver.

The word *skid* may be frequently used so it had better be defined. In an automotive vehicle, skid is an unwanted sidewise movement not planned by the driver – at least that is the use I am going to make of the term skid in this paper. Its prime cause is a combination of a lessened grip of the tire on the road coupled with some sidewise force that is greater than the tire's grip. Its effect is usually a partial or, it may be, a total loss of control of the vehicle by the driver.

Safe operation of an automotive vehicle cannot be accomplished unless the driver always has complete control of direction and speed. Rolling tires have steering ability; sliding tires do not have this ability. The reason for this is the much greater road-gripping power that is inherent in the rolling tire but greatly reduced in the sliding tire.

Premature wheel stoppage means a sliding tire, a lessening of the tire's grip on the road, an excess of brake torque over wheel torque, or in other words, just plain overbraking. Can the driver anticipate and correct for this overbraking? In a few cases, the answer is yes. But in slippery going, things are happening too fast, and it is humanly impossible to make the necessary correction in time to prevent the skid.

With these basic elements of braking and skidding firmly fixed in our mind, let us ask ourselves if there is some sequence of events that always happens – a solid base that we might use to lay out a control principle. Here is such a list:

1. Brake pressure on, brake torque develops.
2. Brake torque exceeds wheel torque.
3. Wheel begins premature slowdown.
4. Wheel stops, tire slides, skid has begun.

When overbraking is present, this sequential order never varies. The laws of physics cannot be repealed. These events will always happen in this order under the conditions given. This orderly and unchangeable sequence furnished the clue to a control method. Why not utilize event 3, listed above; that is, premature wheel slowdown, as a warning of impending events with the hope that this warning signal could initiate a large enough counteracting force to arrest this premature wheel slowdown and correct the impending overbraking and resultant skid.

We have introduced control. The list of sequential events now follows:

1. Brake pressure on, brake torque develops.
2. Brake torque exceeds wheel torque.
3. Wheel begins premature slowdown.

So far, this list is just like the first one.

4. Sensing element signals slowdown.
5. Brake pressure relieved by signal.
6. Brake torque decays, wheel rpm recovering.

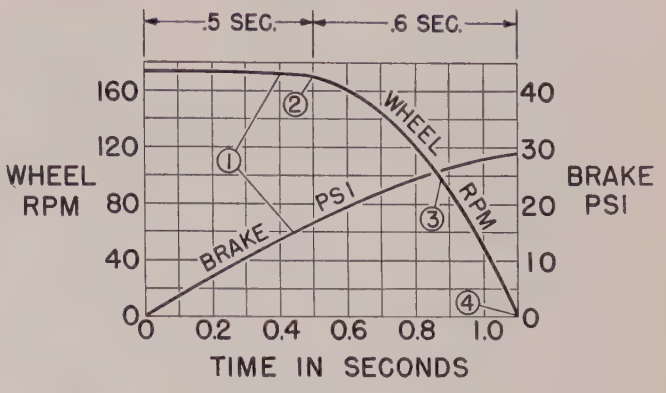


Fig. 3 – Braking sequences without individual control

7. Wheel rpm recovered, pressure reapplied.

In examining these sequential events in more detail, we will first look at the no-control system and try to establish the elapsed time for this cycle. (See Fig. 3.) This plotting of the beginning of an actual stop on a slippery road surface is keyed into our previously shown event schedule, and (1) shows the brake pressure buildup and normal wheel slowdown that occurs when brakes are first applied. Number (2) shows the point where brake torque exceeds wheel torque, and the wheel speed begins its precipitate decay. This event occurred in this example in 0.5 sec after pressure was first applied. Brake pressure continued to build up, and the wheel speed at (3) continues its rapid slowdown until at (4) the wheel has stopped rotating, sliding has begun, and skidding takes place. This last event transpired in 0.6 sec. We have gone from initial brake application to skid in 1.1 sec. Our control system does not begin to act until event (3) starts, at 0.5 sec, so we have only 0.6 sec in which to correct the overbraking. Nothing has happened to warn the driver of the impending wheel stoppage.

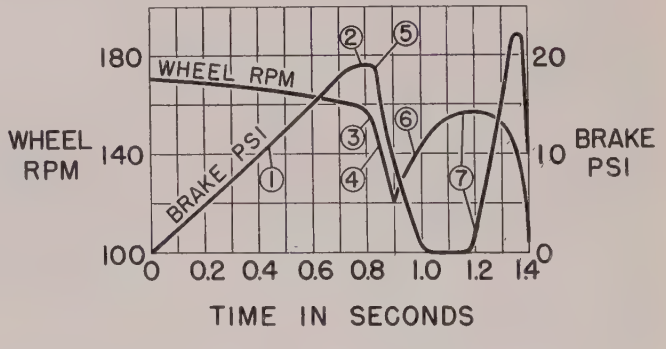


Fig. 4 – Braking sequences with individual control

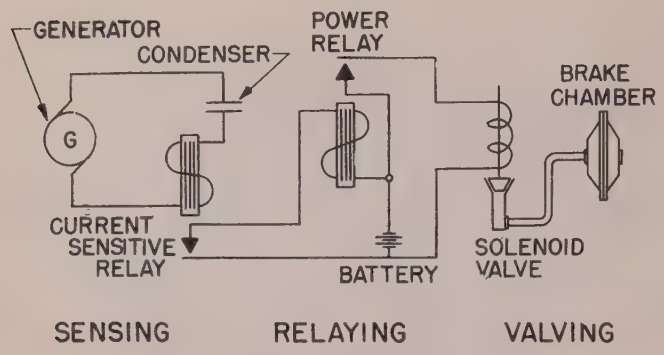


Fig. 5 – Basic control unit

Even if he had been warned, his normal reaction time could not have corrected his overbraking in 0.6 sec. In laboratory dynamometer tests, we have recorded times as short as 0.3 sec between start of slowdown and actual wheel stoppage.

This plotting of a first control cycle well illustrates in curve form the sequential events of a typical control period. (See Fig. 4.) The numbered points tie into the previous listing which were briefly as follows:

(1) Shows the brake pressure as well as brake torque build-up. (2) At this point the brake torque has exceeded wheel torque. (3) Here the wheel begins its faster-than-normal slowdown. (4) The abnormal slowdown has been signaled and at (5) the brake pressure has been relieved and brake torque begins its decline. At (6) the wheel rpm recovery is well under way. At (7) the cycle is completed, wheel recovery is complete, and brake pressure is again applied. Since the wheel never stopped rotating, it was never out of control. In this example, approximately 1.2 sec elapsed for a complete cycle, but again we have recorded in the laboratory such complete cycles that lasted only 0.5 sec. These split-second operations are naturally much faster than any known human reaction time.

Now that we have established a reasonable starting point for better wheel control, we must assemble electrical and mechanical elements capable of properly performing the functions required. These elements must be compatible, properly sensitive, and responsive as well as rugged. (See Fig. 5.)

A small d-c generator having a voltage output directly proportional to its armature speed was chosen for the element needed to quickly detect sudden changes in wheel rpm. A 4/1 step-up gearbox provided sufficient armature speed at low wheel speeds. Airplane wheels roll freely on a hollow axle, which fact influenced the choice of a small-diameter generator. One of these small-diameter generators was used in tests being described. This particular generator has 24.3-v variation per 1000 armature rpm.

Since the power output of the sensing generator is small, it was therefore inadequate to power the solenoid brake pressure valve. Two electrical relays, a condenser, and battery, were interposed between the generator and the brake valve to multiply and reinforce the weak sensing signal.

The very heart of the system, the sensing circuit, will be the first of the three circuits we will describe. The small d-c generator, the condenser, and the coil of the current-sensitive relay are connected together as shown. We must understand the operation of these three elements if we are to understand the overall control system.

To activate the control, current must flow through the coils of the current-sensitive relay. The generator armature is directly connected to the wheel it controls and always rotates in unison with that wheel. We also know that rotation of the generator produces voltage proportional to its speed. Just what are the actions and reactions that enable this circuit to react to a sudden speed change but be unaffected by a normal speedup or slowdown?

You will note that the generator and condenser occupy opposing positions in the circuit diagram. The condenser will not permit a pure, steady, direct current to flow through it – a current such as is put out during a normal wheel slowdown. It will, however, permit the voltage to charge its plates. A sudden change in generator voltage, such as occurs with a sudden change in speed, either a speedup or slowdown, will upset the condenser plate balance or voltage equilibrium. Condensers are inherently balancing mechanisms; and the condenser, in its effort to maintain voltage equilibrium on its plates, will discharge, and current will flow in the circuit. This current will pass through the coils of the relay, causing its switch to close and pass a signal to the relaying circuit.

The signal as received energizes the coil of the second relay and closes its switch. It thereby starts a current through the solenoid coils of the pressure valve. The second or relaying part of the system may seem superfluous to some of you, and you may be asking yourself, why didn't they use the sensing signals to directly energize the valve solenoid? The switch points of the current-sensitive relay we used could not carry as heavy a current as was needed to energize the solenoid. We introduced another relay that was big enough to carry this solenoid current.

In the third or valving circuit, the closing of the switch in the second relay permits power from the battery to flow through the coil of the solenoid valve, closing off the flow of pressure to the brake. Whatever pressure existed in the brake's chamber is released to the atmosphere.

The sensing circuit initially signaled a wheel slowdown. This signal eventually caused the brake valve to open and relieve the excessive brake pressure. Succeeding signals by the sensing circuit,

while the wheel's rpm was recovering, kept the circuits energized and the brake valve open until the wheel had again reached its proper speed. When the wheel speed stabilizes and the sensing signal ceases, the brake valve again closes admitting pressure to the brake chamber, and the cycle of control has been completed.

It might be interesting to look at some views of the actual truck installation. (See Fig. 6.) This view actually taken "on location" shows how the sensing unit was attached by means of an adaptor to the rear truck wheels. The side arm extending out from the truck body provided a suspension means for the wire lead coming from the generator as well as a spring-loaded link that restrained the rotation of the generator body when the wheel was driving its armature.

(See Fig. 7.) This view of the truck's rear axle shows how the solenoid valves were mounted directly on the brake diaphragm cases.

Just one further word about the installation on the truck and the truck's operation during the tests. Leads from the sensing circuit were carried to the cab where the relays and battery were located. From here the leads ran back to the solenoid valves just shown. A pair of switches in the cab permitted the operator to select control or no control at any time. As a safety precaution, a pressure-operated switch, located on the brake system's air supply tank and set to operate whenever the pressure dropped below 30 psi, would automatically throw out the control circuits when the air supply got too low.

During all of the test operation, the only brakes operating at any time were the ones on the rear of the truck. The front wheels and trailer wheels were blocked off from the brake system. These tests were run in February, 1952, on the frozen surface of Shawano Lake near Clintonville, Wis. The driving course used was the same as that used by the National Safety Council's Winter Driving Hazards Committee. The motion pictures taken during these tests give visual and positive evidence that this individual brake control system does prevent the violent skids that were present when conventional braking only was used. These titles must substitute for the actual pictures for readers of this paper.

The following titles appear before the scene in the movie in every case except the last one, which follows the last scene.

"A highway vehicle skidding out of control is a safety hazard. A rolling tire provides control. A sliding tire prevents control. Safety demands that wheels be kept rolling.

"This picture shows that proper individual control devices can prevent out-of-control skidding in trucks and buses. It also demonstrates that a tractor-trailer equipped with this device does not jackknife.

"This film illustrates the advantages to be gained from individual control devices. A 400-ft-diameter circle on the ice at Shawano Lake, Wis., used for testing is now shown.

Fig. 6 – Truck installation showing how sensing unit was attached by means of adaptor to rear wheels

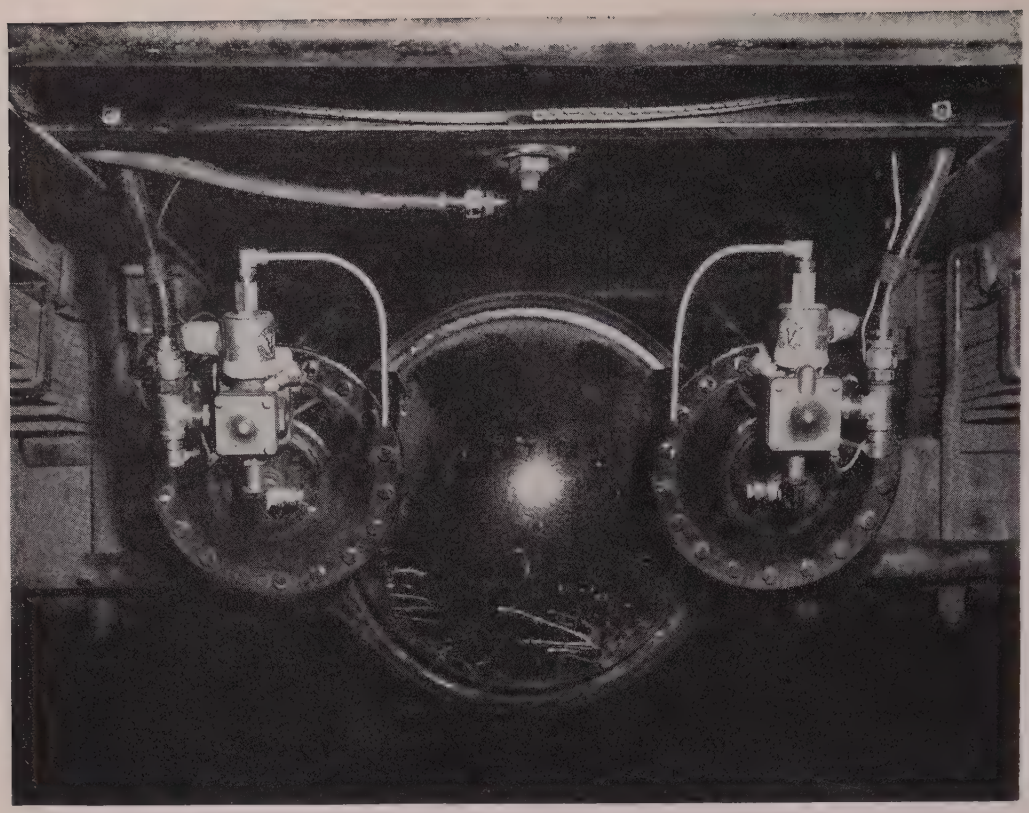


Fig. 7 – Rear axle of truck shows solenoid valves mounted directly on brake diaphragm cases

"Here are three views of a truck making stops on ice *without* individual control. On the second and third stop, the driver unsuccessfully attempts to prevent skidding by fanning his brakes, which means rapidly applying and releasing brake pressure.

"You now see this truck making two stops under the same conditions but *with* individual controls on the two rear wheels.

"The fact that the control device senses an impending skid and prevents it by keeping the tire rolling is the important conclusion from what you have just seen.

"Shorter stops can be made by putting the right wheels on the packed snow often present on an icy road. A test course of this type is now shown.

"Stops from the same speed with all wheels locked are now shown on the ice alone in 132 ft and on the snow alone in 75 ft.

"These stops show that a difference in road friction exists between the ice and snow, and hence between the two sides of the vehicle when straddling these surfaces.

"This differential friction results in a side force which pulls the vehicle off the road. Individual control devices remove this tendency. First we show a stop *without* these controls.

"The differential friction condition caused the vehicle to swerve toward the direction of higher ground coefficient. You now witness the same type stop except *with* individual controls.

"The next series of stops show a tractor-trailer with and without individual control equipment. Jackknifing is caused by the rear wheels of the tractor getting out of control.

"The next two scenes are *without* individual control. In both cases a dangerous jackknife develops. A special cab protector was installed to prevent injury to equipment and personnel.

"Individual control eliminates this jackknifing, as shown in the following stop.

"Observe this stop once more.

"These devices result in a remarkable improvement in controllability and would be a real contribution to highway safety."

We believe that these pictures are very convincing proof of our theory that individual wheel control that prevents wheel stoppage can be most effective in preventing dangerous skids on slippery roads.

(Included at the end of this paper is a table giving all the necessary characteristics of the various elements used in the sensing circuits.)

Our original objective in running these tests was to satisfy our curiosity and prove our theory that individual control of each wheel would be effective in preventing skids, and their probable resultant loss of control of the vehicle by the driver. We did not try to control anything but the driving wheels. We do believe, however, it is fair to conclude that the ultimate in control would require such control on all wheels having brakes.

It is our hope that these conclusive tests will point the way to many future tests and experiments

by other people. Our apparatus was crude. Further study will, we feel sure, show many opportunities for refinement and improvement. We are more than ever convinced that rolling wheels, kept rolling, will permit all drivers to follow the "straight and narrow," the best road to highway safety when the going is slippery.

Following is a detailed list of the various items of equipment used in assembling the control device reported on in this paper:

1. Elinco generator, Electric Indicator Co.; type CB-214; power, 5 w; v/rpm, 24.3 v per 1000 rpm (this output can be affected by surrounding magnetic material).

2. Condenser; type, metallized paper; working voltage, 150 v d-c; capacity, 12 microfarad.

3. Current-sensitive relay, Barber-Colman Co.; type 3887P; coil resistance, 2000 ohms; voltage sensitivity, 0.5 v.

4. Power relay, Guardian Electric Co.; No. 24303; operating voltage, 6 v d-c; coil resistance, 165 ohms; contact arrangement, single pole, single throw.

5. Solenoid valve, Skinner Electric Valve Co.; 3-way, normally open, pilot-operated; Skinner type M3; 6-v d-c solenoid; maximum operating pressure, 150 psi; maximum power consumption, 10 w; coil resistance, 4.3 ohms.

Summary of Oral Discussion

Reported by R. L. Sailer,
Freedom-Valvoline Oil Co.

STEPHEN JOHNSON, Bendix-Westinghouse Automotive Air Brake Co., noted that the paper reported results with controls on the drive wheels and commented on studies which indicated brake controls on all wheels would be most desirable, and on front wheels would be the next desirable condition.

F. B. Lautzenhiser, International Harvester Co., stated that synchronizing of brakes on all wheels is needed. Peter Hay of Vickers, Inc., stressed the safety features of this antiskid device and stated that his company is always interested in any safety brake.

M. C. Horine, Mack Mfg. Corp., commented on the simplicity of the unit which detects impending brake lock and noted that it responds only to rapid speed change. He pointed out that in the motion picture the wheel is locked shortly before the vehicle comes to a stop and it stops locked. He asked why it locked. Mr. Gunsaulus answered that when the rpm of the wheel approaches zero the speed is out of the sensitivity range of the apparatus, and that further work would be needed to determine if added sensitivity in this range could be devised, and if it was really needed to prevent skid.

Hoy Stevens, Bureau of Public Roads, was pleased to see results coming from winter tests, and asked if any work was done on wet pavement and with high pressures in the range of 60 to 70 lb. He was also interested in testing under conditions of running down hill on dry pavement, on to wet pavement where sliding can occur. The author stated that tests had been made on wet pavement but not yet on hills.

E. P. Gohn, Atlantic Refining Co., referred to Mr. Johnson's comments on simultaneous locking of all wheels, and agreed that synchronization is important but other variables exist. He pointed out that a new truck starts out in proper adjustment of brake but this is difficult to maintain. He approved the device described in the paper, but cautioned that it should not be expected to cover for poor maintenance.

Mr. Dunham, Ohio Bell Telephone Co., cited experience in the synchronization of electric brakes, and felt that the author of the paper was on the right track in solving brake control problems.

W. R. Williams, Bendix Aviation Corp., raised the problem of braking under hazardous and semihazardous conditions, and asked if the device might extend the stopping distance on a wet semihazardous pavement. He also noted that most tests were run on air brakes and suggested the possibility that with air over hydraulic or vacuum over hydraulic brakes, where there exists a degree of pedal feel, such a device may not be needed. The author replied that measurement of stopping distances was not included in the tests, but his opinion is that maximum braking is attained under conditions of incipient skid and, therefore, stopping distance should be reduced. This has been proved with airplanes, but not with trucks. While no work was done evaluating pedal feel, this device is more sensitive than operator reaction and therefore it is needed on all brake systems.

Mr. Williams expressed concern about the time interval of wheel speedup with this device, and Mr. Gunsaulus indicated that the time element of the operation could be reduced by the development of faster valves.

G. J. Fabian, Four Wheel Drive Auto Co., reported that work at Clintonville was conducted on all types of units including passenger cars, and results showed that driver reaction from pedal feel is too slow to prevent skids. Even with straight hydraulic brakes, this device is desirable.

B. F. Jones, Autocar Co., asked about additional cost of a truck equipped with antiskid devices and the author replied that no information is available now because the equipment used was improvised from laboratory apparatus. No cost figures will be known until after automotive manufacturers design and build the device into axles and trucks.

J. V. Bassett, Raybestos-Manhattan, Inc., asked how long it would be before the device will become available, and Mr. Gunsaulus stated his company does not now contemplate its manufacture because it must be designed into automotive units. The answer depends upon the development activity of automotive manufacturers.

A. B. Hirtreiter, Reo Motors, Inc., inquired how much time is consumed before the wheels start to slow down in reference to ground speed. The author answered that this depends on the sensitivity of the device, but a relatively few rpm's take place.

Mr. Horine stated that the railroads have conducted similar test work on their equipment, and asked if any report was available. Mr. Gunsaulus stated that American Brakeblok has the answer to that question. R. P. Pogue, American Brakeblok Division of American Brake Shoe Co., reported that three devices have been tested. The problem there is somewhat different since there is a multiplicity of wheels as compared to one trailer. Further they apply sand, and they may encounter wet or frosted rails. Apparantly all forms of transportation have brake problems.

Mr. Johnson announced that the Winter Driving Hazards Committee would test three types of antiskid devices now on the market and perhaps more. Included will be the Scrag skid drag.

What's Ahead in

THE intense interest and serious engineering activity in planning interstellar space travel command attention by virtue of the broad differences of opinion in the selection of powerplants by which such achievement might be attained. The significant point of divergence is fuel. One group of eminent scientists and engineers believes in building a space ship of 7000 tons total weight projected into an orbit 1000 miles beyond the earth, using nitric acid and hydrozine as fuel. At present this source of energy is quite expensive.

A more conservative second group plans a space ship with an overall loaded weight of 12.7 tons, traveling at a maximum velocity of 17,000 mph, using gasoline and liquid oxygen as propellent. This fuel is relatively inexpensive.

The influence of fuel is a pertinent consideration in the development of any powerplant for whatever purpose. An attempt to search the crystal ball for future powerplants must perforce define the fuel to be used as the source of energy, if we would see the image evolve clearly.

The subject of this paper is not intended to deal with a Buck Rogers dream or go beyond the horizon of foreseeable possibilities in vehicle powerplants. Demands of the commercial market focus attention on those developments which are in progress, and which can be expected to appear in field usage within the next five or more years.

For the purposes of this discussion, commercial vehicles are those used on highways and in off-highway services, and consideration is given to significant powerplant developments for heavy-duty trucks, buses, and rubber-tired earthmoving machinery.

The West has held an important place in the development of the internal-combustion engine since the earliest days of the otto cycle. This has been fostered by:

1. Demands of rugged service conditions due to long distances, high mountain terrain, and development of extensive areas.
2. Operation of equipment over the entire year due to favorable climatic and distribution facilities.
3. Availability of economical fuels.
4. Inventive enterprise.

All these factors have contributed to providing a lucrative market for the internal-combustion engine on the West Coast.

Influence of Fuel Cost and Availability

A curve showing the price trend of available fuels on the West Coast is shown in Fig. 1. Here it is seen that the price of gasoline is constantly going up. The demand for gasoline has risen because of the great passenger-car registration in western territory. Diesel fuel No. 2 shows a tendency to be led by gasoline costs. Diesel fuel No. 1 or premium diesel fuel is not shown because of the variation in price due to differences in distribution cost in the various territories. Middle-of-the-barrel cuts are constantly being competed for by both gasoline and diesel-fuel demands. Propane shows need for stabilization of cost. Residual fuels present the real potential market for low-cost fuels for commercial vehicle operation.

Recently D. J. Russell, president of the Southern Pacific Co., addressing members of the American Railroads Association, emphasized the point that the railroads should be seriously concerned with research on petroleum products in two directions: first, toward the use of superproducts available at higher prices but yielding lower operating costs, and second, the use of petroleum products lower in cost but sufficiently economical operationally to warrant reasonable maintenance cost. The two must be challenged against each other in order to evaluate the direction in which major users of diesel fuels must go. Railroads today are purchasing over 50% of the diesel fuels made available.

Recent issues of various petroleum news bulletins indicate a surplus of residual fuels. This situation will become increasingly aggravated should we ever go back to a peacetime economy in which the petroleum industry would be released from the demands of the military. The pile-up of residual fuels is a real opportunity to concentrate interest on this cut as a source of energy for internal-combustion engines. It is a well-known fact that residual fuels, *per se,* would not be satisfactory in most high-speed internal-combustion engines. However, efforts are being made to condition these fuels for consumption in diesel engines. Table 1 describes a fuel of this character. In other words,

Commercial Vehicle Powerplants

C. G. A. Rosen, Caterpillar Tractor Co.

This paper was presented at the SAE National West Coast Meeting, San Francisco, Aug. 13, 1952.

the fuel contains some 12% of residual material. The sulfur content is 1.09%. The cetane number, however, is 35.3. This fuel has some extraordinary properties, as indicated in the photograph in Fig. 2. This is a piston taken from an L-1 test using the fuel described in Table 1, with Supplement 1 oil in the crankcase.

Extended tests of this fuel are being run on various pieces of equipment, and to date have indicated outstanding results. Prior to these tests, at our Arizona Proving Grounds, a tractor was operated for a total of 3746 hr on PS-300 fuel. The last 2600 hr of this test were conducted at a compression ratio of 20/1. This was necessary because of the low cetane value of the fuel, which was about 20. This rules out its possibilities for use in most diesel equipment. Furthermore, PS-300 is a fuel of such widely varying characteristics that no uniformity could be expected in operating results. PS-300 fuel is not to be confused with residual fuel especially prepared for diesel-engine use as described in Table 1. PS-300 fuel required considerable accessory equipment in order to permit it to flow properly to the fuel pumps, and to provide proper starting characteristics. The residual fuel of Table 1 was eminently satisfactory, inasmuch as heat from only the jacket system permitted functioning of the unit under adverse West Coast atmospheric conditions.

If demand for premium diesel fuels continues to rise, the price will naturally approach that of gasoline. Under such circumstances, it is well not to rule out the possibilities of the high-compression spark-ignition engine. Tests conducted at 12/1 compression ratios indicate very favorable fuel consumptions and desirable torque characteristics for commercial vehicle powerplants. Equipment to provide stratified mixtures in a single-cylinder engine has proved quite successful in achieving 128 psi bmep unsupercharged at fuel consumptions of

THIS author sees a need, in the near future, for commercial vehicles with engines of 1000 to 1200 hp – powerplants that yield high outputs but require limited space. He sees an immediate need for more and more horsepower per cubic inch of piston displacement and per unit of space for the engine.

He directs attention to six design potentials which may supply the answer: (1) the gas turbine; (2) supercharging; (3) aircooled diesels; (4) higher engine speeds; (5) 2-stroke diesel improvement; (6) compound engines.

He also links the future development of the internal-combustion engine with basic improvement of components through simplification, calling for the elimination of extraneous gadgetry.

The Author

C. G. A. ROSEN (M '37) is consulting engineer, Administrative Department, Caterpillar Tractor Co., Peoria, Ill. He graduated from Cogswell Polytechnic College and the University of California with B.S. and M.E. degrees. Mr. Rosen was SAE vice-president representing the Diesel-Engine Activity in 1940, and has been an active member of the Fuels and Lubricants Activity Committee. He was also a member of the SAE War Engineering Board.

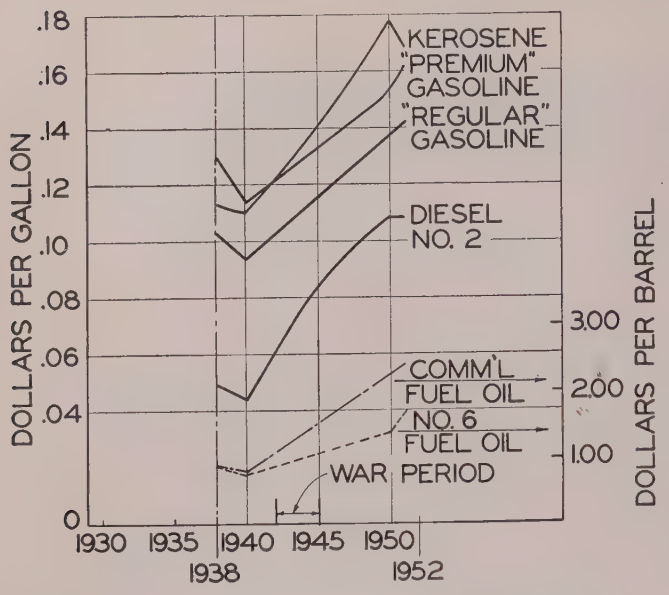


Fig. 1 – Trend in fuel costs on West Coast

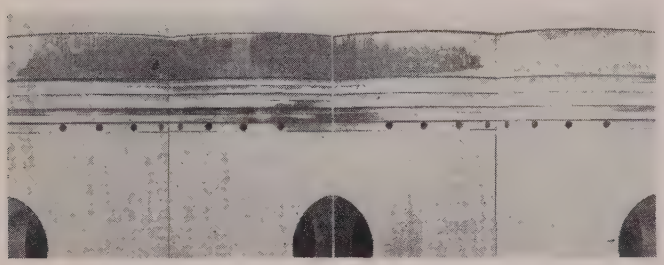


Fig. 2 – Piston from L-1 test on residual fuel

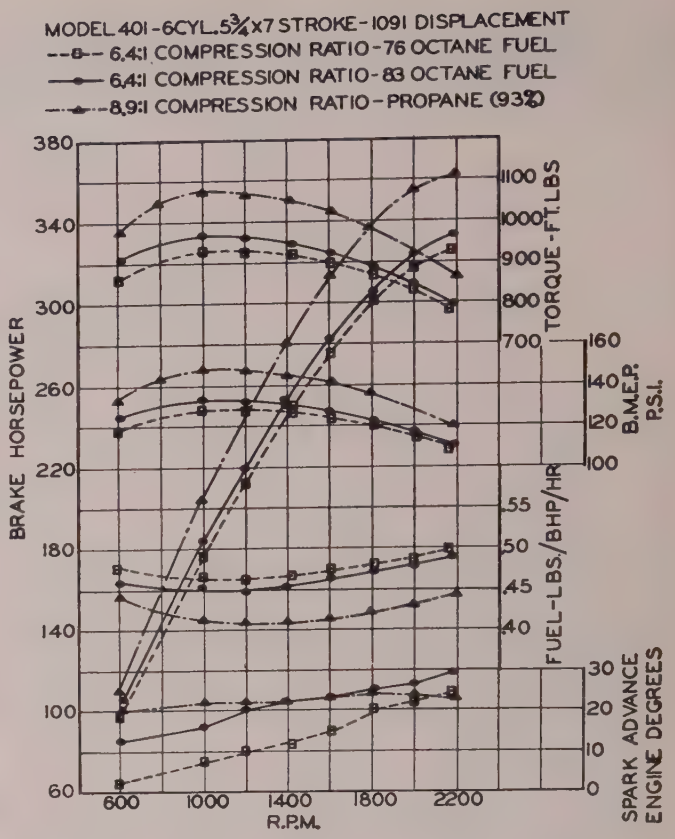


Fig. 3 – Truck engine performance on propane and gasoline

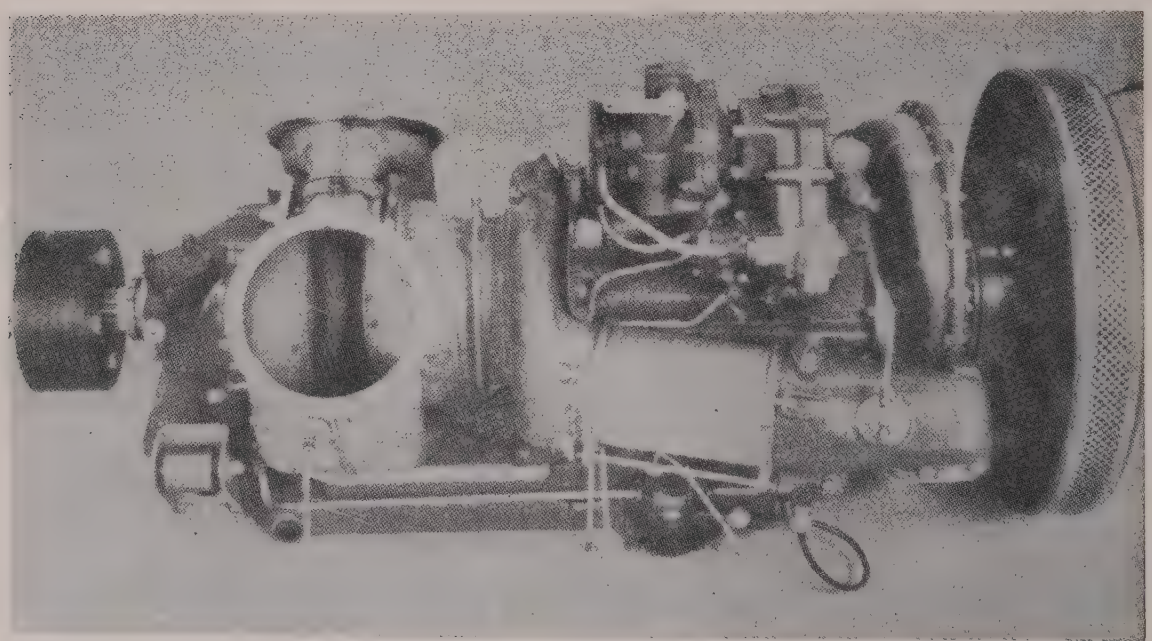


Fig. 4 – Boeing gas-turbine unit for truck installation

Table 1 – Diesel Residual Fuel Specifications

Gravity, deg API	25.4
Flash Point, F	164
Color	Black
Pour Point, F	45
Initial Boiling Point, F	346
10% Recovered, F	428
50% Recovered, F	588
88% Recovered, F	760
Sulfur, %	1.09
Corrosion, at 122 and 212 F	Positive
Ash, %	Nil
Carbon (10% Btms)	1.13
Aniline Number	126.5
Cetane Number	35.2

0.45 lb per bhp-hr using 100-octane gasoline. On diesel fuel, this unit achieved 111 lb bmep at 0.47 lb per bhp-hr.

Likewise, performance properties of commercial propane have revealed interesting advantages. Fig. 3, taken from a paper by J. E. Glidewell,[1] indicates that engines using propane, at 9/1 compression ratio, will give brake mean effective pressures which are comparable to those of supercharged diesel engines. The fuel consumption is also of significant importance. Some of these figures have been verified elsewhere in practice – for example, by the Chicago Transit Co. The principal problems concerning the broadened use of propane are:

1. Need for commercially available propane without admixtures of propylene and butane, to utilize the highest possible compression ratios conducive to desirable efficiencies and low fuel consumptions.

2. Need for dependable and uniform sources of supply.

3. Stabilization of cost in relation to other motor fuels, and not in relation to seasons or to household demands for cooking gas, and so forth.

In some of the Chicago Transit Co.'s operations the overall cost of operating on propane has given the diesel engine a real run for its money. On several earthmoving projects, propane has made a noticeable contribution in lowering the cost of moving earth. Heavy demands by refineries for middle-of-the-barrel cuts for use as charging-stock in making gasoline, the requirements of military services, particularly the Navy and the Air Force (for jet planes), together with large railroad demands, all place a constraining future on the supply of premium diesel fuels for high-speed diesel engines. The powerplant designer must be concerned with designing equipment suitable for fuels which have a trend toward costs lower than at present foreseeable with premium diesel fuel. Military services have ruled out the use of diesel fuel in military vehicles because of two factors:

1. The unavailability of sufficient quantities of diesel fuel, in the event of an emergency, to satisfy wholesale conversion to diesel powerplants.

2. The difficulty of distributing two fuels, diesel fuel and gasoline, to combat areas.

On the other hand there is great need for equipment which will provide economical fuel consumption, particularly at light load or during idling. The diesel cycle is the most economical cycle for providing fuel economy over a wide range of load and speed.

Influence of Installation Requirements

In Mr. Glidewell's paper, referred to previously, he makes the statement: "It seems to be generally understood that most large highway trucks are underpowered, or at least could use more power to advantage, especially in the West where gross weights of 68,000 to 76,000 are allowed, and where there are many grades of 7% and altitudes of 7000 or 8000 ft. It would seem that the trucking industry could do a better job of selling itself to the public if trucks were powered to keep up a minimum of 25 to 30 mph on highway grades, whereas at present, many trucks on some grades crawl at 3 to 10 mph. One great advantage in having increased power in trucks, especially on long hauls, is to reduce running time."

History is repeating itself. We know the story of the automobile, and of how the earliest models were seriously underpowered. Gradually, customer demands showed the way toward using higher-powered engines in automobiles in order to provide desired performance. The motor truck went through the same sequence of events, but at a later period of time. At the present time, the earthmoving industry is in this same cycle of moving up in power demands to provide desired performance and to reduce operating costs. In present rubber-tired earthmoving equipment, engines as high as 500 hp are being used. Contractors and operators could very well use 700-hp engines and obtain many desirable advantages. For instance, on one particular job, a scraper fitted with a 90-hp engine showed a cost of 20¢ per cu yd for moving earth. By installing an engine of 250 hp in this same piece of equipment, the work was done at 15½¢ per cu yd. The desire to move loads at high speeds on steep grades will foster the necessity for the manufacturer to provide higher-powered engines. It is quite conceivable that the day is not far distant when 1000 to 1200 hp will be needed in order to accomplish the desired reduction in earthmoving operating costs. In providing such horsepowers, the powerplant becomes a dimensional problem concerned with space and weight limitations.

The trend in vehicle powerplants for commercial usage is:

1. Higher horsepower outputs per cu in. of piston displacement.

2. Higher horsepower per unit of space.

[1] "Large Conventional Engines for Trucks and Buses," by J. E. Glidewell. Paper presented at SAE National Transportation Meeting, Chicago, Oct. 29, 1951.

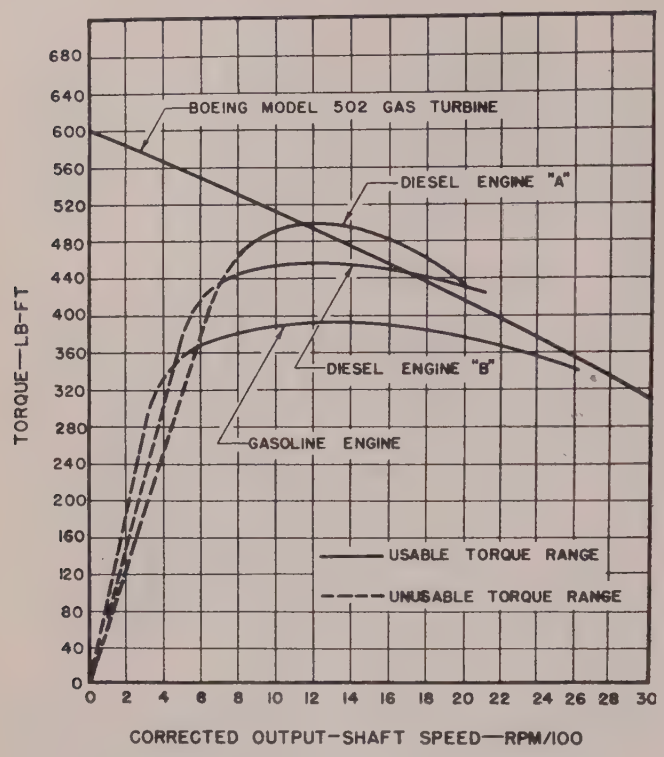


Fig. 5 – Gas-turbine torque curves as compared with diesel engine in truck installation

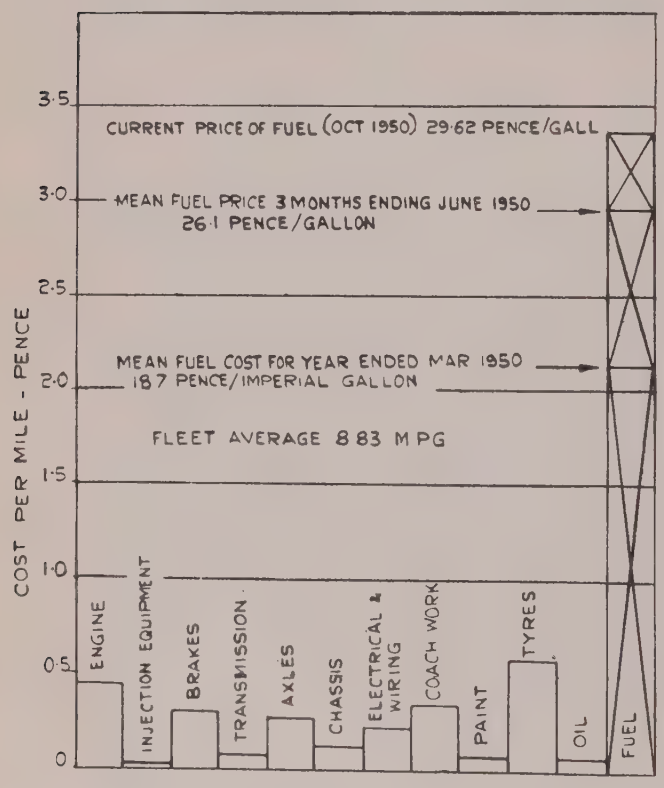


Fig. 6 – Breakdown of operating costs of London buses

As we project our vision to the horizon of possibility, we see the milestones of achievement trending in this direction. When one is confronted with the necessity of providing 1000 to 1200 hp in commercial vehicles to achieve a higher output within limits of space, one turns to the prospects of the gas turbine.

Borrowing from H. C. Hill's paper,[2] the unit shown in Fig. 4 immediately conjures up a picture of lots of power in small space at low weight ratios. The performance characteristics as compared with the diesel are shown in Fig. 5.[2] Particularly for earthmoving equipment, the torque characteristics of the gas turbine have a fascinating appeal. Such torque characteristics also are influential in reducing the cost of the transmission system. It provides wider variation of speed changes with simple mechanisms. Naturally, such equipment must compete with the diesel engine in overall operating costs which include first cost, fuel cost (as related to fuel economy) and maintenance cost. It is undoubtedly true that, in time, the first cost of the gas turbine will be considerably reduced. Its exact commercial position in this respect is as yet uncertain. Its fuel economy is known to be much poorer than that of the diesel, but the possibilities of using multiple units to overcome high consumption rates at light loads on gas turbines, may permit the cutting-in and cutting-out of certain units, resulting in desirable economies and satisfactory operating performance. In scraper operation, for instance, it requires twice as much power to load as it does to run with the load. Under such conditions, the powerplant could be cut in half and one unit shut down after the scraper is loaded. This scheme also suggests the use of an auxiliary unit such as a gas turbine as a booster unit to meet necessary power requirements for short periods. More reasonable economy can be attained on low-cost fuels than on fuels necessary in diesel practice, particularly when premium fuels are required for the latter service.

Here it is to be noted that we should never consider burning Bunker C or other uncertain types of uncontrolled specification fuels in vehicle gas-turbine operation. The railroad situation has definitely shown that the problems of burning Bunker C fuel are far more difficult than originally anticipated. However, residual fuels of the character indicated in Table 1 offer promise of low maintenance in gas-turbine operation.

In our American economy it is difficult to evaluate the significance of specific fuel consumption. Great emphasis is placed on this factor in Europe, as shown in Fig. 6, taken from a paper by C. B. Dicksee.[3] In Fig. 6 the relative evaluation of the

[2] See SAE *Quarterly* Transactions, Vol. 6, July, 1952, pp. 395-408: "Progress of Gas Turbine Truck Tests," by H. C. Hill.

[3] See SAE *Quarterly* Transactions, Vol. 5, April, 1951, pp. 151-172: "Diesel-Engine Design and Performance in Great Britain," by C. B. Dicksee.

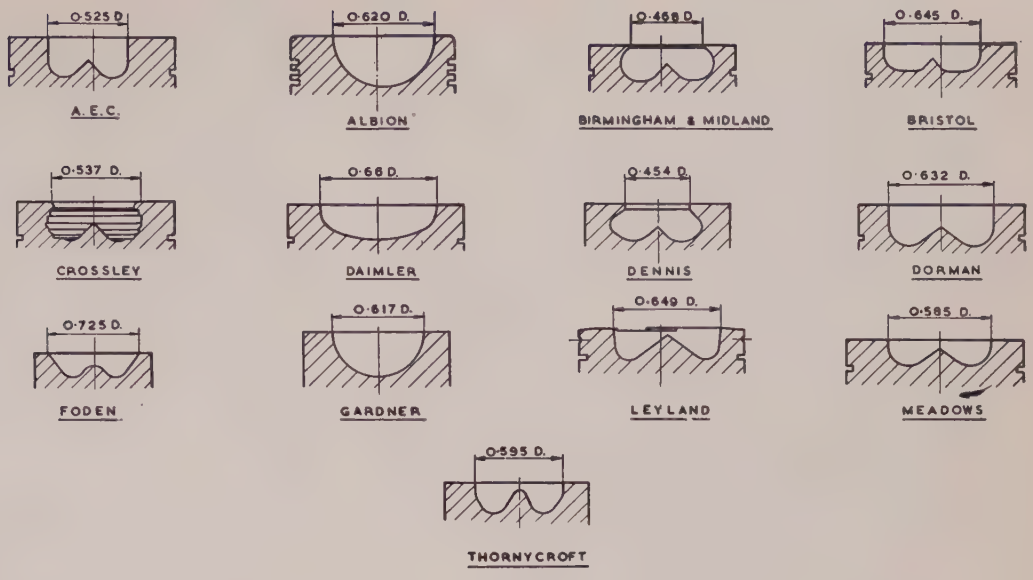


Fig. 7 – Direct-injection toroidal combustion chambers

factors entering into bus operation is shown in barograph form. Here it can be seen that fuel is the highest factor in the operating cost of the British omnibus. The placing of such significant values on fuel cost has naturally fostered development of the direct-injection combustion system. Various combustion chambers which have been developed to achieve high economy of fuel are shown in Fig. 7.[3] The AEC design, as developed by Mr. Dicksee, has been responsible for achieving fuel consumptions as low as 0.36 lb per bhp-hr. Interest is being displayed in the fact that licenses have been obtained recently in the United States for two of the engines shown in Fig. 8.[3] These are to be used in the truck and bus field. In bus operation, particularly, where extreme cleanliness of exhaust and absence of odor are highly desirable in congested areas, the exhaust must be free of contaminants which produce undesirable effects. American manufacturers are becoming cognizant of the appeal of this engine which gives satisfactory operating performance without resorting to premium fuels.

Developments now in process should aid in settling the contest between the use of premium fuels and available low-cost fuels. Operations using premium fuels at higher cost per gallon should be weighed against the possibility of similar performance with lower-cost fuels, and resultant maintenance and operational advantages compared.

The trend toward 4-stroke supercharged vehicle engines is quite significant. Here is a real opportunity to achieve higher output per cu in. of displacement and to take advantage of the corre-

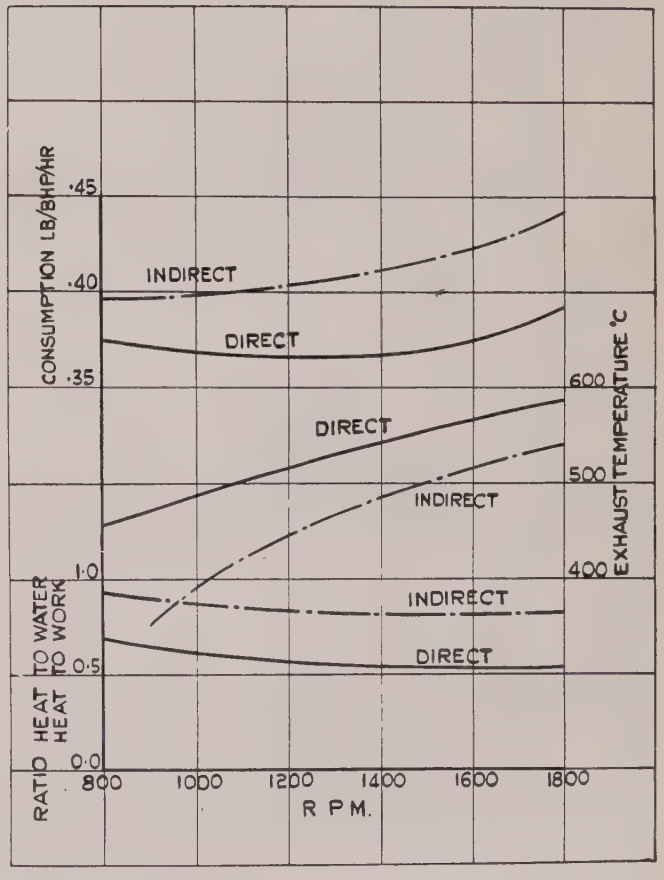


Fig. 8 – Performance data of high-economy British bus engines

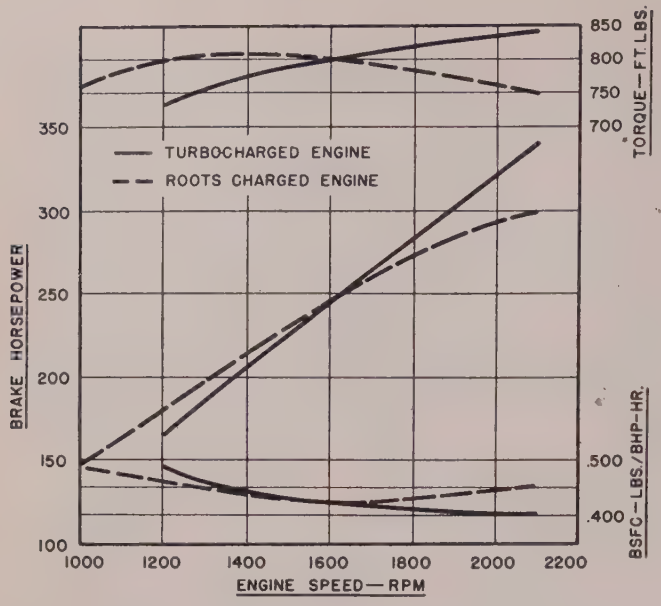


Fig. 9 – Comparative performance curves for turbocharged and Roots-charged engine

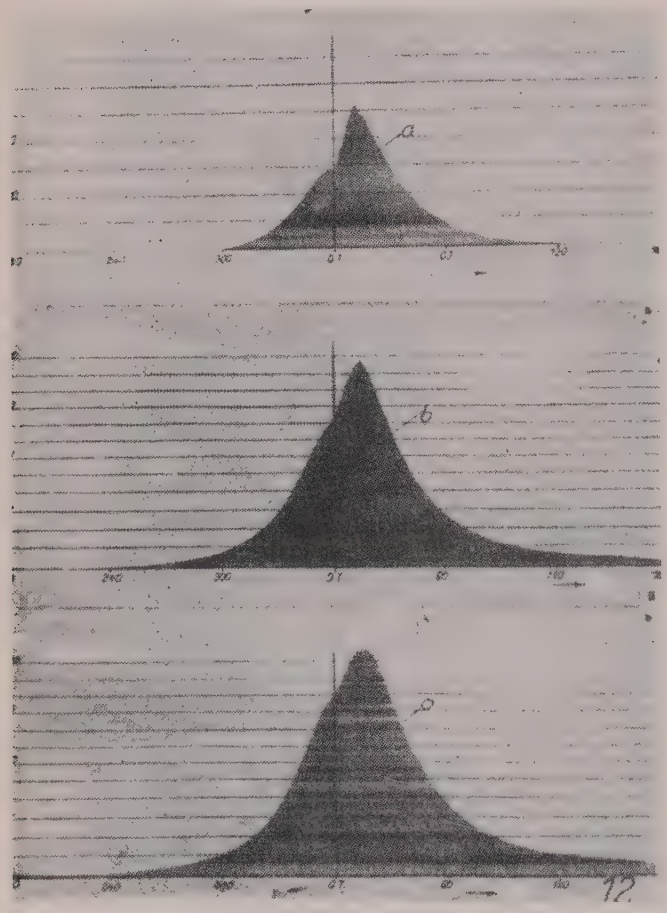


Fig. 11 – Indicator cards comparing nonsupercharged and highly supercharged MAN diesel engines

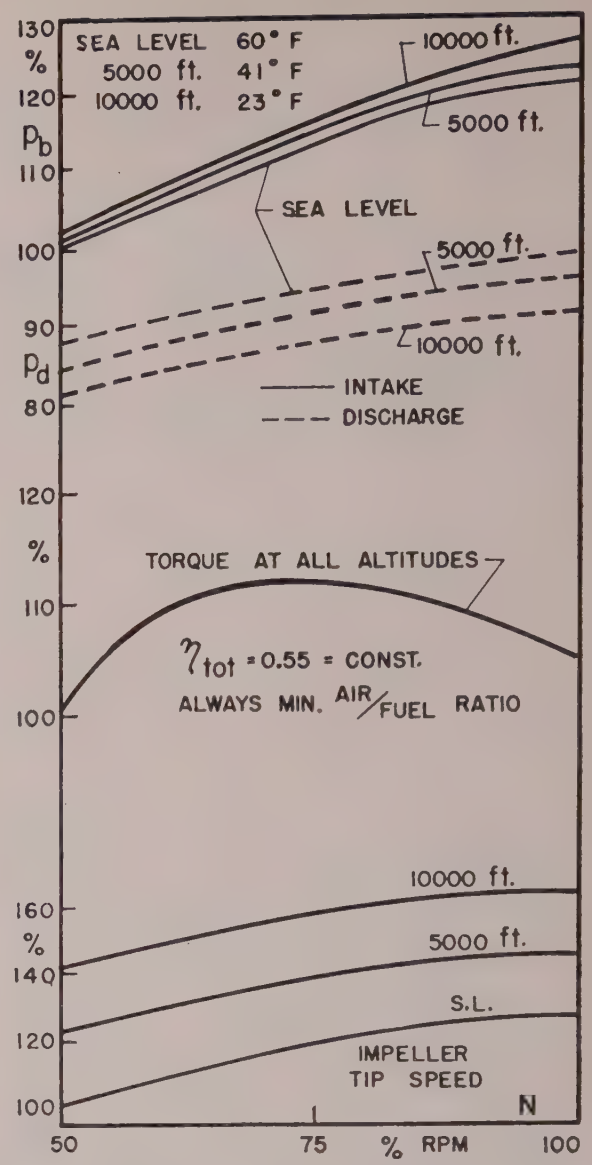


Fig. 10 — Performance characteristics of turbosupercharger system using variable-area nozzle control

Fig. 12 – Deutz aircooled diesel engine for vehicles

sponding reduction in space requirements. The supercharged 4-stroke diesel engine, as now manufactured and marketed for vehicle use, is averaging about 135 psi bmep with about a 6 to 10-lb boost in supercharge pressure. The real crux of the problem in achieving still higher output is the blower. Under present conditions the blower is retarding the possibilities of supercharging because of its inefficiency or its inability to match the performance requirements of the vehicle. The Roots blower has definite limitations, and it is generally conceded by authorities like Ricardo that its optimum discharge pressure is about 7½ psi. The centrifugal supercharger has possibilities, but has not been fully explored as to its adaptability to variable-load, variable-speed engines. One promising goal of supercharging is the utilization of the turbosupercharger. A problem exists, however, in matching the turbosupercharger to the engine in order to achieve the desired torque characteristics, as illustrated in Fig. 9 and discussed by Reiners and Schwab.[4] The positive-displacement blower has the opportunity, through suitable controls, of providing the desired torque curve for vehicle operation, whereas, inherently the turbosupercharged engine has a drooping torque curve not satisfactory for variable-speed operation. The influence of these two blowers is revealed in the specific fuel consumptions obtained. However, the possibilities have not been fully developed in practice as revealed by W. T. von der Nuell.[5]

The problem of matching the blower to achieve desired torque characteristics under varying altitude conditions and under extremes of ambient temperature, calls for construction features which modify the performance of the turboblower to obtain satisfactory results. Fig. 10, taken from Dr. von der Nuell's paper,[5] illustrates the opportunities for achieving desired performance by a system of variable-nozzle controls of the gas-turbine unit of the turbosupercharger.

The significant development by MAN of Germany,[6] as shown in Fig. 11, in achieving high output (200 to 300 psi bmep) in supercharged diesel engines, is very refreshing and bids well to stimulate comparable developments in 4-stroke diesel vehicle engines. At the present time, where the top performance of the high-speed 4-stroke supercharged diesel engine may seem limited by the blower, there are further opportunities in the thermodynamic cycle to achieve higher output with low heat distress loads. The blower is largely responsible for problems associated with the filling and scavenging of the engine cylinder to achieve optimum combustion performance. The retention of residual gases, or inability of the system to fill the cylinder completely with air at relatively low temperature of charge, presents an intriguing challenge.

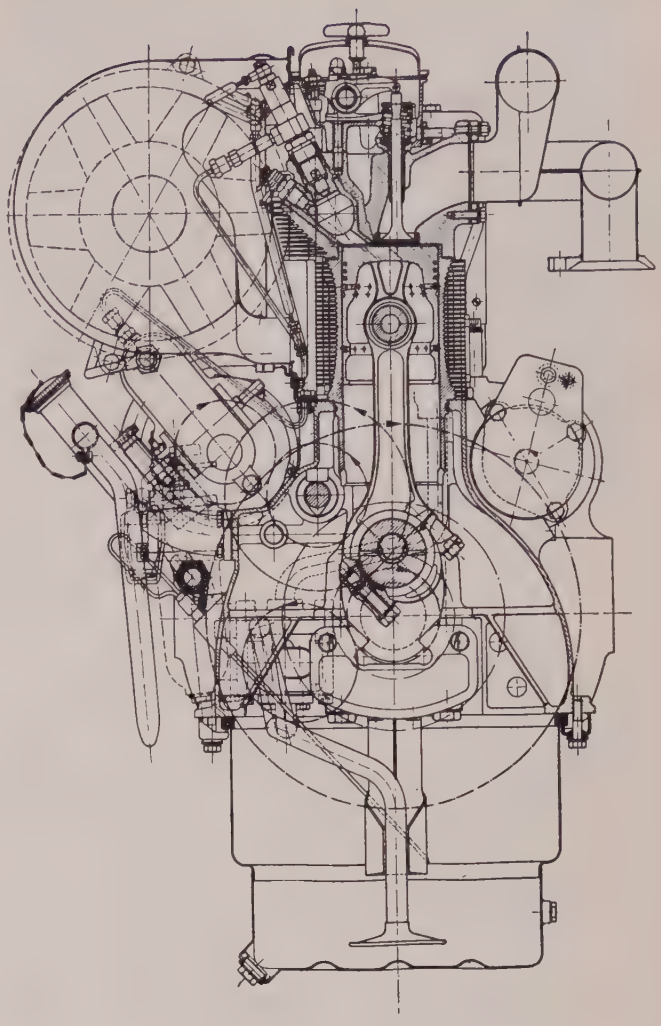


Fig. 13 – Cross-section of Deutz divided combustion chamber diesel engine

The MAN engineers have achieved increased outputs of 300% by efficient supercharging, with only 30% increase in heat load through the piston. This has been accomplished by the superior loading of the cylinder with excess pounds of air, in such form that it is readily available to the injected fuel for rapid combustion without shock and with completion of burning during the early part of the expansion stroke. Particular attention has been given to the cycle from 25 deg before firing top-center to about 30 deg after top-center. In this part of the cycle, the peak pressures have been raised to 1700 or 1800 psi for 215 psi bmep output. In

[4] "Turbosupercharging of High-Speed Diesel Engines," by N. M. Reiners and W. D. Schwab. Paper presented at SAE National West Coast Meeting, Seattle, Aug. 13, 1951.

[5] See SAE *Quarterly* Transactions, Vol. 6, October, 1952, pp. 753-782: "Superchargers and Their Comparative Performance," by W. T. von der Nuell.

[6] See *Verein Deutscher Ingenieur Zeitschrift*, Vol. 93, Jan. 21, 1951, pp. 66-67: "Supercharging of 4-Stroke-Cycle Diesel Engines," by Karl Zinner.

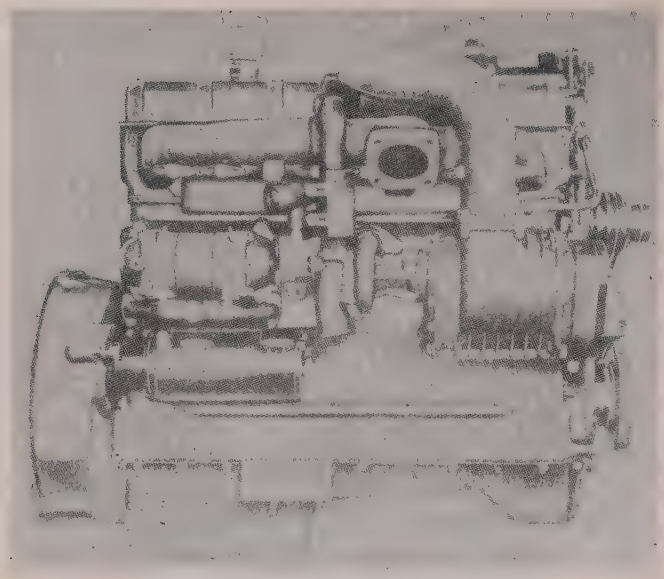


Fig. 14 – Cummins 4000-rpm light-weight diesel engine

view of the fact that during this period the piston is moving slowly and uncovers a minimum of cylinder wall surface, the accomplishment of rapid combustion, without shock, yields high output with low heat distress and a fuel consumption of 0.308 lb per bhp hr. In supercharging high-output engines, it is essential to prevent the deadly afterburning, or combustion of fuel, far down the stroke, thus overheating the piston and causing distress to rings, cylinders, and so forth, and ultimately resulting in poor fuel consumption.

There is a very inviting challenge toward the

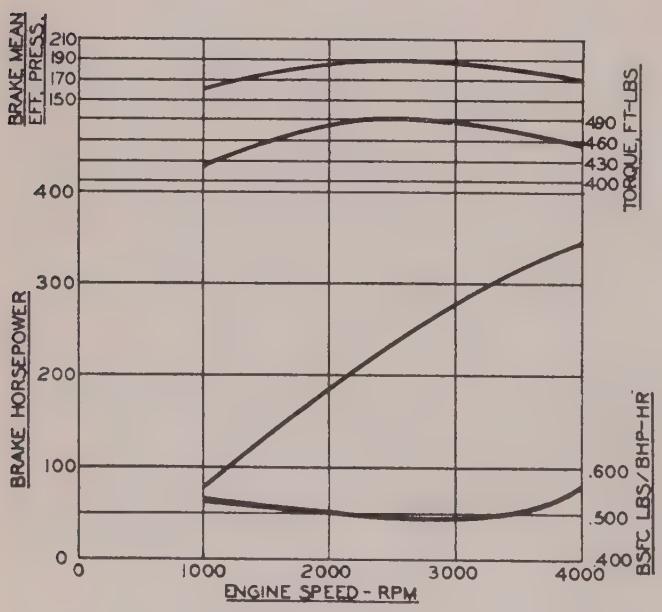


Fig. 15 – Performance curve of Cummins high-speed diesel engine

achievement of higher output in the future in supercharged diesel engines with improved blower adaptations and designs and greatly improved combustion characteristics. The 4-stroke supercharged diesel engine could well be increased to 300% of the present normally aspirated engine without undue stress of parts in thermal loading.

Before closing my reference to the 4-stroke diesel engine it would be well to consider the significance of recent European developments in achieving low weights of units by aircooled diesel engines. During World War II, considerable activity in Germany as well as in England and in Switzerland was centered around development of the aircooled diesel engine. Problems of portable operation over wide ranges of atmospheric temperature and the need for setting up emergency powerplants in almost any location, gave great impetus to the development of the aircooled diesel engine. At the present time a number of manufacturers on the Continent and several in England are making available aircooled diesel engines for vehicle and industrial application. The Deutz unit,[7] as illustrated in Fig. 12, is perhaps the most aggressively promoted aircooled diesel engine now being built in Germany. It is built in sizes from 12 to 175 hp and in designs from single-cylinder to V-8 engines. In the V-8 model the engine weighs 10.2 lb per bhp, using conventional materials and commercial manufacturing techniques. The bmep is about 80 psi for industrial applications and 90 psi for intermediate service. These engines are receiving considerable attention from the consuming public throughout the world. One of the factors which has contributed to the development of the aircooled engine in its present satisfactory form in Europe is the advancement made in cooling fans. The expanded knowledge of aerodynamics, associated with directing suitable cooling currents of air over cylinder fins and temperature-dissipating surfaces, has proved the desirability of the aircooled system.

Also of significance is the growing tendency toward the use of the swirl chamber-combustion system for this type of construction. The Swiss Locomotive Works has an aircooled engine with a direct fuel-injection system and a toroidal combustion chamber, but the Germans have centered their efforts on the swirl chamber.

Fig. 13 is a cross-section of the Deutz engine, and shows the design of the large-volume, big-throat swirl chamber.[7] This design is being promoted for future applications of supercharging.

Another avenue of approach in developing high-output engines is by the increase of engine speed. Significant in this direction is the development of the Cummins high-speed engine. The problems of designing diesel engines to operate at high speeds are even more difficult than for operation at high

[7] See *Motortechnische Zeitschrift*, Vol. 12, Oct. 19, 1951, pp. 125-131: "Die Deutzer Luftgekuhlten Diesel-Motoren," by J. P. Kramer.

bmep. The Cummins[8] unit is shown in Fig. 14, and its performance curve in Fig. 15.[8] Here the maximum mean effective pressure is 170 psi at 4000 rpm. The old theory that the diesel engine is hamstrung in speed, leads one to recall the publication of the old textbook on: "The Slow-Combustion, Internal-Combustion Engine – the Diesel Engine."

It has been found by the example shown in the Cummins engine, as well as by work conducted at Caterpillar on a small-bore, high-speed aircooled unit, that speeds up to 6000 rpm are achievable without jeopardizing essential diesel economy. It is anticipated that in the future more of these designs will be developed in an effort to capture the market for high-output, high-economy engines.

In 2-stroke diesel engine practice we have become accustomed to the capability of the uniflow scavenge system to achieve small packages at light weight, as in the GMC 110 Series diesel engines. The 2-stroke engine is not so susceptible to high-output improvement by supercharging as is the 4-stroke engine, although results indicate that a 30% increase in output is possible by the utilization of turboblowers for supercharging the 2-stroke diesel engine.

Of considerable interest at the present time are new developments in the loop-scavenged 2-stroke diesel engine. The GMC Series 50 diesel engine is used as a Chevrolet replacement unit in European markets. Recently, it has been noted that the German 2-stroke diesel engine is bidding for a place in commercial usage in American vehicles. Perhaps most significant in this direction is the Schnurle system of loop-scavenging as developed in the Krauss-Maffei engine (Figs. 16 and 17). This was described in an article by Paul Schweitzer.[9] The engine is now being investigated by several American companies for vehicle application. It is built in a V-design to achieve minimum overall length, and has manufacturing features which are of much interest in cost reduction of diesel engines. The loop-scavenged system permits an overall reduction in size, even though its output is not always as high as achieved by the uniflow scavenge system. This engine at 2200 rpm shows a fuel consumption of 0.38 lb per bhp hr, at a bmep of 70 psi. Peak loadings of 80 psi are indicated by performance curves.

Another bid for consideration in future powerplants is the so-called compound engine. In this development an opposed-piston engine furnishes the energy fed to the turbine, which is the source of power for propulsion. The illustration in Fig. 18[10] shows the general arrangement of this system

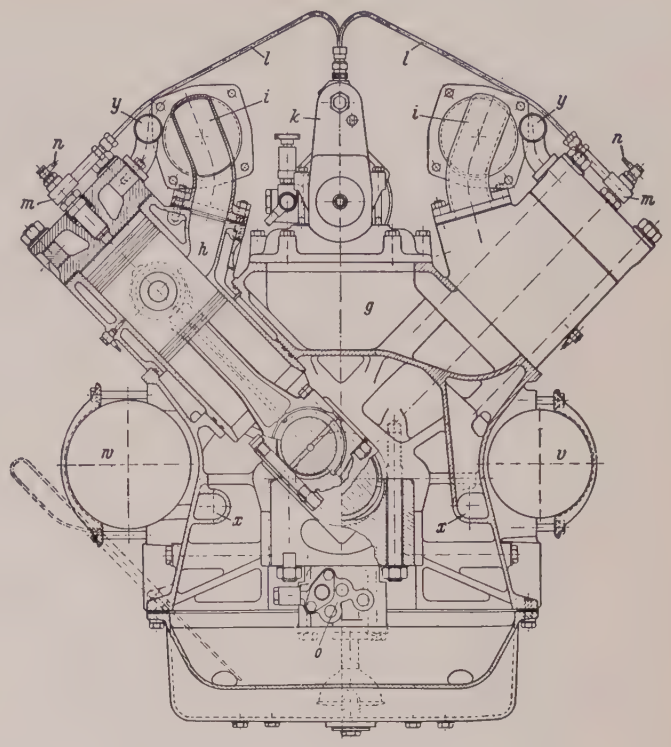


Fig. 16 – Krauss-Maffei 2-stroke Schnurle-scavenged diesel engine for vehicles (end section)

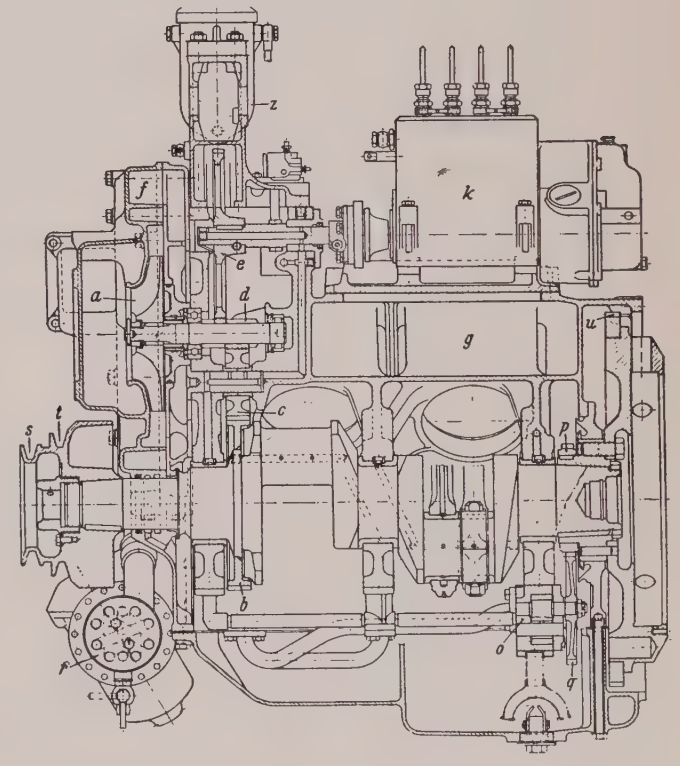


Fig. 17 – Krauss-Maffei 2-stroke Schnurle-scavenged diesel engine for vehicles (side section)

[8] "Experimental High-Speed Cummins Diesel Engine," by N. M. Reiners and R. C. Schmidt. Paper presented at SAE Summer Meeting, French Lick, June 6, 1951.

[9] See *Automotive Industries*, Vol. 106, Feb. 15, 1952, pp. 40-45: "Almost 300 Ton-Miles per Gallon with Two-Stroke Diesel," by F. Sass and P. H. Schweitzer.

[10] See ASME Transactions, Vol. 74, November, 1952, pp. 1349-1361: "Free-Piston Engine Development – Present Status and Design Aspects," by A. L. London and A. K. Oppenheim.

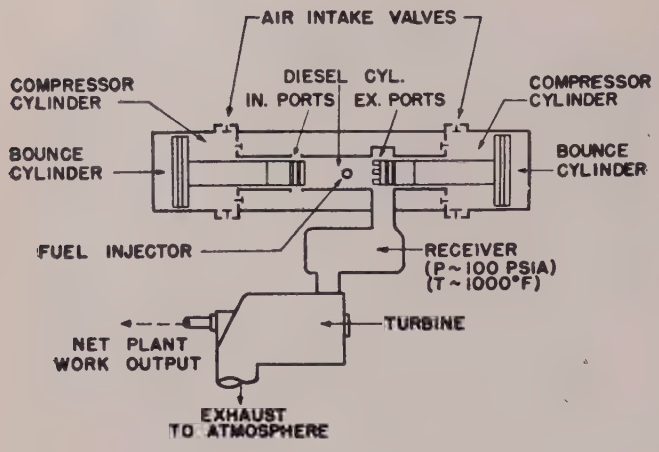


Fig. 18 – Schematic arrangement of compound powerplant

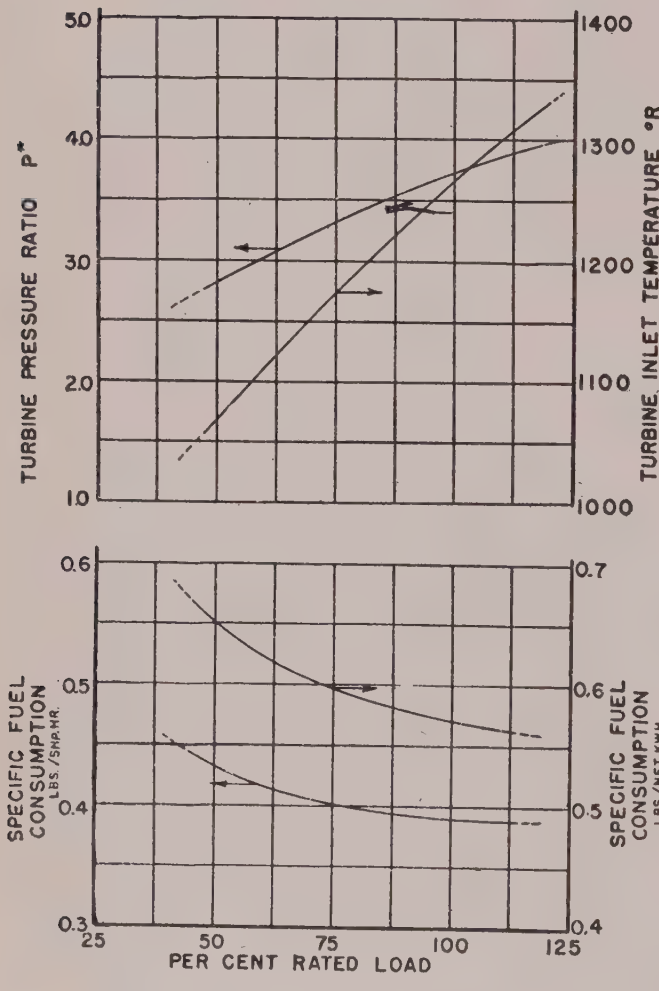


Fig. 19 – Performance curves of compound cycle

when utilizing the free-piston principle rather than the crank and flywheel opposed-piston engine. The net work is accomplished on the turbine shaft driving the vehicle. This powerplant is receiving attention for military purposes at the present time. These achievements may ultimately develop the practical potentialities of this system as means for certain types of vehicle propulsion.

A recently published paper by London and Oppenheim[10] is worthy of study and consideration. Fig. 19 shows performance data.

The SIGMA free-piston gas generator has operated in France with indicated mean effective pressures up to 230 psi. These authors show their confidence in this system:[10] "Such a system, while preserving the desirable torque speed characteristics of a simple-cycle gas-turbine plant with a separate power turbine, possesses substantial advantages in terms of lower turbine operating temperatures and better fuel economy. This free-piston application may be attractive as a truck powerplant, especially in view of the apparently serious consideration given to the use of the simple gas-turbine plant for this development." In this same connection, an aircraft development by Ricardo using a 2-stroke single-acting, uniflow-scavenged diesel engine with high supercharge as a heat generator supplying energy to a gas turbine, is reported as yielding fuel consumption below 0.30 lb per bhp hr.

Reference is also made to the work of NACA in a paper by Sather, Schuricht, and Biermann.[11] Work performed on two loop-scavenging systems and one uniflow system in opposed-piston engines, resulted in the following conclusions by the authors:

"The results of this investigation indicate that an engine of the 2-stroke-cycle gas-generator type would provide a powerplant of low specific weight and fuel consumption with favorable altitude performance characteristics. It was found possible to operate a 2-stroke-cycle cylinder at the pressure and temperature levels required for the gas-generator engine. Mechanical functioning of the piston engine under gas-generator operating conditions was in general satisfactory."

Here we have instances of powerplants which on the surface appear complex in the number of units involved, yet may, under the stimulus of engineering development, eventually serve as acceptable commercial units.

Improvement in Components

The development of future internal-combustion engines is quite well linked with fuel-injection equipment economies. Even in high-compression

[11] See ASME Transactions, Vol. 74, July, 1952, pp. 637-653: "Summary of Investigation of Two-Stroke-Cycle Gas-Generator Aircraft Engine," by B. I. Sather, F. R. Schuricht, and A. E. Biermann.

Fig. 20 – Evolution of fuel-pump design

spark-ignition engines, fuel-injection equipment is considered advantageous in obtaining the desired stratification of fuel mixtures. The use of fuel-injection equipment is mandatory in all diesel engines. There is need for concerted effort in the development of fuel-injection equipment along lines which will produce low-cost mechanisms that will measure up to expected performance. In this connection the development work conducted by Caterpillar on fuel-injection equipment[12] is of interest.

The evolution of the fuel pump for the past 20 years is best expressed by the sequence of designs in Fig. 20.[12] The original replacement, shown as A, was fitted with conventional parts for that period.

Design B is a modification for installation in a housing outside the engine, yet sealing the crankcase of the fuel-pump housing from dust and moisture. Designs C, D, and E, are intermediate in development but have improvements in the check-valve arrangement which resulted in better performance.

Design F is the most simplified unit. It is apparent that considerable reduction in cost has been achieved in the manufacture of this fuel pump, and that it is reduced to its essential working parts. The backflow channel from the lower portion of the barrel engages the lantern channel around the plunger with an oblique drilled hole, permitting the trapped fuel under pressure to be returned to the suction side of the fuel pump. This has resulted in considerable reduction in leakage of the plunger and barrel into the crankcase of the fuel-pump housing.

Fig. 21[12] shows in sections the stages of development in the evolution of the fuel valve. At A, the original pintle-type valve is shown. Various stages of development in improving and simplifying the closed nozzle are shown at B, C, D, and E. Also variations are shown in the design of the precombustion chamber to accommodate itself to the changing characteristics of the fuel valve.

[12] See The Institution of Mechanical Engineers, Automobile Division, Proceedings (London), 1950-1951, Part 1, pp. 7-25: "Significant Contributions of Diesel Research Laboratory," by C. G. A. Rosen.

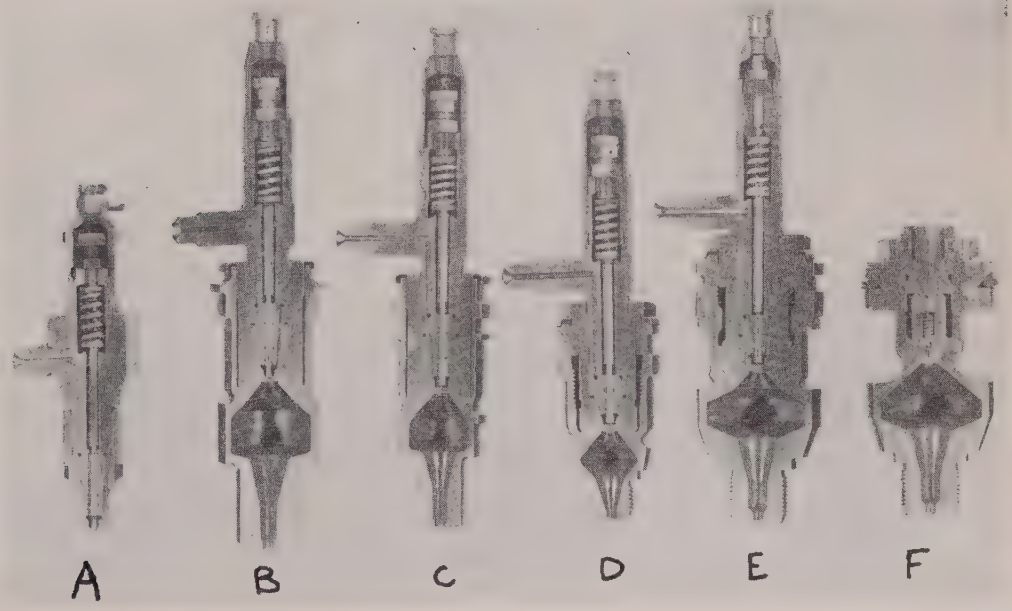


Fig. 21 – Evolution of fuel-valve design

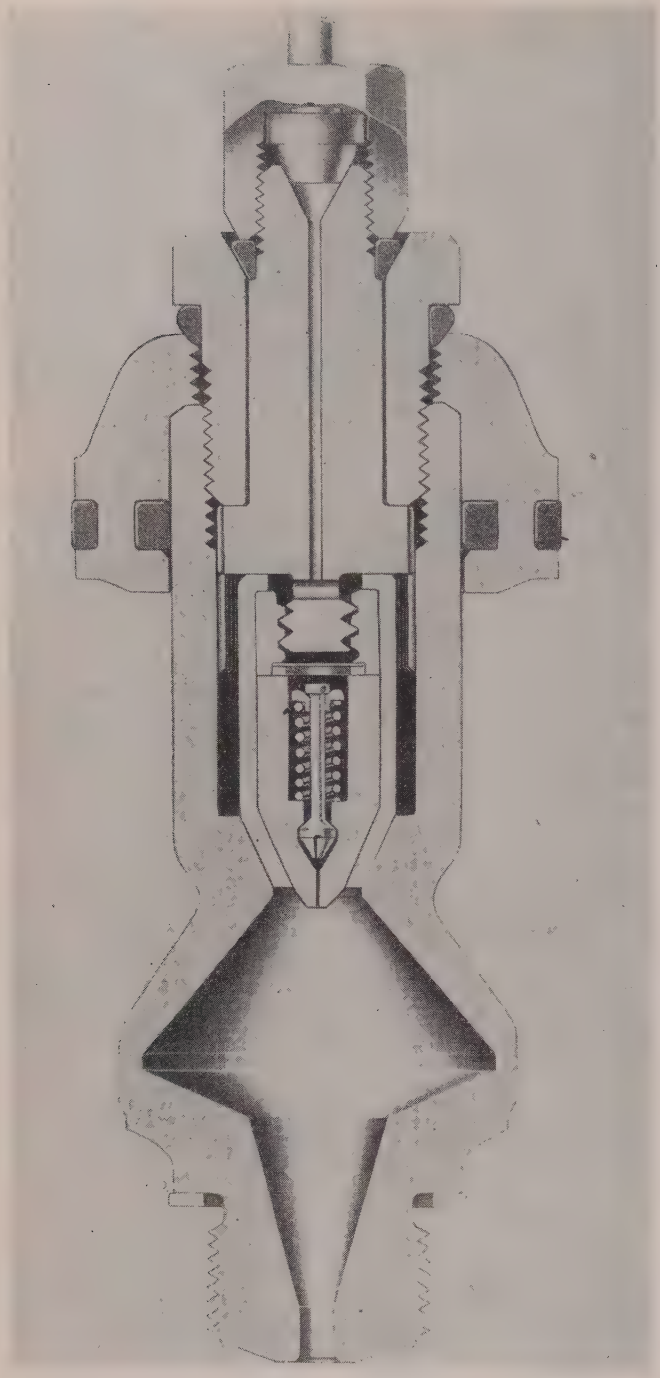


Fig. 22 – Cross-section of simplified fuel nozzle

At F the latest fuel valve is illustrated. The fuel-valve assembly is built into a cartridge package and the fuel-valve needle is an outward opening, or open-nozzle type. Other modifications have been made to improve the sealing of the chamber in the cylinder head. Further details of this valve are more clearly shown in Fig. 22.[12]

To achieve higher-output engines, effort must be expended toward considerable improvement in pistons and rings. The development of suitable materials which will function under the heat flows experienced at high output and high speeds, must be supplemented with improved knowledge on combustion. There is still a vast field for exploration in proper loading and scavenging of air to improve the charge to the cylinder, and to accomplish proper mixing of fuel and air to promote the rapidity of combustion necessary for efficient burning. The two developments must go hand in hand.

Above all, there is a great need for simplicity of design, as a general rule, in all powerplant projects. There have been too many gadgets and extraneous accessories accumulated, which are of no essential value in improving the thermodynamic performance of the powerplant. The friction horsepower has been moving upward because of accessories and gadgetry which do not contribute to the ultimate economy of operation.

Future powerplants must be concerned with improvement of cyclic efficiency, and proper allocation of temperature levels in relation to the capacity of parts to furnish adequate heat transfer.

Avenues open for further improvement of the combustion engine, combining higher output and higher speed with reasonable maintenance, include:

1. Control of the flame front of the gas-injected – or otherwise introduced – combustible, to prevent destructive concentration of heat flow through metals.
2. Equalization of heat transfer through the main combustion-chamber envelope to relieve cylinder-head and piston structures of undue mechanical and thermal stresses.
3. Development of adequately matched and efficient superchargers.
4. Continued improvement of materials and lubricants.
5. Provision for controlled thermal levels throughout the geometry of the engine in order to stratify temperature ranges for maximum efficiency and effective cooling.
6. Provide overall economy by the utilization of economically available fuels.

ORAL DISCUSSION

F. W. Fingerle, Hiller Helicopters: When using an exhaust supercharger, was the smoke excessive when accelerating the engine?

Mr. Rosen: No. This problem was overcome by matching the exhaust supercharger to the engine. Quicker supercharger response was obtained by cutting down the inertia of the parts and the friction of the bearings in the supercharger.

E. W. Templin, Timken-Detroit Axle Co.: The goal for truck speed upgrade should be 45 mph instead of the 25 or 30 mph proposed. This could be accomplished by using a 200-hp engine for 95% of the mileage and using an auxiliary engine of 600 hp when going upgrade.

Today's Aluminum Aircraft Alloys

J. A. Nock, Jr., Aluminum Co. of America

This paper was presented at SAE National Aeronautic Meeting, Los Angeles, Oct. 3, 1952.

ANY reference to aircraft will immediately suggest high-strength aluminum alloys to those familiar with the design and construction of a modern airplane. One may say that modern aircraft are largely the product of "today's aluminum aircraft alloys." The alloys used structurally in the aircraft industry are principally 14S, 24S, and 75S, although other alloys are used in certain applications. New structural and other alloys have been presented at opportune times – the latest commercial alloy to be added to the high-strength branch of the aircraft alloy family being Alcoa 75S. Although today's alloys are satisfactory, attention is naturally focused on the future and new materials. Research to develop new alloys, new thermal treatments, improved fabricating procedures, and a clearer understanding of the various problems associated with aluminum alloys is constantly in progress. In this paper, attention will be given to current developments in the aircraft field, and research progress as related to aircraft and other alloys.

Alloys in Use Today

Aluminum alloys are the keystone of aircraft structures and "today's aluminum aircraft alloys" are predominantly high-strength heat-treatable alloys. Alloys from a number of basic alloy systems are used to a limited extent, but two of the three most widely used structural alloys (14S and 24S) belong to the well-known aluminum-copper-magnesium system[1] containing additions of silicon and manganese, and the third (75S) belongs to the aluminum-zinc-magnesium-copper[2] group. The in-

[1] See AIME Transactions, Vol. 99, 1932, pp. 119-131: "Equilibrium Relations in Aluminum-Copper-Magnesium and Aluminum-Copper-Magnesium Silicide Alloys of High Purity," by E. H. Dix, Jr., G. F. Sager, and B. P. Sager.

[2] U. S. Patent No. 2,240,940, May 6, 1941.

INASMUCH as new developments in the field of aluminum alloys are immediately reflected in the aircraft industry, the author of this paper reviews past and present achievements which relate directly to aviation needs of the future. The alloys in use today are described as satisfactory. They have been brought to this point by intensive research leading to high strength, resistance to corrosion and fatigue, and ease of fabrication.

A clearer understanding of the problems associated with present alloys is responsible for the presentation of successful new alloys. The author describes how a recent newcomer, 75S, has already attained dominance in the extrusion field, and has made measurable inroads in the sheet, plate, and forging fields.

He also describes how military demands for alloys of higher and higher strengths have led to a complete revision of concepts concerning elongation, followed by a reappraisal of the need for artificial aging.

The Author

J. A. NOCK, JR., is assistant chief of the Physical Metallurgy Division of the Aluminum Research Laboratories, Aluminum Co. of America. His work has been mainly on the development of new aluminum alloys and of new fabricating practices and thermal treatments of wrought alloys. He joined Alcoa in 1923, after obtaining an electrochemical engineering degree from Pennsylvania State College.

Table 1 – Composition of Alloys

Alloy	Cu	Si	Mn	Mg	Zn	Cr
14S	4.4	0.9	0.8	0.4	—	—
24S	4.5	—	0.6	1.5	—	—
75S	1.6	—	—	2.5	5.6	.3

Table 2 – Amounts of Alloys in Aircraft

Alloy	Sheet-Plate[a]	Extrusions	Rod-Bar	Forgings
14S	25	20	—	90
24S	55	30	90	—
75S	20	50	7	10

[a] Alclad products included.

troduction of 75S alloy in 1943[3] was the culmination of several years of intensive research work on this alloy, to say nothing of the work carried out on related alloys before this period.

The nominal compositions of the three major aircraft alloys are given in Table 1.

The relative amounts of these three major alloys used in the structure of today's aircraft is of interest. This is given for sheet and plate, extrusions, rod and bar, and forgings produced by Alcoa in 1950. (See Table 2.)

The above figures tell an interesting story of the progress made by the newcomer in this field. In a relatively short period 75S alloy has attained dominance in the extrusion field, and has made measurable inroads in the sheet and plate, and forging fields.

At the present time, there is some trend toward the use of more 75S forgings, and the use of 75S extrusions is slowly increasing. The latter product is finding favor because of the greater uniformity of properties, and the greater freedom from warpage and distortion during machining operations than is shown by 24S-T4. Recent developments in the sheet field may tend to increase the use of 75S sheet and plate, as will be discussed later.

Appraisal of Trends in Aircraft Field – The use of aluminum alloys in aircraft structures in the 1920 and 1930 decades was almost entirely confined to naturally aged materials (17S-T4 and 24S-T4). One exception of consequence was the use of 14S-T6[4] forgings in the 1930 decade. Artificial aging was not a new process, but the high yield strength/tensile strength ratio and concurrent low elongations were not acceptable for sheet. The

[3] See AIME Transactions, Vol. 35, 1945, pp. 130-155: "New Developments in High-Strength Aluminum Alloy Products," by E. H. Dix, Jr.

[4] See AIME Transactions, Vol. 71, 1925, pp. 828-863: "New Developments in High-Strength Aluminum Alloys," by R. S. Archer and Zay Jeffries.

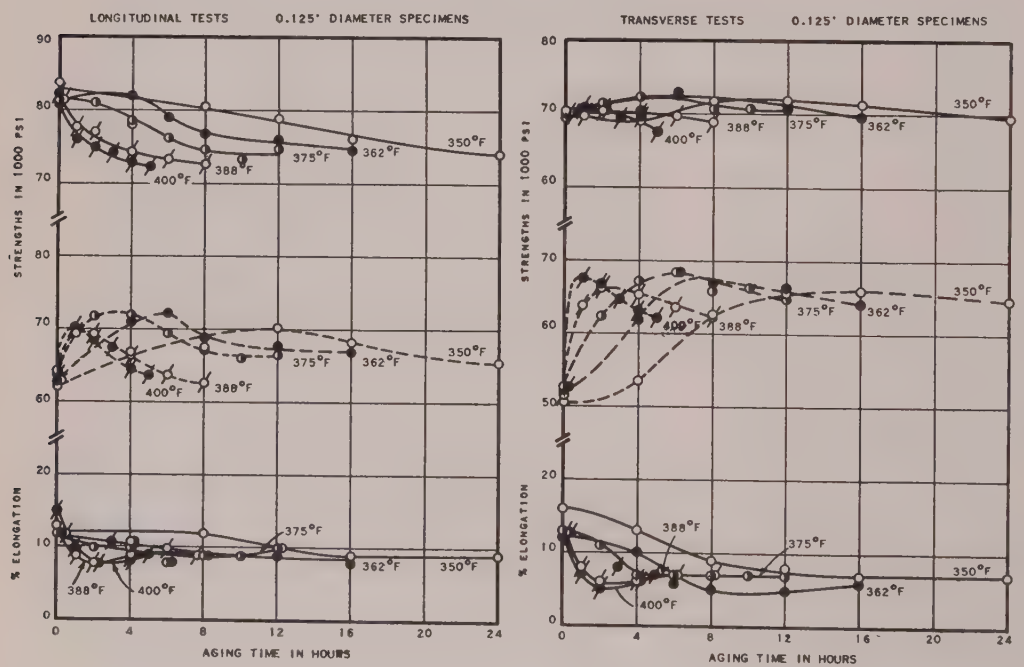


Fig. 1 – Aging of 24S-T4 extrusions at 350 to 400 F – section approximately 1.1 by 2.1 in.

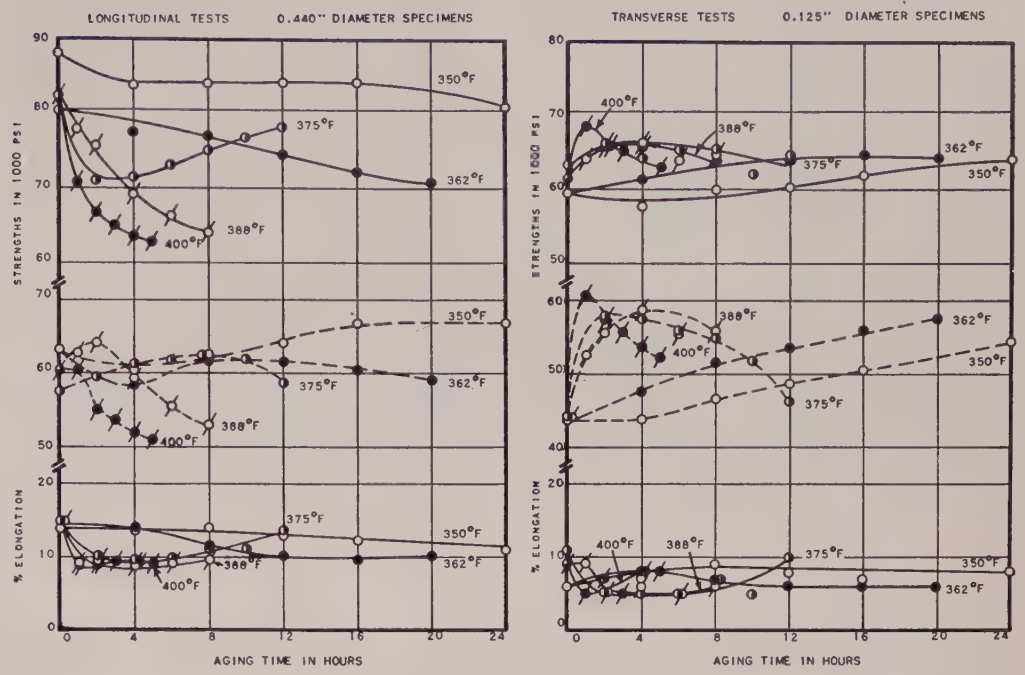


Fig. 2 – Aging of 24S-T4 extrusions at 350 to 400 F – section approximately 3.5 by 3.5 in.

higher strengths of 24S[5] were welcomed in the 1930 period, and strain-hardening to the -T36 temper was used to a limited extent with rigid control upon elongation. Artificial aging was proposed, but the elongation requirements of the aircraft manufacturers at that time made it impossible to carry the aging process far enough to obtain maximum strength or satisfactory resistance to corrosion. The properties specified permitted aging only to the degree that resistance to corrosion was seriously lowered because of highly selective precipitation at grain boundaries.

The advent of World War II increased the demand for higher strength, and a decided change in the thinking of aircraft engineers concerning elongation requirements took place. This action came from the realization that higher strengths for the known alloys and new alloys would of necessity require artificial aging. A review of the data accumulated during the earlier studies of the artificial aging of 24S sheet, when aged material was suggested for dirigible structures, led to a reappraisal of the resistance to corrosion in view of the desirable increase in strength that was immediately available. This was prompted by the realization from other studies that aging first decreased and then increased resistance to corrosion as the aging period was prolonged. In addition, new techniques for evaluating resistance to corrosion, particularly the effect of an externally applied stress, had been developed for sheet specimens during the intervening years. This appraisal soon indicated that by simply extending the aging period, at temperatures of about 375 F, to develop maximum strength, or even a slight degree of overaging, good resistance to corrosion was secured. Strain-hardening accelerated aging and increased the maximum strength attainable in general confirmation of work by Meissner.[6] He recommended combinations of working and aging at temperatures that were too low to develop good resistance to corrosion. By a fortunate metallurgical coincidence, maximum strength and resistance to corrosion of 24S and 75S sheet are attained simultaneously.

The artificially aged products of 75S and 14S are now commonly used in aircraft construction, and some artificially aged 24S sheet has been utilized. The aging of 24S extrusions[3] has received some attention, but little commercial progress has been made. Data collected during recent years present an interesting picture, particularly from the viewpoint of resistance to corrosion.

The curves in Figs. 1 and 2 illustrate the aging at 350 F to 400 F of two sections of commercial 24S-T4 extrusions of rather widely varying cross-section, namely, 1.1 x 2.1 in. and approximately

[5] See "Alcoa Aluminum and Its Alloys." Published by Aluminum Co. of America, Pittsburgh, 1932.

[6] See *Zeitschrift für Metallkunde*, Vol. 24, April, 1932, pp. 88-89: "Influence of Cold Rolling on Age-Hardening of Duralumin," by K. L. Meissner.

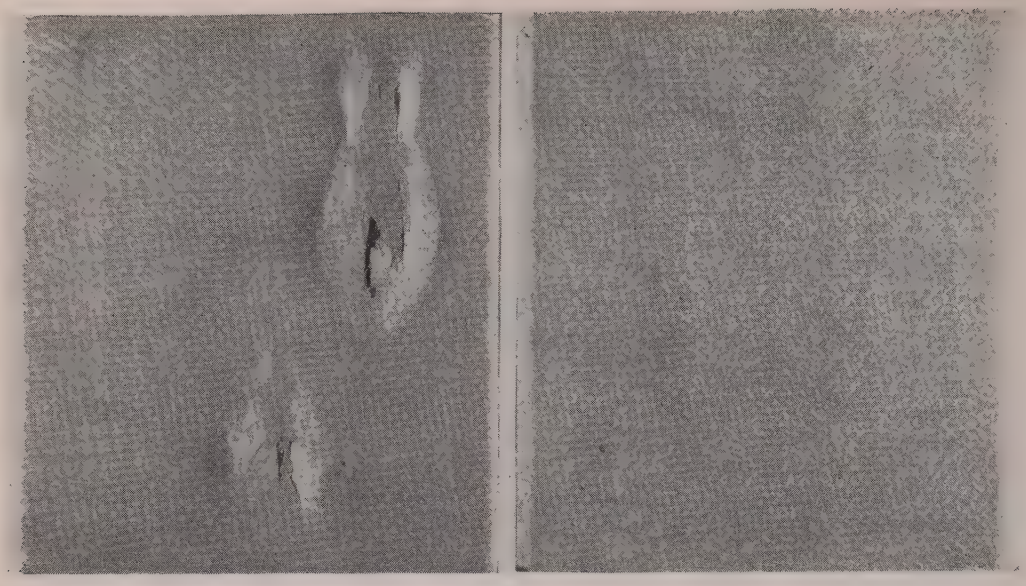


Fig. 3 – Extruded channel exposed at Point Judith, R. I. Left: 24S-T4 exposed 4 yr; right: 24S-T6 aged 11 hr at 375 F, exposed 5 yr

3.5 in. square for the body of the section. In Fig. 1, it is evident that the tensile strength in the longitudinal direction is lowered by aging, while a slight increase is secured transversely. The yield strength is increased about 10% longitudinally and about 30% transversely, while the elongation is lowered 50-60%. The larger section, as shown in Fig. 2, suffers a measurable decrease in strength longitudinally, but transversely the tensile strength and the yield strength increase by about the same amount noted for the smaller section. These data indicate an anistropic response to aging for 24S extrusions, but a very useful set of properties is obtained.

The resistance to corrosion of these sections, as evaluated by exposing stressed and unstressed specimens of 0.125- and 0.44-in. diameter in the $NaCl$-H_2O_2 solution for 1 or 7 days respectively by total immersion, reveals that losses in tensile strength in the longitudinal direction were cut in half by adequate aging, with no change in elongation losses and no acceleration of losses by stress. In the transverse direction, tensile strength losses (unstressed) of the smaller section were reduced 60% by the aging, and the effect of stress reduced so that the losses under stress were only slightly more than those for unstressed specimens. The same general effect was secured for the larger section, but the improvement by aging in the transverse direction was more striking, as some of the 0.125-in. diameter naturally aged specimens failed under stress, while losses in tensile strength of the 0.125-in. diameter 24S-T6 specimens exposed under stress were of the order of 25% after proper aging at 360 to 385 F.

Additional data from extended atmospheric exposures of 24S extruded sections aged 11 hr at 375 F confirm the accelerated test results. Fig. 3 shows sections taken from an extruded channel, about 9 in. wide, having a web thickness of 0.375 in., in the -T4 and -T6 tempers, after an exposure of 4 to 5 years in the seacoast atmosphere at Pt. Judith, R. I. Severe local exfoliation has occurred on the web of the -T4 sample, while only mild general surface attack will be noted on the -T6 sample. Exfoliation attack often occurs in 24S-T4 and 14S-T4 extrusions in both accelerated and extended atmospheric exposures. Aging at 365-385 F has consistently produced an appreciable improvement in the resistance to corrosion and stress corrosion of large sections of 24S – extrusions, rolled rod and bar, and plate.

It is believed that more consideration should be given to the use of 24S-T6 extrusions. It is planned to evaluate the effect of strain-hardening before aging upon the tensile properties and resistance to corrosion. This situation is recognized for 24S sheet where strain-hardening by rolling or stretching accelerates the aging, and results in higher strength and somewhat better resistance to corrosion. It is proposed to stretch the extrusions controlled amounts after quenching and then age them. Some predictions can be made from the behavior of 24S-T4 extrusions of present commercial production in the longitudinal direction. Long experience indicates that the first sign of overaging is a decrease in tensile strength. This occurs in the aging of 24S-T4 extrusions in many instances. Since some strain-hardening from straightening is present, and strain-hardening accelerates the aging of 24S sheet (-T36 to -T86), it is conceivable that additional strain-hardening may produce even more rapid lowering of the longitudinal tensile strength. It should be noted that an incompletely recrystallized, highly oriented structure prevails in heavy extrusions. This probably accounts for some of the

directional effects in the aging results just discussed.

Increased Importance of Fatigue – An increase in strength of aluminum alloys produced by changes in composition or by changes in thermal treatment of established alloys is not usually accompanied by a proportionate increase in endurance limit. Although the rotating-beam fatigue strength of smooth longitudinal specimens of 75S-T6 products is somewhat higher than similar products of 14S-T4 and -T6 and 24S-T4, notch effects cause a greater lowering of fatigue strength than for the other materials.[7] The actual endurance values for notched specimens are in general about the same for 14S-T6 and 75S-T6.

The role of fatigue may not be of extreme importance in military aircraft, but the long life expected from commercial airplanes highlights the importance of this property. Fatigue cracks have been encountered after long periods of service, and in view of replacement costs, selection of the proper material and particular attention to improved design assume greater importance. Where fatigue is felt to be the critical factor, the design may well center about this property rather than static tensile properties.

In early production of 75S, some very coarse chromium constituent particles[8] were found in 75S extrusions, but with current methods of production the occurrence of such large particles is practically eliminated. The data from tensile tests indicate that the coarse chromium constituent particles had little if any influence on strength; that is, they behaved like constituent particles of normal size. It was felt in some quarters that the effect of these particles on fatigue strength would be harmful. Tests of thin extruded sections which showed the presence of numerous particles of chromium constituent, Fig. 4, indicate that the path of fracture had no relation to the location of these particles. The reproduction of a radiograph of a flexural fatigue specimen cut from a thin 75S-T6 extrusion is shown in Fig. 4. The path of failure did not go through the particles of chromium constituent, indicating that the presence of these particles did not initiate the crack nor adversely influence the fatigue strength.

High Temperatures in Structures – The higher speeds of new aircraft and guided missiles bring the problem of high temperatures into the design of structural members. Thermal de-icing also involves high-temperature use of structural materials. The properties of the three structural alloys at elevated temperatures[9] indicate that 75S-T6 shows a much greater decrease in strength at temperatures above 250 F, than either 14S-T6 or 24S-T4. The selection of an alloy from the three major aircraft alloys for use above this temperature would fall between 14S and 24S in an artificially aged temper.[10]

In addition to higher initial strength which will be available in artificially aged 24S products, the use of these materials rather than naturally aged materials is strongly recommended for other reasons. Thus, the measurable decrease in strength resulting from the initial exposure of the naturally aged material to elevated temperatures (resolution effects) will be avoided. This low point in strength is then followed by a period of increasing strength (artificial aging) at the temperature in question, until the maximum level of strength is attained. These changes are illustrated by the aging curves in Fig. 5. This period can be extremely long at temperatures in the low range (200-250 F) as illustrated in Fig. 6 for aging at 212 F. Another reason for avoiding the use of naturally aged material for parts which will operate at an elevated temperature, is that resistance to corrosion will be lowered. The initial exposure to elevated temperatures results in a decrease in resistance to corrosion of 24S products. The magnitude of this decrease and the duration are determined by the previous fabricating history, the temperature, and the time of exposure. Eventually, maximum strength and associated satisfactory resistance to corrosion will be attained, but at 212 F this has not been accomplished in a period of 10 yr as illustrated in Fig. 7. The maximum decrease in resistance to corrosion was reached after about 3 yr at 212 F, but at higher temperatures this condition would be attained in much shorter periods. Unstable tensile properties and variable resistance to corro-

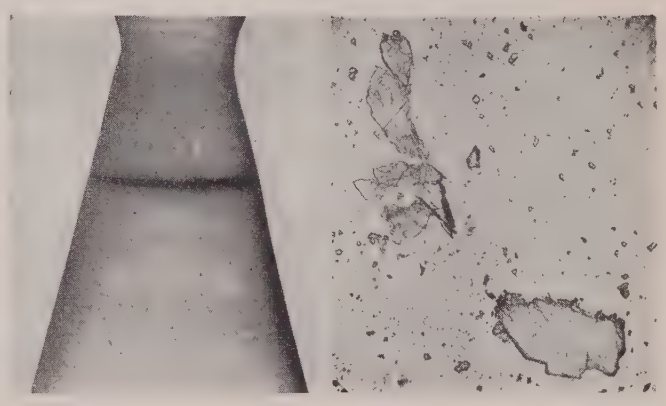


Fig. 4 – Left: reproduction of radiograph of flexural fatigue specimen of 75S–T6 extrusion. Chromium constituent is present as white spots which are absent along path of failure. Microstructure of sample is shown at right (reduced from microphotograph taken at 100X). Particles of chromium constituent are representative of larger ones present.

[7] See *Aviation Week*, Vol. 51, Oct. 10, 1949, pp. 21-38: "How to Use High-Strength Aluminum Alloy," by E. C. Hartmann, F. M. Howell, and R. L. Templin.

[8] See *Iron Age*, Vol. 156, Oct. 4, 1945, pp. 64-68: "Metallography of Alcoa 75S Alloy," by F. Keller.

[9] See "Alcoa Aluminum and Its Alloys," pp. 119-120. Published by Aluminum Co. of America, Pittsburgh, 1950.

[10] See *Iron Age*, Vol. 160, Dec. 25, 1947, pp. 73-79: "Reheating of 24S and 75S Aluminum Alloy Sheet," by J. A. Nock, Jr.

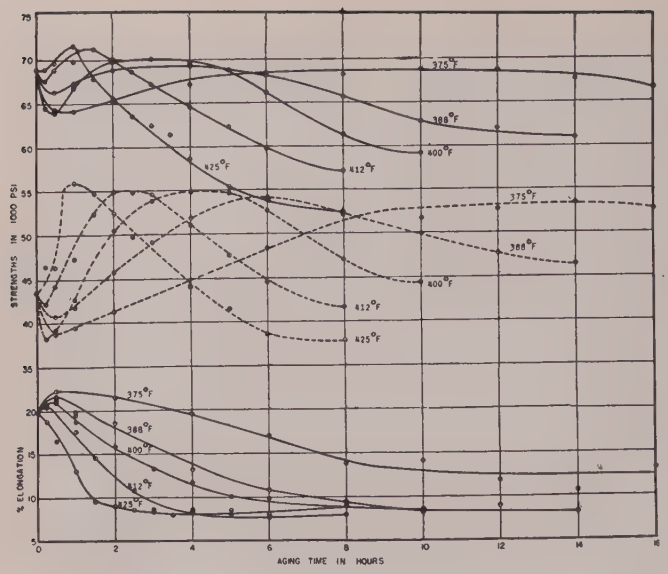


Fig. 5 – Effect of aging temperature and time on properties of 24S-T4 sheet. Tensile strengths (solid lines); yield strengths (broken lines); x-grain specimens from sheet 0.064-in. thick

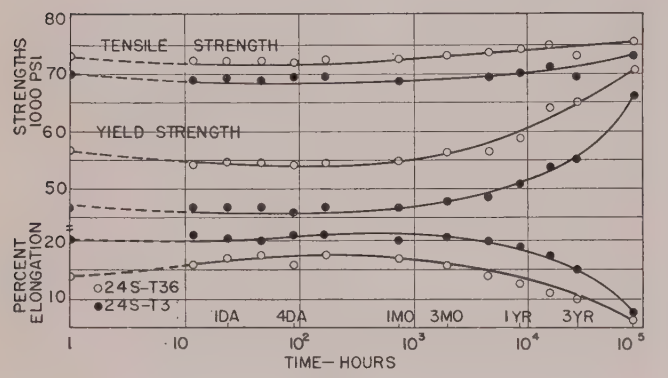


Fig. 6 – Effect of heating at 212 F upon mechanical properties of 24S-T3 and -T36 sheet

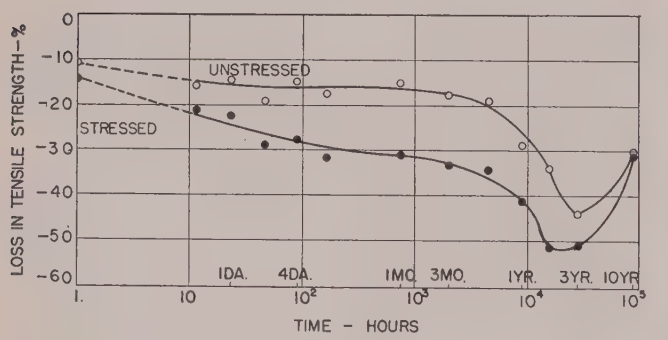


Fig. 7 – Effect of heating for 10 years in oil bath at 212 F upon resistance to corrosion of 24S-T6 0.064-in. thick sheet. After heating as indicated, tensile specimens were removed and exposed for 48 hr in NaCl-H_2O_2 solution by alternate immersion, under stresses of 0 and 75% of yield strength, to evaluate resistance to corrosion. Losses were calculated by comparing properties of exposed specimens with properties secured prior to exposure

sion are potent reasons, indicating the advisability of using artificially aged materials for structures exposed to elevated temperatures. Later in this paper new alloys for high temperature use will be discussed.

Directional Properties – Forgings and Extrusions

Tensile Tests in Different Directions – The properties of aluminum alloy forgings and extrusions show considerable directionality, which is in part a reflection of the preferred orientation resulting from the directional nature of most fabricating processes, and of other factors. Among the latter are composition effects including impurities, ingot structure, amount of reduction from ingot, thermal treatment, and quenching. These factors are reflected in the minimum mechanical properties of 14S-T6 and 75S-T6 hand forgings as given in published data.[11]

These data cover tests in the three principal directions of hand forgings ranging in cross-sectional area from 36 to 144 sq in. In any direction a decrease in strength and elongation will be noted as the area increases. It is known that this is associated with quenching rate and the degree of reduction. The quenching rate[12] decreases as the size of the forging increases, and total reduction from the ingot is usually less for the very large forgings than for the smaller forgings. With test direction, longitudinal versus transverse, an additional decrease in strength and a very sharp lowering of elongation will be noted in the transverse direction. This decrease in strength reaches a maximum of 11,000 psi for both factors – size and direction. The elongation decrease is of the order of ⅓ to ½ for the increase in area from 16 to 144 sq in. for 14S-T6 in all directions, and for 75S-T6 in the longitudinal and long transverse directions. In the short transverse direction, the elongation of 75S-T6 forgings tends to be somewhat lower than in the other directions. Directional effects indicate that long transverse elongations are about half as great as longitudinal values, and short transverse values are about half as much as the long transverse values.

Every effort is being made to improve the transverse elongations of large hand forgings and other products, and definite progress is being made. However, careful attention to the design and fabrication of each part must be maintained to avoid high stresses in the least favorable direction. In particular, the machining of smaller sections or parts from large extrusions and forgings should be entirely avoided in some instances or carried out only after a careful analysis of each situation.

---

[11] See "Hand Forgings." Published by Aluminum Co. of America, Pittsburgh, 1952.

[12] See *Metals Technology*, Vol. 14, August, 1947, AIME Technical Publication No. 2225, 13 pages: "Quenching of 75S Aluminum Alloy," by W. L. Fink and L. A. Willey.

Corrosion Tests

The effect of direction upon the tensile properties of large sections of the structural alloys is accompanied by other variations associated with directional effects. Evaluation of the resistance to corrosion of structural alloy plate, extrusions and forgings, has established a definite difference in the behavior of longitudinal and transverse specimens in accelerated corrosion tests and on atmospheric exposure. To obtain this evaluation, machined tensile test specimens were exposed both without stress and stressed in tension to 75% of the yield strength. After various exposure periods, the strength and elongation of the exposed specimens were determined and the losses computed by comparing these properties with the original properties. The losses in properties of unstressed specimens in the longitudinal direction were considered to be satisfactory for the several products of the structural alloys. The effect of the constantly applied stress of 75% of the yield strength on specimens taken in this direction was negligible. In the short transverse direction, however, the losses for unstressed specimens were considerably higher than those for longitudinal specimens. In this direction, the applied stress caused an appreciable increase in losses and in some instances failure before the end of the test period.

It should be recognized that the constantly applied stress used in the routine laboratory tests, 75% of the yield strength, is extremely high and much above any stresses normally encountered in aircraft structures. In the use of the various products of the structural alloys in aircraft structures, there have been very few recognizable cases of actual stress corrosion cracking. Locked-up stresses resulting from poor design or unsatisfactory manufacturing practices may attain unexpected levels and should be avoided or kept as low as possible.

Experience with products of 14S in the -T6 temper has been very satisfactory. Laboratory evaluation of 14S and 75S products shows that the two alloys are generally similar in behavior. This would indicate that there should be no hesitancy in using either alloy. However, locked-up stresses, always undesirable, should be avoided by careful design and good manufacturing procedures.

New Developments in Aluminum Alloys

New High-Strength Structural Alloy – Alcoa XA78S – In the realm of new developments in aluminum alloys, attention is first directed to XA78S which was developed specifically for aircraft use where greater strength than that of 75S is desired. This alloy, requested by the Bureau of Aeronautics and the Air Force, was introduced experimentally about a year ago. It may be most aptly described as a higher strength 75S-type alloy. The new alloy develops strength about 10% higher than 75S, accompanied by some slight decrease in formability and resistance to stress corrosion cracking. In the development of XA78S, many other compositions were studied, including alloys that would produce appreciably higher strength, but XA78S was selected as the most desirable compromise on the basis of fabricating characteristics, strength, resistance to corrosion, and resistance to stress corrosion cracking. The properties of XA78S products in comparison with the properties of similar 75S products are given in Table 3. The properties in the annealed temper have been purposely omitted since they are the same for both alloys.

Table 3 – Alcoa 75S and XA78S – Comparison of Specification Properties

Alloy	Thickness, in.	Tensile Strength, psi	Yield Strength, psi	Elongation in 2 in., %
		Sheet		
75S	0.016–0.039	76,000	65,000	7
XA78S	0.016–0.040	83,000	72,000	7
75S	0.040–0.499	77,000	66,000	8
XA78S	0.041–0.499	84,000	73,000	8
		Alclad Sheet		
75S	0.016–0.039	70,000	60,000	7
XA78S	0.016–0.040	76,000	66,000	7
75S	0.040–0.499	72,000	62,000	8
XA78S	0.041–0.499	78,000	68,000	8
		Extrusions		
75S	To 0.249	78,000	70,000	7
XA78S	To 0.249	84,000	76,000	5
75S	0.250–2.999	80,000	72,000	7
XA78S	0.250–2.999	86,000	78,000	5

The evaluation of sheet, alclad sheet, and extrusions of the new alloy by various aircraft companies is in progress. The reaction of the aircraft industry has been very conservative, and logically so, as the utilization of 75S to the maximum degree has not been accomplished, and the advantage in strength secured by using XA78S is relatively small. To date, the evaluation confirms that of experimental material, and no trouble is anticipated in the use of Alclad XA78S-T6 products. Some products are slightly less corrosion-resistant than similar products of 75S; but on the whole, the results are encouraging. It is felt that with careful design to eliminate any unusual stress concentrations, XA78S should perform in a satisfactory manner.

Coating Thickness on Alclad 75S-T6 Sheet and Plate – In the production of alclad sheet of the structural alloys, the cladding thickness is fixed at certain percentages of the total thickness. This method provides adequate electrolytic protection for the thinner gages, but generally the cladding is more than necessary when the thicker gages are considered, and adds unnecessary weight. This situation has been recognized in the case of Alclad 14S and Alclad 24S sheet products. Thus, in the case of the latter, the cladding is reduced from a nominal 5% on each surface for a sheet 0.063 in. or less in thickness, to 2.5% on each surface for a sheet 0.064 in. or more in thickness. The cladding on Alclad 75S sheet was formerly fixed at 4% on each surface for all thicknesses.

Table 4 – Comparison of Minimum Properties

Thickness, in.	Tensile Strength, psi	Yield Strength, psi	Elongation in 2 in., %
0.188 to 0.50	75,000	64,000	8.0
0.040 to 0.187	72,000	62,000	8.0

Table 5 – Nominal Compositions

Alcoa Alloy	British Alloy	Cu	Fe	Mg	Mn	Ni	Ti
XF18S	RR58	2.20	1.10	1.60	—	1.05	0.07
XA19S	RR57	6.00	—	—	0.25	—	0.15

Recent developments in the methods of applying thin coatings now make it possible to offer Alclad 75S sheet and plate in a range of thicknesses from 0.188 in. to 0.50 in. with a coating thickness of the order of 1.5% on each surface. The new material will develop minimum properties which will be advantageous, as shown in the comparison in Table 4.

The reduced thickness of cladding will provide ample electrolytic protection to the 75S-T6, remove the weight penalty of excess cladding, and result in a useful increase in strength. The new material will not be available in all widths and lengths currently available for the Alclad 75S sheet which has the thicker coating.

High-Shear-Strength Rivets – Attention has been directed to the high shearing strength, 49,000 psi, of 75S-T6 products with the idea of utilizing this property in aircraft rivets. Currently, 24S rivets are employed where the maximum shear strength is required, with a typical value of 42,000 psi being attained from 24S-T31 rivets that are driven immediately after quenching or after limited periods of refrigeration.

Rivets of 75S alloy in aircraft sizes will develop shearing strengths of the order of 54,000 psi when heat-treated at 860-920 F and aged 24 hr at 250 F. Driving at room temperature cannot be accomplished without severe shear cracking of the driven heads. Aging by the 2-step treatment of 3 hr at 250 F plus 3 hr at 325 F results in shearing strengths of about 52,000 psi, and satisfactory room temperature driving characteristics. While the shearing strength is very interesting, the driven rivets, aged 3 hr at 250 F plus 3 hr at 325 F, show some degree of susceptibility to stress corrosion cracking, and are not recommended for general use. This problem has received the attention of the Aluminum Research Laboratories and others for somewhat different reasons. Some investigations have concentrated on developing the utmost in shearing strength and overcoming the cracking troubles of driving at room temperature, by driving warm at temperatures about 300 F. This method shows promise of eliminating the head-cracking during driving, but the susceptibility to stress corrosion cracking is quite pronounced. Attempts to develop rivets with satisfactory resistance to stress corrosion cracking have been successful, but at a sacrifice in shearing strength. Rivets aged by a step-treatment are practically immune to stress corrosion cracking, and will develop a shear strength in riveted joints of about 48,000 psi. This shear strength is an improvement of about 15% over 24S-T31 rivets, which develop a typical shear strength of 42,000 psi. This increase, coupled with the fact that the 75S alloy rivets would be received in the -T6 temper, thus eliminating the necessity of solution heat-treating immediately before driving or refrigerating after quenching to preserve the driving characteristics, may be of interest to the aircraft industry.

Table 6 – Comparison of Tensile Properties

	Tensile Strength, psi	Yield Strength, psi	Elongation in 2 in., %
Tensile Properties at Room Temperature			
XF18S-T61	63,000	52,000	11
XA19S-T6	61,000	43,000	13
B18S-T61	60,000	41,000	15
18S-T61	61,000	46,000	12
14S-T6	70,000	60,000	13
Tensile Properties at 400 F after 1000 Hr			
XF18S-T61	34,000	31,000	24
XA19S-T6	39,000	29,000	18
B18S-T61	26,000	20,000	25
18S-T61	25,000	19,000	18
14S-T6	22,000	17,000	31
Tensile Properties at 600 F after 1000 Hr			
XF18S-T61	10,000	6000	70
XA19S-T6	14,000	11,000	28
B18S-T61	6000	3500	75
18S-T61	6000	4000	50
14S-T6	7000	5500	64

Alloys for High-Temperature Applications – XF18S and XA19S: As noted in an earlier section of this paper, certain aluminum alloys possess the ability to maintain a relatively high level of strength at elevated temperatures. Alloys such as 14S, 24S, 18S, and 32S are wrought alloys of this class. With the advent of jet engines, elevated temperature strength is one of greater concern than in the days of piston engines. Licenses for the construction of several jet engines of British design have been secured by American concerns. Aluminum alloy parts in these engines were specified in British alloys RR57 and RR58. These alloys are being produced by Alcoa under the designations XA19S and XF18S. The nominal compositions are given in Table 5.

In general, parts of XF18S are recommended for

Table 7 – Aluminum Alloys for High-Temperature Use – Creep Properties

At 400 F	Stress (psi) for Minimum Creep Rate of				Stress (psi) for Rupture in		
	0.000001 In./In./Hr	0.00001 In./In./Hr	0.0001 In./In./Hr	0.001 In./In./Hr	10 Hr	100 Hr	1000 Hr
				400 F			
18S-T61	19,000	24,000	30,000	39,000	36,000	27,000	19,000
B18S-T61	16,000	22,000	31,000	43,000	37,000	25,000	17,000
XF18S-T61	23,000	25,000	28,000	32,000	31,000	28,000	24,000
XA19S-T6	.....	18,000	21,000	26,000	26,000	21,000	17,000
APMP[a]	15,000	16,000	17,000	18,000	18,000	17,000	16,000

At 600 F	Stress (psi) for Minimum Creep Rate of				Stress (psi) for Rupture in			
	0.00001 In./In./Hr	0.0001 In./In./Hr	0.001 In./In./Hr	0.01 In./In./Hr	1 Hr	10 Hr	100 Hr	1000 Hr
XF18S-T61	.....	.....	6400	8000	9600	7100	.....	.....
XA19S-T6	.....	7000	8600	11,000	12,000	9000	7500	.....
APMP[a]	10,500	11,500	13,000	13,000	.....	13,000	11,500	10,000

[a] Aluminum powder metallurgy product.

use at temperatures as high as 475 F, while XA19S is suggested for temperatures of the 600 F level. The room and elevated temperature tensile properties of these alloys are given in Table 6, with 18S and B18S data for comparison.

The strength of the new alloys is somewhat higher than that of the standard alloys at the temperatures mentioned. Alloy XF18S possesses excellent creep resistance at 400 F, while XA19S is somewhat better at 600 F in this respect, as will be noted in Table 7. Fatigue properties in Table 8 show that XF18S-T61 is superior to the other alloys at 400 F, while at 600 F B18S-T61 appears to head the field.

Other alloys are being studied in the general effort to develop new alloys that will have properties superior to those of alloys now available for use in the elevated temperature field. Several experimental alloys show promise. One alloy appears to have desirable characteristics for use in the lower range of temperatures up to about 500 F, while a second alloy develops higher tensile properties than all currently available alloys for use at temperatures of the order of 600 F. Considerable work remains to be carried out in order to more accurately define composition limits and other factors, before these alloys may be considered satisfactory for commercial exploitation.

Aluminum powder metallurgy products: In the high-temperature field, a new development of unusual interest has been under investigation by Alcoa for several years. A new process, of Swiss origin, is described in Swiss Patents 250118 and 259878 for producing articles of aluminum powder which have exceptional properties at room temperature and elevated temperatures for pure aluminum. The inventor ascribes the high properties obtained in these articles to the influence of the particle size of the powder and the oxide film covering the particles of the powder.

At room temperature and elevated temperature the mechanical properties of aluminum powder metallurgy products (APMP) produced at the Aluminum Research Laboratories, but not according to the Swiss patents, are given in Table 9 and compared with several alloys normally used for high-temperature applications.

It is apparent that the strength of the latter alloys is much higher than that of the aluminum powder metallurgy products at room temperature, and at temperatures as high as 500 F some degree of superiority still exists for the regular alloys after short heating periods. However, after long periods at 400 F and 500 F and at higher temperatures, the new product has tensile properties far superior to any alloy now commercially available. An outstanding characteristic of the new product is the stability of strength as the period of exposure is increased at the elevated temperature.

In general, commercially available alloys show a continued decrease in strength as the time at temperature is increased, but data at 600 F, 800 F and 1000 F indicate no change in strength for the powder metallurgy product over a period of 1000 hr. Creep properties are shown in Table 7, with the indication that at 400 F most of the alloys generally used are somewhat superior to APMP. The data at 600 F show a definite advantage for the new prod-

Table 8 – Aluminum Alloys for Elevated-Temperature Use – Fatigue Properties

	Stress (psi) for Failure in			
	10^5 Cycles	10^6 Cycles	10^7 Cycles	10^8 Cycles
		400 F		
18S-T61	.....	21,000	14,500	9500
B18S-T61	.....	22,000	15,500	10,000
XF18S-T61	.....	22,000	17,000	13,000
XA19S-T6	.....	19,000	13,000	9500
APMP[a]	.....	11,500	9500	8500
		600 F		
18S-T61	13,000	10,000	7000	5000
B18S-T61	19,000	14,000	9500	6500
XF18S-T61	16,000	11,000	7500	5500
XA19S-T6	17,000	12,000	8000	5500
APMP[a]	12,000	9000	6500	6000

[a] Aluminum powder metallurgy product.

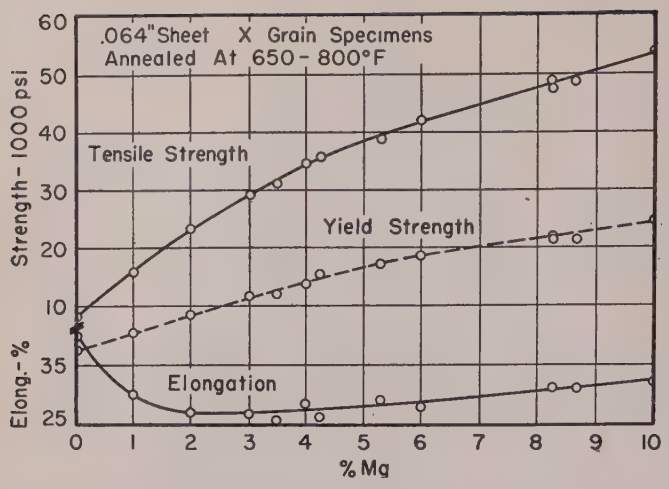


Fig. 8 – Effect of increasing the magnesium content upon strength of aluminum-magnesium alloy sheet

uct. Fatigue properties at 400 F and 600 F are given in Table 8, and these data indicate that most of the commercially used alloys for high-temperature applications are somewhat superior in fatigue to the new product. However, there is some indication that the latter may have superior fatigue properties at still higher temperatures, but data at these temperatures will be required to substantiate this indication.

Samples of APMP may be obtained for test purposes. It should be pointed out here that, while no patents have been obtained in this country corresponding to the Swiss patents, there may be pending applications dealing with modifications of the Swiss process. The product made experimentally by Alcoa differs somewhat from the samples produced abroad, in that the strength is somewhat lower and the elongation somewhat higher. Samples are being furnished primarily to determine where products of this type may be of importance in the defense effort. It should be noted that facilities are not available for quantity production of the new product.

Aluminum-Magnesium Alloys for Sheet and Plate – XA54S and X56S – The assembly of many articles by welding is common practice. This general practice is applicable to many items produced from certain aluminum alloys, but for nonheat-treatable alloys the strengths across the weld are relatively low. In other instances where heat-treatable aluminum alloy products are involved, the strengths are generally acceptable, but the ductility across the weld is much lower than that of the original component parts. The nonheat-treatable alloys 2S, 3S, and 52S are used in applications involving welding; while the aluminum-magnesium-silicon alloys, for example 61S, 62S, and 63S, represent the heat-treatable alloys most generally used in applications where welding is required. The heat of welding adversely affects both strain-hardened and heat-treated parts, but the strength of the latter is lowered to an undesirable and, in many cases, an unknown level if welding conditions are not carefully controlled. The maximum properties of welds in heat-treatable alloys can be developed only by solution heat-treatment of the weldment, and this is obviously impractical in most instances.

Aluminum and magnesium form a continuous series of solid-solution alloys with about 3% magnesium being soluble[13] at approximately 400 F; while almost 15% is soluble at the eutectic temperature of about 840 F. The properties of aluminum-magnesium alloys containing up to 10% magnesium

[13] See AIME Transactions, Vol. 83, 1929, pp. 351-372: "Equilibrium Relations in Aluminum-Magnesium Alloys of High Purity," by E. H. Dix, Jr., and F. Keller.

Table 9 – Properties of Aluminum Alloys and Aluminum Powder Metallurgy Product at Elevated Temperatures

Temperature of Test	Time,	APMP[a]			18S-T61			XF18S-T6			XA19S-T6		
		Tensile Strength, psi	Yield Strength, psi	Elongation in 2 in., %	Tensile Strength, psi	Yield Strength, psi	Elongation in 2 in., %	Tensile Strength, psi	Yield Strength, psi	Elongation in 2 in., %	Tensile Strength, psi	Yield Strength, psi	Elongation in 2 in., %
Room, F	.....	35,800	24,600	16	61,000	46,000	12	63,000	52,000	11	61,000	43,000	13
400	0.5	21,400	18,000	16	42,000	35,000	12	45,000	41,000	19	40,000	31,000	18
	100	.....	.....	..	37,000	31,000	15	36,000	32,000	24	39,000	29,000	18
	1000	.....	.....	..	25,000	19,000	18	34,000	31,000	24	38,000	27,000	21
	10,000	.....	.....	..	19,000	13,000	25	.....	.....	..	.....	.....	..
500	0.5	18,600	16,400	17	31,000	27,000	15	29,000	26,000	24	29,000	23,000	19
	100	.....	.....	..	18,000	14,000	25	21,000	17,000	35	28,000	21,000	19
	1000	.....	.....	..	14,000	9500	30	19,000	13,000	47	28,000	20,000	20
	10,000	.....	.....	..	10,000	6500	40	.....	.....	..	.....	.....	..
600	0.5	15,700	14,000	15	13,000	10,000	25	17,000	14,000	32	20,000	15,000	18
	100	16,900	15,000	13	9000	7000	40	11,000	8000	49	19,000	14,000	18
	1000	16,000	14,400	17	6000	4000	50	10,000	6000	70	14,000	11,000	28
	10,000	.....	.....	..	5500	3000	60	.....	.....	..	.....	.....	..
700	0.5	14,200	13,000	8	.....	.....	..	8500	6500	..	.....	.....	..
	100	.....	.....	..	.....	.....	..	5500	4500	..	.....	.....	..
800	0.5 to 1000	12,000	11,000	5	.....	.....	..	.....	.....	..	.....	.....	..
900	0.5 to 1000	10,800	10,500	4	.....	.....	..	.....	.....	..	.....	.....	..
1000	0.5 to 1000	7300	7200	2	.....	.....	..	.....	.....	..	.....	.....	..

[a] Aluminum powder metallurgy product.

Table 10 – Typical or Estimated Typical Properties of Commercial and Experimental Aluminum-Magnesium Alloy Sheet

Alloy	Mg	Cr	Mn	Tensile Strength, psi	Yield Strength, psi	Elongation in 2 in., %
			Annealed Temper, —O			
Al (99% Min)	....	....	....	13,000	5000	35
XA50S	0.80	....	....	17,000	6000	28
XE52S	1.00	....	0.40	19,000	7000	28
B50S	1.20	....	....	21,000	8000	24
4S	1.00	....	1.20	26,000	10,000	20
52S	2.50	0.25	....	27,000	12,000	25
XA54S	3.50	0.25	....	34,000	16,000	27
X56S	5.25	0.10	0.10	41,000	21,000	26

Table 11 – Typical or Estimated Typical Properties of Commercial and Experimental Aluminum-Magnesium Alloy Sheet

Alloy	Mg	Cr	Mn	Tensile Strength, psi	Yield Strength, psi	Elongation in 2 in., %
			Hard Temper, —H38			
Al (99% Min)[a]	....	....	....	24,000	22,000	5
XA50S	0.80	....	....	28,000	25,000	6
XE52S	1.00	....	0.40	29,000	26,000	6
B50S	1.20	....	....	31,000	28,000	6
4S	1.00	....	1.20	40,000	34,000	5
52S	2.50	0.25	....	41,000	36,000	7
XA54S	3.50	0.25	....	47,000	39,000	12
X56S	5.25	0.10	0.10	56,000	46,000	12

[a] —H18 temper.

in the annealed temper are shown by the curves in Fig. 8. An outstanding characteristic of wrought aluminum-magnesium alloys in the annealed temper is a very high elongation combined with a low yield/tensile ratio similar to that of mild steel used in structural applications. In the cast condition, the strength and elongation of the aluminum-magnesium alloys approach the level secured in annealed wrought products. In the strain-hardened tempers, very interesting mechanical properties are obtained, and while the heat of welding will remove the effects of strain-hardening in a narrow zone parallel to the weld, the limit of softening coincides with the very desirable combination of strength and elongation which is characteristic of the annealed temper.

For this reason, these alloys may be classified as a "natural" for weldments as compared to the heat-treatable aluminum alloys whose properties in the cast condition, which is present to some degree in the weld, leave much to be desired. In addition, the annealed strength of these alloys is lower than that of XA54S and X56S. Alloys of the aluminum-magnesium system are not new commercially, as sheet of an aluminum-4% magnesium alloy was produced as early as about 1925 in this country. In more recent years, Alcoa 52S was introduced about 1932 along with a similar alloy containing 3.5% magnesium, now designated XA54S. Alloys containing 5-6% magnesium were produced and sold in wire form at about the same time.

From these alloys, Alcoa 56S was derived; this alloy has been produced for many years as fine wire for insect screening and other applications. The nominal composition and the properties of these and other aluminum-magnesium alloys are listed in Tables 10 and 11. With this background, it is logical to question why sheet production of these alloys has been limited until recently to a maximum magnesium content of about 3.5% and wire production to about 5%. Several factors have influenced this situation. On the one hand, it has been known for many years that a stress corrosion problem is associated with strain-hardened sheet containing about 5% magnesium or more. Moreover, full utilization of the properties of these alloys was impossible, as satisfactory methods of welding were not available.

During the earlier investigations of these materials, occasional welds were made whose properties indicated the extremely desirable characteristics of these alloys for weldments, but consistently sound welds were not attainable. In the occasional occurrence of sound welds, the strength and elongation of the welds closely approached annealed values, and weld efficiencies in the 90-100% range were attained, using annealed properties as the reference point.

Research in the field of welding methods has resulted in the development of the argon-shielded arc method of welding and the elimination of the welding flux required in other welding processes. The properties of experimental butt welds in plate of several of the aluminum-magnesium alloys made by the argon-shielded arc processes are listed in Table 12. Magnesium contents ranging from 1% in 4S to over 5% in X56S are included. The uniformity of the weld strength is evident when it is realized that the values given in Table 12 apply to plate ranging from 0.25 to 1.00 in. in thickness. Welds made by the same methods in 61S-T6 plate, using 43S filler wire, develop a tensile strength of 30,000 psi in 0.250-in. plate, but this drops to about 26,000 psi for other thicknesses of plate.

The recently perfected inert-gas-shielded arc methods of welding largely eliminate the oxidation of the magnesium in these alloys and, in experi-

Table 12 – Properties of Argon-Shielded Arc Butt Welds – Aluminum-Magnesium Alloy Plate[a]

Plate[b]	Filler Metal	Tensile Strength, psi	Free-Bend Elongation, %	Minimum Tensile Strength Annealed Plate, psi
4S-F	52S	27,000	27	23,000
52S-F	52S	28,000	38	25,000
XA54S-F	XA54S	33,000	40	30,000[c]
X56S-F	56S	37,000	22	35,000[c]

[a] Average of consumable electrode and tungsten arc methods.
[b] Plate thickness 0.25 to 1.00 in.
[c] Tentative.

Table 13 – Properties of 75S-T6 and Alclad 75S-T6 Sheet

Material	Tensile Strength, psi	Yield Strength, psi	Elongation in 2 in., %	Endurance Limit,[a] psi
75S-T6	82,000	72,000	11	20,000
Alclad 75S-T6	76,000	67,000	11	13,000
Alclad 75S-T6, Experimental Cladding[b]	80,000	70,000	10	18,000

[a] Reversed bending, cantilever specimen.
[b] Average of experimental lots – other values typical.

mental work, sound ductile welds have been secured for alloys containing magnesium as high as 10%. In all cases, the filler wire should be of about the same composition as the plate or other parts of the weldment, in order to secure the desired combination of strength and ductility. The use of an aluminum-silicon alloy (43S) filler wire for aluminum-magnesium alloy welds results in some lowering of strength and a marked decrease in elongation in plate of 0.50 in. or more in thickness. Free-bend elongations are constant at an average level of 15%. The use of 43S filler wire for welding aluminum-magnesium alloy plate is not recommended for 52S or alloys containing higher amounts of magnesium, as the strength, particularly in the case of the thicker gages, as well as free-bend elongation values are sharply reduced.

The aluminum-magnesium alloys offer extremely desirable combinations of strength and ductility combined with these same qualities in the case of argon-shielded arc welds using filler wire of the same composition as the plate. The alloy XA54S is available in sheet and plate form, and can be used without question as regards corrosion and stress corrosion troubles. The magnesium content in X56S is within the range that may cause some stress corrosion cracking if strain-hardened sheet, stored for long periods at room temperature (— H1 tempers), or stabilized sheet (— H3 tempers), is subjected to enduring high surface tensile stress and a corrosive environment. The problem of minimizing susceptibility to stress cracking of high-strength aluminum-magnesium sheet has received much attention here and abroad, and investigations are being continued. The answer will determine the practical level of strength that may be made available commercially. It is of interest to note that X56S plate in annealed temper is being used experimentally.

High-Strength Cladding for Alclad 75S – A characteristic of high-strength alclad sheet is the relatively low endurance limit of the coating. The coatings used to secure the desired electrolytic protection are generally very soft, as few contain alloying elements in sufficient amounts to respond to heat-treatment. The single exception in commercial use is the application of aluminum-magnesum-silicon alloy coatings to alloys of the 14S composition range. Even with this combination, the endurance limit of Alclad 14S-T6 sheet is measurably less than for other 14S products and, in general, similar to other alclad sheet of the high-strength heat-treatable alloys. This problem has been under investigation for many years, and much interesting information has been accumulated. It is common knowledge that many nonheat-treatable aluminum alloys containing magnesium have endurance properties in the annealed temper only slightly lower than those of many heat-treatable alloys in the — T3 or — T6 tempers. This temper is chosen as a reference point, since any of these materials that might be used for alclad coatings on high-strength heat-treatable alloys would be in this condition in the final product. Many tests of such alloys as alclad coatings have definitely established that the high-endurance limits of alloys such as 52S, 56S, and others, do not materially improve the endurance limit of Alclad 75S-T6 sheet or any other high-strength clad sheet. Several types of heat-treatable alloys having intermediate strength and high endurance limits likewise offer little or no improvement in the endurance limit of the ultimate product.

The problem is also complicated by the desirability of maintaining an adequate potential difference between the high-strength core and the cladding, so that the latter will electrolytically protect the core. Recent work in this field has indicated that Alclad 75S-T6 sheet can be provided with an alloy coating that will develop in the ultimate product an endurance limit which approaches that of 75S-T6 sheet. The mechanical properties (Table 13) of such experimental sheet are intermediate between those of Alclad 75S-T6 and those of 75S-T6 sheet, and the new coating will resist scratching and abrasion to almost the same degree as 75S-T6. On the debit side of the ledger, the proposed cladding will show a discoloration typical of heat-treated aluminum alloy sheet and it is anticipated that the resistance to corrosion will not be equal to that of 72S now used. Both of these factors may be objectionable from the appearance standpoint. Fabricating difficulties also remain to be overcome in the production of the proposed material, and this will be reflected in the cost of such a product.

Experimental material has been produced successfully, but more detailed evaluation of the new material must be completed before it can be considered commercially. This development is being discussed primarily to acquaint the industry with the progress being made. The new material may be of interest if fatigue becomes a problem in the use of high-strength clad sheet.

Acknowledgment

The writer is indebted to the management of the Aluminum Co. of America for permission to present this paper and to various members of the Aluminum Research Laboratories staff for help and advice in the task of preparing the manuscript.

HOW ENGINES WEAR

J. F. Kunc, Jr., D. S. McArthur, and L. E. Moody, Esso Laboratories

This paper was presented at the SAE Summer Meeting, Atlantic City, June 4, 1952.

ENGINE wear is now the major factor limiting engine life. Based on the experience of the Esso Laboratories, it is believed that this wear is mainly due to friction and abrasion between the rubbing surfaces. This frictional and abrasive wear between piston rings and cylinder walls increases engine clearances to the point where the oil consumption of the engine finally becomes excessive. The engine must then be overhauled. Prolonged engine life can be realized through the development of lubricants which will reduce friction-type wear.

Survey Shows Engines Are Overhauled Because of Wear – Over a year ago a survey was made in the New York metropolitan area to determine why engines are brought in for overhaul. What factors limit engine life? It was agreed by garages and rebuilders that cars are brought in for overhaul because they are "burning oil." Field tests conducted by the Esso Laboratories had indicated that high oil consumption can result from either heavy engine deposits or excessive engine wear. Therefore, an attempt was made to learn which

RESULTS of both laboratory and field tests to determine causes of engine wear are presented in this paper. Among 100 passenger cars brought in for engine overhaul 57% showed excessive engine wear, were relatively free from heavy deposits, and indicated no mechanical failures.

Laboratory examination of the worn rings showed that in 90% of the cases the wear resulted from either friction or abrasion, or a combination of the two. What appeared to be low-temperature corrosive wear was observed in only 10% of the engines.

The field tests were completed with findings on the use of three different types of oil.

THIS paper and the six that follow – by Raviolo, Moore-Kent, Kavanagh, Jackson, Palmer, and Ellis-Edgar – comprise the complete Symposium on Engine Wear that was presented at the 1952 SAE Summer Meeting.

The discussion that took place after the presentation of the papers is published at the end of the last paper in the group, by Ellis-Edgar.

The Authors

J. F. KUNC, JR. (M '38) is in charge of exploratory product quality and development research at the Esso Laboratories – Research Division of the Standard Oil Development Co. He is a graduate of the Polytechnic Institute of Brooklyn, and joined Standard Oil in 1926.

D. S. McARTHUR (M '49) is a chemical engineer with the Standard Oil Development Co. He obtained his B.E. and Master's degrees in chemical engineering from the University of Saskatchewan and his Doctor's degree from the University of Toronto. Dr. McArthur joined Standard Oil in 1947, and has been occupied with development work on crankcase lubricants since that time.

L. E. MOODY is a research chemist for the Standard Oil Development Co. He did his undergraduate work at Wesleyan University, Middletown, Conn., receiving a B.A. degree with a major in physics. Upon completing his graduate work at the University of Wisconsin, he received a Ph.D. degree in physical chemistry.

Table 1 – Reasons for Engine Overhaul

Overhauled Because of:	Per Cent of Engines
Wear alone	57
Wear and deposits	20
Wear and mechanical failure	8
Total badly worn	85
Deposits alone	5
All other reasons	10
Total examined	100

of these two factors is the real offender in limiting engine life.

In the tour of garages and engine rebuilders, over 100 popular make passenger-car engines were examined as they were dismantled for overhaul. Each engine was rated for engine deposits such as sludge, oil-ring plugging, and oil-screen plugging. In addition, wear measurements were made on each engine. Cylinder wear at the top of ring travel was measured, and the ring gaps were determined by bringing the worn rings back to the laboratory and measuring them in a standard ring gage. The bearings, valves, and other engine parts were examined for evidence of failure. In each instance, on the basis of the inspection made, a decision was reached as to why the engine required overhaul.

As shown in Table 1, 57% of the engines examined had been brought in solely because of excessive engine wear. These engines were relatively free of deposits and showed no evidence of mechanical failure. The wear was excessive. Median cylinder wear in these cars was 0.015 in. Thirty % of the cars actually had cylinder wear in excess of 0.020 in. The ring gaps (in a standard gage) were at least six times the manufacturers' recommended gap. In some instances the ring gaps exceeded 0.200 compared with the recommended gap of 0.010 to 0.015 in. It is obvious that these engines had been brought in for overhaul because they were worn out.

Among the engines examined, 20% had both high wear and heavy deposits. In these instances it was not possible to lay the blame solely on either the deposits or the wear. There was no way of knowing which was the primary cause of engine failure. In another 8% of the engines, excessive wear was accompanied by some type of mechanical failure. All told, 85% of the engines examined were excessively worn. Heavy deposits alone accounted for only 5% of the failures. It was concluded that engine wear was the principal cause of engine failure.

Where possible, mileage figures were obtained on the engines. It was found that, on the average, the car's engine lasts only half as long as the car. It would be a tremendous advantage to the car owner if his engine would last (without overhaul) as long as his car.

Laboratory Tests Tell How Engines Wear – The worn surfaces of the piston rings from the engines surveyed in the field were examined to see if anything could be learned about how the engines had worn. To get a basis for comparison, three special wear tests were run in a Lauson engine. One test was run under high-temperature, friction-type conditions. In this test the jacket temperature was 180 F, and the engine was run for 800 hr. Another test was run under abrasive conditions. The test

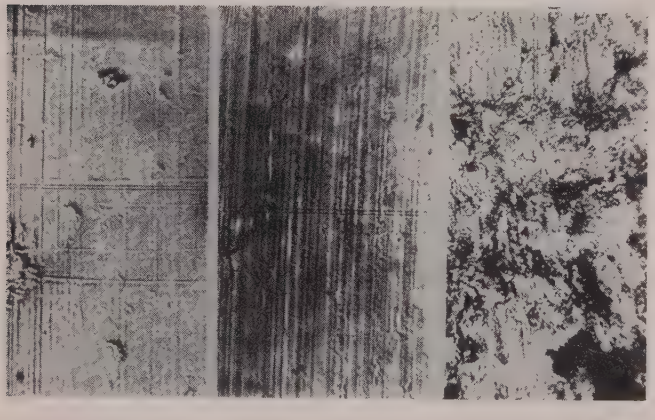


Fig. 1 – Worn ring surfaces (reduced from photomicrographs taken at 440X) from laboratory-tested engines show three types of wear. Left: friction-worn surface is relatively smooth; center: abraded surface is heavily ridged; right: low-temperature corroded surface is covered with pockmarks and craters

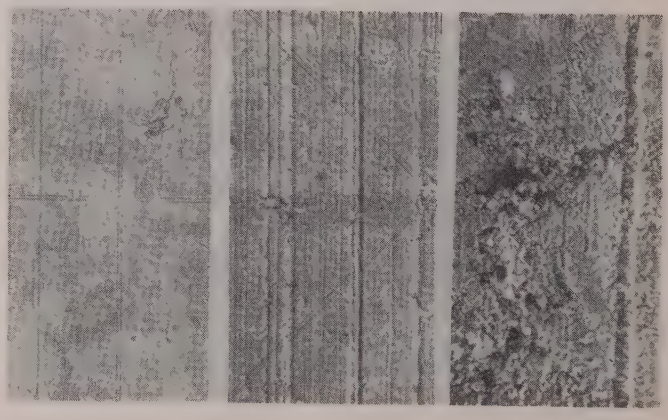


Fig. 2 – Worn ring surfaces (reduced from photomicrographs taken at 440X) from engines used in field tests. Left: 51% of rings tested showed friction wear; center: 39% of rings showed abrasive action; right, 10% of surfaces were corroded

conditions were identical with those used in the first test, except that 0.5% of 500-mesh carborundum was dispersed in the oil. The engine lasted 5 hr. The third test was run for 40 hr with the jacket temperature held at 60 F to promote low-temperature corrosion.

The worn ring surfaces from each of these Lauson engine tests were then examined using a replica technique. In this technique a transparent plastic replica of the surface is shadowed with chromium and examined under a microscope at 440X. In this way it was possible to distinguish between each of the three types of wear, as shown in Fig. 1. It was found that a friction-worn surface is relatively smooth, an abraded surface is heavily ridged, and a low-temperature corroded surface is completely covered with pock marks and craters. In subsequent work it was found that if an engine had operated for even a short period of time under low-temperature corrosive conditions, the resulting pockmarks and craters are not erased even after 100 hr of operation under friction-type conditions. The differences in the appearance of these worn surfaces provided a means of determining what type of wear had occurred in the field engines.

Replicas of the field rings were made and examined under the microscope in the same manner. It was found that, by far, the biggest proportion of the field engines appeared to have worn by friction and abrasion. Pictures of typical field ring surfaces are shown in Fig. 2. Of the surfaces examined, 90% appeared to have been worn by either friction or abrasion, or a combination of the two. What appeared to be low-temperature corrosive wear was only observed in 10% of the engines. It was concluded from this experiment that most engine wear occurring in the field is due to friction and abrasion. The abrasive wear seems to be mild. Most of the wear rates compare with those for friction-type wear.

Friction wear is much less severe than low-temperature corrosive wear. For example, in laboratory engines, friction wear is at least an order of magnitude less severe than low-temperature corrosive wear. This is illustrated in Table 2 which shows the wear rates obtained in the three Lauson tests mentioned above.

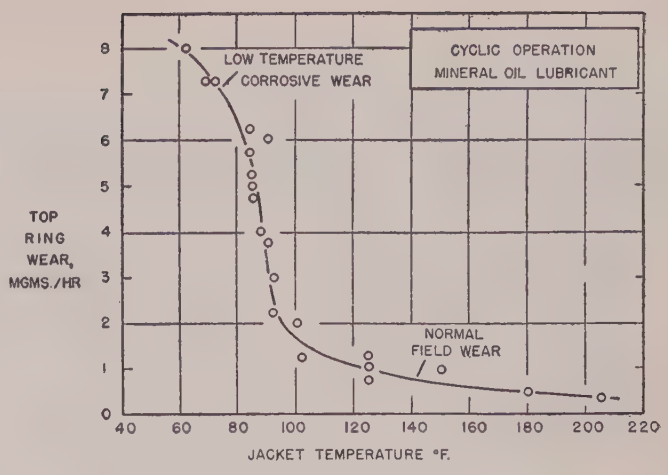


Fig. 3 – Effect of jacket temperature on Lauson engine wear

The pronounced effect of jacket temperature on wear shown in this table has been well established. As shown in Fig. 3, the wear rates jump sharply when the jacket temperature in a single-cyl Lauson engine is held below 100 F. Data are not available on the jacket temperature required to produce low-temperature corrosive wear in the average passenger-car engine. However, the appearance of the worn surfaces suggests that low-temperature corrosive wear is not common in average operation. Apparently, most engines are worn by friction and mild abrasion, at rates corresponding to the lower wear level shown in Fig. 3, rather than at the higher level.

Field Results Verify Laboratory Findings – Measurement of actual wear rates in the field substantiate this conclusion. A field test was recently completed in which three oils were run under low-temperature, stop-and-go type conditions. Four light delivery trucks were operated on each oil.

Continued on p. 230

Table 2 – Comparison of Different Types of Wear in a Lauson Engine

Wear Mainly Due to:	Conditions	Average Top Ring Weight Loss, mg per hr
Friction	800 hr at 180 F jacket temperature	0.2
Abrasion	5 hr with 0.5% carborundum in oil	800
Low-temperature corrosion	40 hr at 60 F jacket temperature	10.0

Table 3 – Comparison of Laboratory and Field Wear Data
(Light Delivery Trucks, Stop-and-Go Driving, Parked Outside Through Winter)

	Top Ring Wear Rates, mg per hr	
Lubricant	Lauson Engine, 60 F Jacket Temperature	Delivery Trucks at 20,000 Miles
Oil A	4.2	0.30
Oil B	1.9	0.58
Oil C	2.0	0.51

Critical Lubrication Areas in an Engine

Victor G. Raviolo, Ford Motor Co.

This paper was presented at the SAE Summer Meeting, Atlantic City, June 4, 1952.

IN the past few years we have seen more and more attention given to a few special qualities of motor oils. Since there has been this concentration on one or two requirements to the exclusion of others, I propose that we re-examine the whole engine lubrication problem, with an eye to fundamentals.

Despite its complexity, the problem can be reduced to various combinations of the factors discussed in the following paragraphs. These are essentials, so the order in which they are listed does not necessarily indicate their relative importance.

Lubrication Ability

The first of these factors is the ability to lubricate; and here I refer you to Mr. Webster, who says: *"A lubricant is a substance which may be interposed between moving parts of machinery to make surfaces slippery, reduce friction, and*

THIS author suggests that despite the complexity of the lubrication problem, it can be reduced to various combinations of seven essential factors. Under the first, lubrication ability, he defines the term, pointing out that lack of lubrication leads to scuffing or galling of the surface with consequent mechanical abrasion. Cooling, another requirement, can be corrected mechanically.

Corrosion prevention is discussed as the third factor, and noise control through cushioning with a viscous fluid is considered as a fourth positive factor, those things which a lubricant must do. The other three are regarded as negative, those things which a lubricant must not do. It must not oxidize or decompose of itself; it must not combine with other materials and form sludge; it must not change in viscosity.

The Author

VICTOR G. RAVIOLO (M '42) is an engineer on Ford Motor Co.'s engineering staff. He has been continuously engaged in new engine design and research. He joined Ford in 1940 and assisted in the design and development of the Ford tank engine. In connection with his large-engine work he engaged in research and development of compressors and turbines and was a member of the NACA subcommittee on compressors.

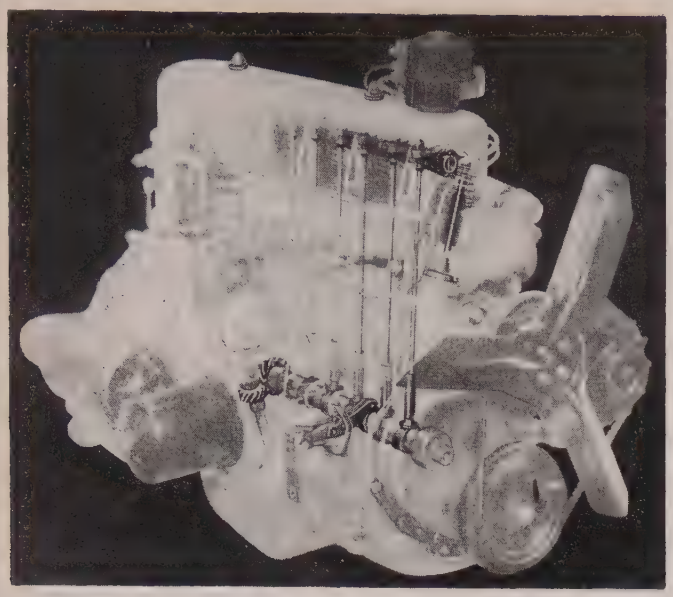


Fig. 1 – Proper lubrication is vital in critical areas in all engines

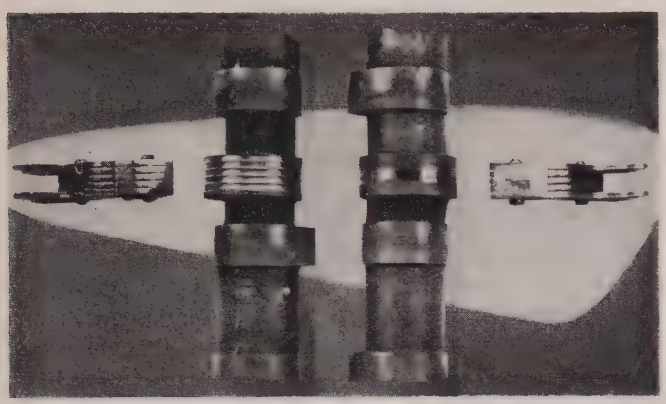


Fig. 2 – (Left) galling of eccentric surface for fuel pump arm, (right) results of correction

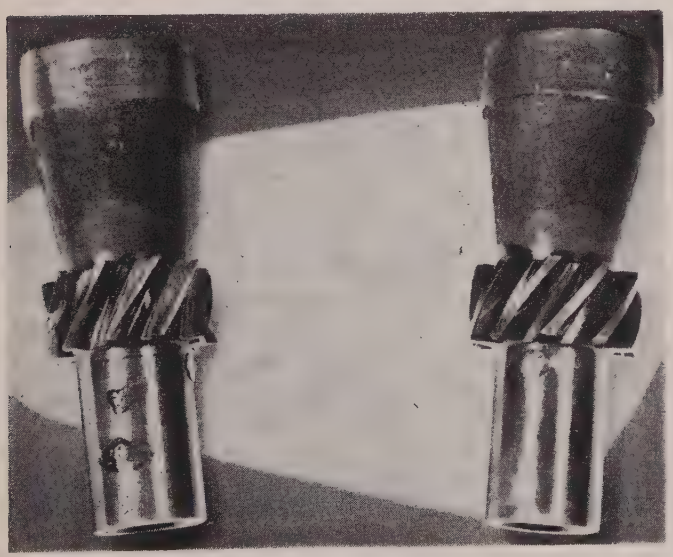


Fig. 3 – Oil pump drive gear showing scuffing of surface (left); condition corrected in view on right

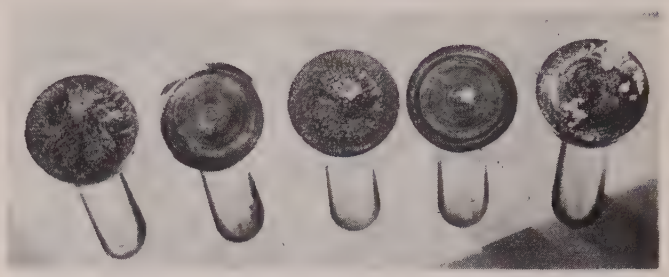


Fig. 4 – Tappets showing common types of failure and results of correction

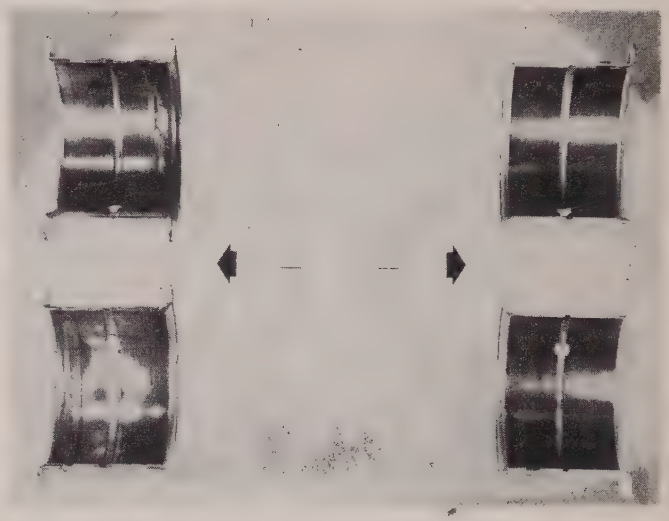


Fig. 5 – (Left) bearing failure due to inadequate oil flow; (right) a small chamfer provided additional channels for flow of oil

Fig. 6 – Corrosion of valves and guides was stopped by changing type of initial fill oil

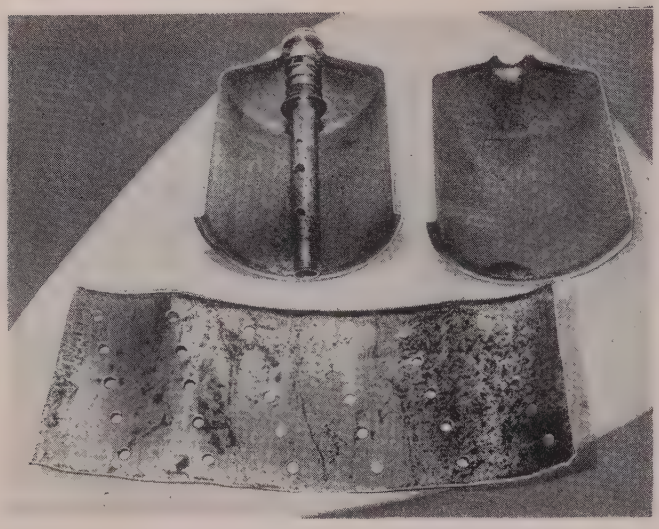


Fig. 7 – Oil filter parts illustrating type of failure caused by condensing vapors carried by lubricating oil to exposed metallic surfaces

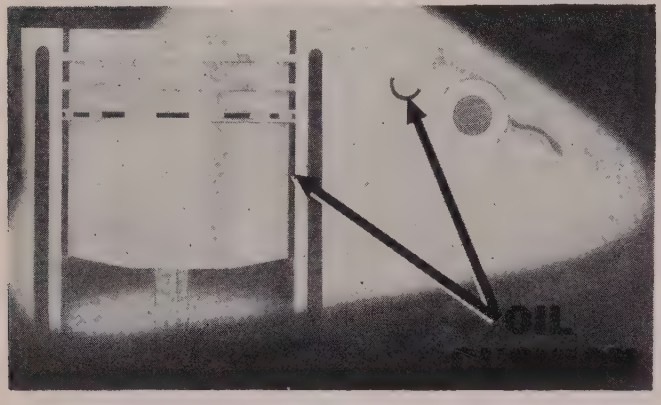


Fig. 8 – Viscous lubricant provides oil cushion for noise control

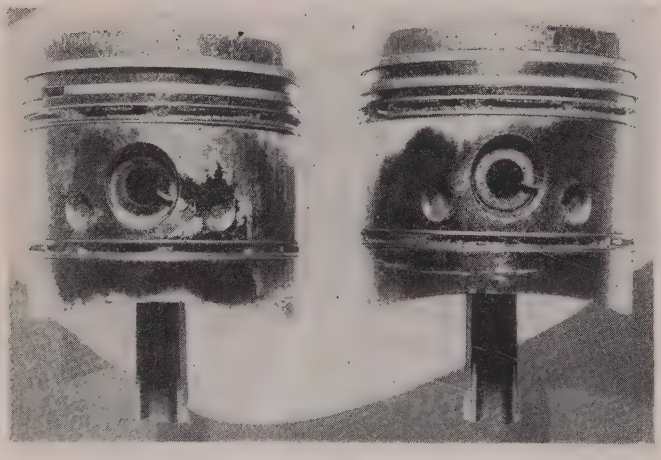


Fig. 9 – Two pistons show results of oxidation of lubricant which failed to withstand mechanical and thermal conditions

prevent sticking between the lubricated surfaces." This definition must be satisfied in the lubrication of certain critical areas in all engines. Familiar combinations, shown in Fig. 1, are: the rocker arm and shaft, valves and guides, tappets and cams, fuel pump arm and eccentric, and oil pump drive gears. Although these differ widely in lubricating method, in type of loading, and in possible metallurgical conditions, they all are subject to a common type of failure — a scuffing or galling of the surface with consequent mechanical abrasion.

Change of Materials

Figs. 2, 3, and 4 exemplify this kind of failure and show what can be done to correct it. You will notice that corrections have been of a mechanical nature, such as a change to more compatible materials. As we know, these failures could also be solved by a change in the lubricant. Oil pump drive gears, for example, have been c r i t i c a l throughout the industry for many years. Perhaps the same ingenious lubricators who helped to solve the hypoid axle problem could also help to remove these from the critical category.

Cooling

Another requirement of good lubrication is cooling. This is best exemplified in the lubrication of engine bearings. Fig. 5 illustrates a bearing failure that was due to inadequate oil flow. You will note that again the correction was a mechanical one. In this case, a small chamfer at the parting lines between upper and lower bearing halves provided additional channels for a greater flow of oil. Therefore, the success of the bearing on the right, using the same lubricant, was due solely to the more efficient use of the lubricant's cooling properties.

Corrosion Prevention

Next, a good lubricant must combat corrosion. Here we mean the kind of corrosion that attacks valves and guides, cylinder walls, and timing chains. I would like to call your attention to the valves and guides shown in Fig. 6. There is a considerable history of such failures in engines prior to their installation in vehicles. Many of our engines were built at one branch plant and shipped to another for installation. During the interim between shipment and installation, corrosion of the valves and/or guides led to this type of failure. In this case, the initial fill oil was changed from a stabilized mineral stock to what is substantially a MIL 2-126 specification oil, and the problem was solved.

Unpainted Parts

There is also the kind of corrosion which affects most of the unpainted and exposed metallic parts

in an engine to a greater or lesser degree. This occurs primarily during temporary storage periods between operating schedules in the sense that it is caused by vapors which condense and are carried by the lubricating oil to these surfaces. The oil filter parts shown in Fig. 7 illustrate this kind of failure which, depending on the lubricant and the type of operation, may continue to be a problem throughout the life of an engine.

Noise Control

Factor number four is noise control. In this sense, the oil is used as a soft cushion between metallic parts to prevent excessive noise. Here, the problem is a fairly simple one, in that the lubricant need only be a viscous fluid to satisfy the requirement. Examples of this cushioning are shown in Fig. 8.

Now, we have spoken so far of the things which I call positive factors, the things which a lubricant *must* do, and I think we are ready to talk about negative factors, the things which the lubricant *must not* do. These are probably more familiar, since they are the ones which have been receiving the greater part of our attention during the past few years.

Oxidation

First, the lubricant should not oxidize or decompose of itself. That is, it must withstand the mechanical and thermal conditions which exist in the engine without becoming itself a source of difficulty. It must not plug oil holes, or fill the clearance volume so that pieces are forced against each other with excessive pressure, or cause plain gummy sticking. The two pistons of Fig. 9 show the results of oxidation of the lubricant.

Sludge and Deposits

Second, the lubricant should not combine with other materials in the engine. For example, if it condenses in the crankcase, or combines with combustion byproducts in the crankcase, it tends to form sludge, as pictured in Fig. 10. Nor must it combine with lead to form deleterious deposits in the combustion chambers.

Viscosity Instability

Third among the negative qualities, the lubricant must not change in viscosity. Here we are referring primarily to the light oils which are made to meet, or to substitute for, the 5W specification. Obviously, if these oils change in viscosity, the engine operator cannot be sure of the starting condition. The effect of increased viscosity on cranking speed is shown in Fig. 11. In this case, the increase was sufficient to put the unstable 5W oil into the 10W range.

Fig. 10 – Condensation of lubricant in crankcase tends to form sludge when it combines with combustion byproducts

More Research Needed

The depth and scope of our examination of the engine lubrication problem has been limited by the time available. The message or purpose of this paper is to call your attention to the fact that all of these qualities are essential to good lubrication. We need more research, testing, and development, particularly of the positive qualities I have mentioned.

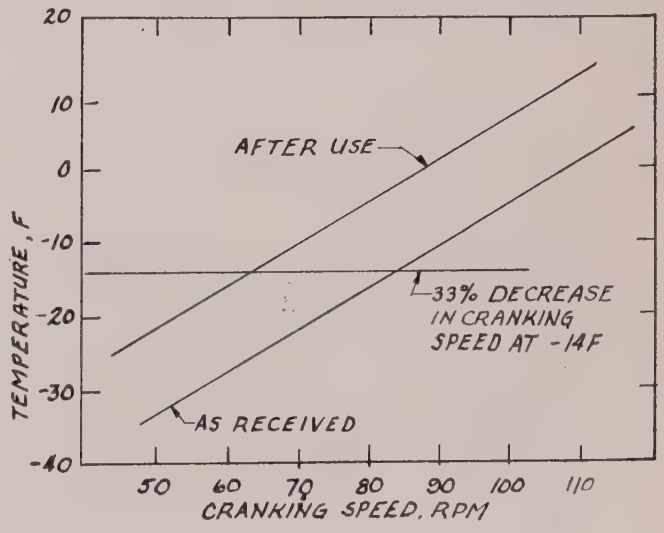


Fig. 11 – Effect of unstable 5W oil on cranking speed

Wear Measurements by Physical,

WEAR measurements on engines are made to predict the service performance of the engine, or more usually some particular parts of the engine such as cylinders and rings, under some set of operating conditions. There are at present two popular methods for measuring wear rate, namely, the direct physical measurement of the worn parts, and the less direct measurement of radioactive material worn off an irradiated part. There is also a third method, somewhat less popular, which measures engine wear by chemical analysis for iron appearing in the used lubricating oil. Our laboratory has had considerable experience with the iron analysis method, and we have had our share of experience with direct physical measurements. While we have not had direct experience with the radioactivity method, we would like, with the aid of published information on the latter method, to compare the merits and special fields of application of each of the three ways of obtaining wear rates.

The direct approach to the problem of wear measurement is to run the engine under the selected operating conditions for a certain period of time, and simply "mike" the parts we are interested in. We should, of course, have measured these same parts before the start of the test, and by subtracting the final measurements from the initial measurements we arrive at the amount of wear that has occurred. If the engine parts in question have any reasonable life, it is necessary to run a long time before enough wear has occurred to be measured with any degree of certainty. For example, liner wear rates of less than 0.002 in. per 1,000 hr are considered desirable in one make of diesel. Since measurements of liner diameter much closer than 0.001 in. are difficult to make, 500 hr is about the minimum time required for a single wear test.

If we try to repeat this test, using a new piston and liner of the same design and, let us say, from the same production lot, we will find that the amount of wear occurring under selected test conditions will not be exactly the same, even within the accuracy of our micrometer readings. Minor differences in surface finish, or hardness, or in mechanical fit, can make large enough differences in wear rates to spoil the reproducibility to some extent. Vagaries of the break-in process and even differences in the torquing of nuts can add to the

THE measurement of piston-ring wear by use of radioactivated rings has attracted considerable interest within the past few years, largely because the method is extremely sensitive.

In this paper the author compares the irradiated system of wear measurement with that of chemical analysis for iron, and in turn compares both with direct physical methods of measuring, citing the fact that the latter is the fundamental basis for radioactive and iron analyses.

The Authors

C. C. MOORE (M '40) is supervisor of the Research Division of the Union Oil Co. of California. He is a graduate of the Massachusetts Institute of Technology and for some thirty-three years has been associated with fuels and lubricants research.

W. L. KENT (M '46) is a research chemist with the Union Oil Co. of California. His work has largely been concerned with the development and testing of special fuels and lubricants. Mr. Kent is a graduate of the California Institute of Technology.

Chemical, or Radioactivity Methods

C. C. Moore and W. L. Kent, Union Oil Co. of Calif.

This paper was presented at the SAE Summer Meeting, Atlantic City, June 4, 1952.

uncertainties of wear measurements by the direct method.

For designers of engines and their component parts, the direct physical measurement of wear is frequently the only applicable method. To test a new design of piston ring, for example, the engineer has little choice but to put it into the engine and try it out. Improved gages have permitted finer wear measurements, but the method remains time-consuming and, for some purposes, somewhat lacking in accuracy. The petroleum people and those interested in the effect of fuels, lubricants, and operating conditions on engine life, have been dissatisfied with the direct measurement of wear and have sought quicker and more precise wear measurements.

One method which has proved to be more precise and quicker than direct measurement is that of analyzing the used oil for the iron worn from the moving parts of the engine. The method owes its precision to the fact that as little as one-tenth part of iron per million parts can be detected in lubricating oil, and iron contents in the range of 5 to 100 parts per million can be determined with accuracy of 5 to 10%. Translated into terms of cylinder and ring wear, this corresponds to the measurement of change in cylinder diameter of 7 millionths of an inch, with an accuracy of 3 or 5 ten-millionths of an inch. The high sensitivity of the method allows significant wear values to be obtained in short tests, of 50 to 100 hr. Furthermore, the engine does not need to be dismantled to obtain the measurements. Thus mechanical factors such as surface finish, break-in, or distortion can be practically eliminated as a variable affecting wear results. Also, a large number of tests can be made in the same engine between the time when it is broken in and the time when it is worn out, under practically constant mechanical conditions.

The $64 questions about the iron analysis method of measuring wear are: "How do you know where the iron in the lube oil comes from? Why do you think it comes from cylinders and rings?" In the first place, as we run the test engine under more and more favorable operating conditions and with better and better fuels and lubricants, the iron values are reduced to a small minimum, which is interpreted as being derived from the small residual cylinder and ring wear plus wear on crankshaft, camshaft, and so forth. In the second place, wear results by the iron analysis method have been checked against direct wear measurements. For example, iron analysis wear measurements show that the wear in a certain diesel engine increases four times in going from a low sulfur fuel to one containing 1.0% sulfur. Actual cylinder and ring wear measurements show a similar increase in wear. Similarly, decreasing cylinder wall or coolant temperature give the expected increase in the iron content of the crankcase oil.

The measurement of wear by use of radioactive piston rings has attracted considerable interest in the past few years, largely because the method is extremely sensitive, and changes in wear rates occurring in a few minutes or a few hours can be readily measured. The effect of a large number of operating variables can be measured in a short period of time, measurements that might have required years to make by direct physical methods. The effects of transitory phenomenon, such as the effect of changing from an oil of high wear rate to one of low wear rate, can be observed, which would be impossible to detect by any other method. It is estimated that the radioactive method is several hundred times more sensitive than the iron analysis method. It would be almost ridiculous to think of measuring a tenth of a millionth of an inch by direct physical means. Needless to say,

the engine is not dismantled to make radioactive wear measurements, so that mechanical factors should not influence wear values.

The radioactive methods offer an advantage over the iron analysis method in that there is no question as to where the iron in the lubricating oil came from. It can only come from the radioactive part, usually an irradiated piston ring. This fact may also be a disadvantage, for it is often assumed that the wear rate of the piston ring is the same as the wear rate of the cylinder wall. Such a one to one relationship between ring wear and cylinder-wall wear is not necessarily true. One can speculate that under conditions of corrosive wear, large areas of the cylinder wall are exposed to the corrosive gases. The piston ring, on the other hand, is in contact with the cylinder, and is relatively protected from the corrosive gases. Therefore, it might be expected that cylinder wear would be higher than ring wear under some circumstances. In many cases the rate of wear on the top ring is of the same order of magnitude as the rate of wear on the cylinder. However, there are data to show that under low jacket temperature conditions and with high-sulfur fuels, a certain oil effected a substantial reduction in top ring wear over that obtained with a reference oil, whereas the reduction in cylinder wear with this oil was relatively minor.

The high sensitivity of the radioactivity method can be a disadvantage from another point of view. Because wear values may be obtained with such short runs, there is a tendency to overlook the influence of the factor of oil deterioration on wear rates. In longer tests, such as are normally run when measuring wear by iron analysis, or by direct measurement, the average wear rate with an oil over its period between oil changes, is obtained.

There is no question but that the radioactivity method of measuring wear is very precise and has opened up new fields in engine research. It does require elaborate preparations for a test, such as irradiating the ring, and providing special equipment and safety precautions for handling the "hot" engine and oils. The iron analysis method can be applied to any engine in the laboratory or in the field on a moment's notice, with no other preparation than setting up the test conditions and taking the oil samples.

Each of the wear methods discussed has its field of application; the physical methods for mechanical studies, the radioactivity method of extreme sensitivity for studying operating conditions or fuels and lubricants problems, and the iron analysis method for ready application to the latter problems. Wear rate measurements by direct physical methods are, in a sense, the fundamental values on which the radioactive and iron analysis measurements are based. Both of the latter methods have been proved to be valid by checking against the physical method.

HOW ENGINES WEAR

Continued from p. 223

These trucks are used in light-delivery service in northern New Jersey. The fleet was operated over a period of two years on premium quality fuel, and covered 20,000 miles. The trucks were parked outside at night during the winter to assure good low-temperature conditions. It is believed that if low-temperature corrosion were important in average operation, it would be evident in this fleet. The wear rates with all the oils, however, were of the same order of magnitude as that obtained in laboratory engines operated under high-temperature conditions, for example, in the range of 0.3-0.6 mg top ring weight loss per hour.

In addition, oils which were known to be good in the laboratory in reducing low-temperature corrosive wear did not show up to advantage in the field, as shown by the data in Table 3.

Oil A was much poorer in the laboratory Lauson engine than oils B and C when the engine was run under low-temperature corrosive conditions. However, in the field oil A was better than oils B and C. Upon examination, the ring surfaces from the engines operated in the field all resembled friction worn surfaces, as shown in Fig. 4.

In conclusion, it appears that friction, along with some mild abrasion, is the principal cause of engine wear. Low-temperature corrosive wear does not seem to play an important part in wearing engines in average field service. It is felt that efforts should be directed toward reducing frictional wear in engines as a means of solving the engine wear problem. The Esso Laboratories is presently working on this problem, using the radioactive tracer technique developed by the Atlantic Refining Co.

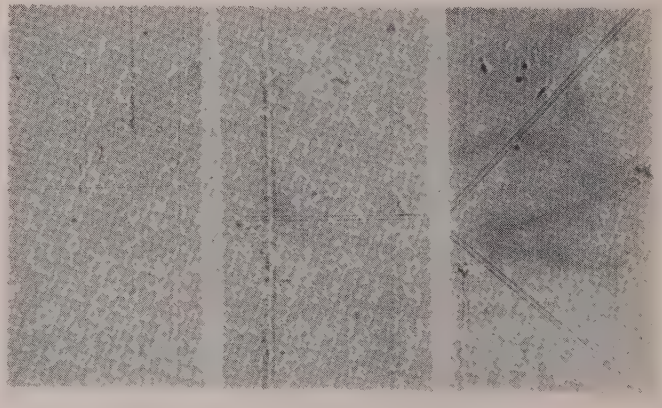


Fig. 4 – Three types of oils used in field test engines resulted in friction-worn ring surfaces. Left, oil A proved better in field than oil B, (center) or oil C (right)

Engine Wear—Comparison of Radioactive Wear and Field Measurements

F. W. Kavanagh, California Research Corp.

This paper was presented at the SAE Summer Meeting, Atlantic City, June 4, 1952.

THE technique of measuring engine wear through the use of irradiated piston rings containing radioactive iron has been publicized in the papers by P. L. Pinotti, D. E. Hull, and E. J. McLaughlin,[1] and by H. R. Jackson, F. C. Burk, L. J. Test, and A. T. Cowell,[2] as well as in a number of publications by the Atomic Energy Commission. These papers showed results of laboratory tests and little longtime service information was available. Such data have now been obtained, and are presented here in comparison with radioactive wear measurements.

Fig. 1 shows wear data obtained with three oils, using radioactive rings in a laboratory engine operated under FL-2 conditions. These conditions are usually considered representative of slow-speed, stop-and-go driving. It will be noted that the wear was essentially linear with heavy-duty motor oil A. The premium motor oil was better than the former oil early in the test, but became poorer when it had been used too long. Heavy-duty motor oil B maintained much lower wear than the other two oils throughout the test period, but the slope of the curve after the 20-hr point

[1] See SAE *Quarterly* Transactions, Vol. 3, October, 1949, pp. 633-638: "Application of Radioactive Tracer to Improvement of Fuels, Lubricants, and Engines," by P. L. Pinotti, D. E. Hull, and E. J. McLaughlin.

[2] See SAE *Quarterly* Transactions, Vol. 6, July, 1952, pp. 518-530: "Some Phenomena of Engine Wear as Revealed by the Radioactive Tracer Technique," by H. R. Jackson, F. C. Burk, L. J. Test, and A. T. Cowell.

ALTHOUGH the technique of measuring engine wear through the use of piston rings which have been radioactivated is already well advanced, there has been little opportunity to secure long service records to substantiate the data obtained in laboratory tests. Considerable data of this kind have now been secured, and are presented in this paper.

Wear measurements were determined by testing rings taken from a fleet of 32 taxicabs in service, and were compared with similar measurements made in laboratory tests.

The Author

F. W. KAVANAGH (M '36) is technical assistant to the manager of the petroleum products research development and technical service, California Research Corp. He is a mechanical and electrical engineering graduate of the University of California. He has been with Standard Oil Co. of California and later its subsidiary, California Research Corp. since 1930.

(equivalent to about 1000 miles) indicates that the additive had been consumed.

Table 1 shows wear measurements from 11 taxicabs operated in normal city driving, using two of the same oils employed in the laboratory tests, and Table 2 shows results from 21 taxicabs lubricated with the third oil used in the laboratory test series.

Referring to Tables 1 and 2, particular attention is called to the reproducibility of the data, which are considered exceptionally consistent for field test data. The spread for the three points with the premium motor oil is only 10%. Only two of the 8 points with heavy-duty motor oil A are below 0.0018 in. per 10,000 miles, while with heavy-duty motor oil B only 7 of the 21 points are above 0.0010 in. per 10,000 miles. These taxicabs were all used in normal city service through their usual overhaul periods, which amounted to 30,000 to 50,000 miles. The laboratory and field data are compared in Fig. 2. This data is considered proof of the correlation of laboratory versus field results for the products employed in these tests. This correlation was best where the 16-hr points from the laboratory engine were compared with the field data. Fig. 2 should not be interpreted as proof of a general correlation between piston-ring wear and cylinder wear. It merely illustrates that the wear rates of the rings in the laboratory engine as influenced by lubricating oil-type, cor-

Continued on p. 236

Table 1 – Cylinder Wear – Taxicab Service

Cab No.	Oil	Average Cylinder Wear in. per 10,000 miles
1	Premium motor oil	0.0021
2	Premium motor oil	0.0021
3	Premium motor oil	0.0019
	Average	0.0020
4	Heavy-duty motor oil A	0.0027
5	Heavy-duty motor oil A	0.0018
6	Heavy-duty motor oil A	0.0033
7	Heavy-duty motor oil A	0.0011
8	Heavy-duty motor oil A	0.0013
9	Heavy-duty motor oil A	0.0042
10	Heavy-duty motor oil A	0.0018
11	Heavy-duty motor oil A	0.0026
	Average	0.0023

Table 2 – Cylinder Wear – Taxicab Service

Cab No.	Oil	Average Cylinder Wear, in. per 10,000 miles
12	Heavy-duty motor oil B	0.0010
13	Heavy-duty motor oil B	0.0006
14	Heavy-duty motor oil B	0.0036
15	Heavy-duty motor oil B	0.0009
16	Heavy-duty motor oil B	0.0014
17	Heavy-duty motor oil B	0.0011
18	Heavy-duty motor oil B	0.0011
19	Heavy-duty motor oil B	0.0007
20	Heavy-duty motor oil B	0.0008
21	Heavy-duty motor oil B	0.0011
22	Heavy-duty motor oil B	0.0006
23	Heavy-duty motor oil B	0.0009
24	Heavy-duty motor oil B	0.0008
25	Heavy-duty motor oil B	0.0009
26	Heavy-duty motor oil B	0.0009
27	Heavy-duty motor oil B	0.0009
28	Heavy-duty motor oil B	0.0008
29	Heavy-duty motor oil B	0.0008
30	Heavy-duty motor oil B	0.0016
31	Heavy-duty motor oil B	0.0012
32	Heavy-duty motor oil B	0.0014
	Average	0.0009

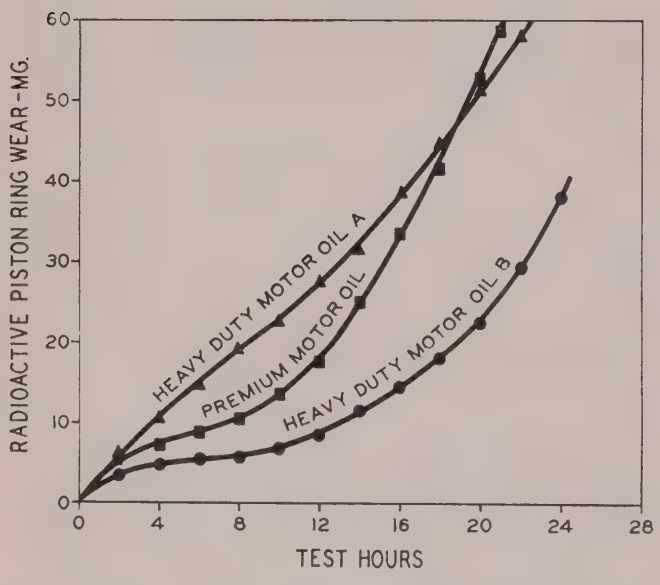


Fig. 1 – Radioactive piston-ring wear from Chevrolet engine operated under FL-2 conditions

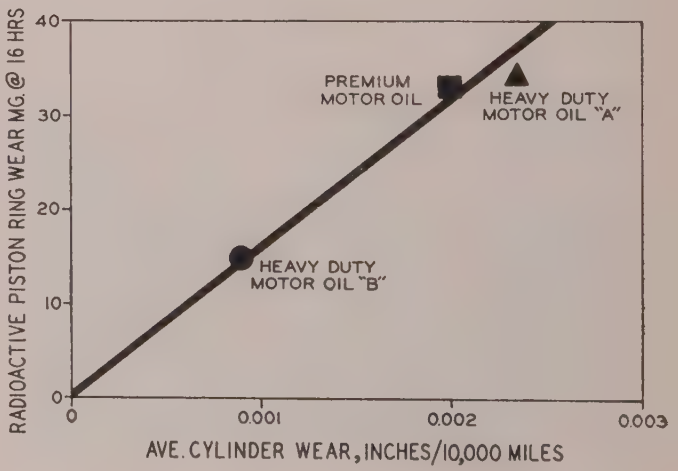


Fig. 2 – Correlation of laboratory radioactive piston-ring wear and service

Laboratory and Field Wear Tests Using Radioactive Tracers

H. R. Jackson, Atlantic Refining Co.

This paper was presented at the SAE Summer Meeting, Atlantic City, June 4, 1952.

IN November, 1951, we presented a paper[1] showing the effect of many variables on engine wear as determined in single-cyl laboratory engines employing the radioactive tracer technique.

Much of the information obtained from the single-cyl laboratory tracer tests is being investigated further in full-scale dynamometer engines and in actual passenger-car service tests. This paper will cover the results of two separate full-scale wear test programs, and their relation to laboratory tracer tests.

The first program to be discussed is one involving MIL-O-10295 or arctic oils. The CRC arctic oil group found that these low-viscosity, high V. I. oils gave rise to increased oil consumption and wear in engines, once normal engine temperatures were reached; the wear problem being particularly severe in certain 2-stroke cycle multi-cylinder diesel engines.

An examination of the viscosity-temperature data for a typical arctic oil in relationship to oils of conventional viscosities (Table 1) shows that at 300 F or above, the viscosity of arctic oil exceeds that of SAE 10. Since engine ring belt temperatures above 300 F are normal, it would appear that the lubrication failures of the arctic oils are not due to low viscosity, but perhaps to high volatility. In an effort to evaluate the effect of adding low volatility components to arctic oils, preliminary work was done with a COT (Cooperative Oil Test) engine, using the radioactive tracer technique. The engine was operated at 2400 rpm, with an upper cylinder-wall temperature of 280 F. Under these conditions, it was found that arctic oils gave much higher wear than either SAE 10 or 30 oils, but that both the addition of 15%

[1] See SAE *Quarterly* Transactions, Vol. 6, July, 1952, pp. 518-530: "Some Phenomena of Engine Wear as Revealed by Radioactive Tracer Technique," by H. R. Jackson, F. C. Burk, L. J. Test, and A. T. Cowell.

USE of the radioactive tracer technique in single-cyl laboratory engines showed the effect of many variables on engine wear, as recorded in a series of test programs conducted by the author of this paper, in cooperation with a group of his associates.

The author now presents further results covering two separate full-scale wear test programs carried out with the use of both laboratory engines and actual passenger cars in service. The resultant data are correlated with previous tracer tests in the laboratory.

The Author

H. R. JACKSON is project engineer in the Lubricants Section, Automotive Laboratory, Atlantic Refining Co. He joined Atlantic immediately following his graduation from Massachusetts Institute of Technology, and has held his present position since 1949.

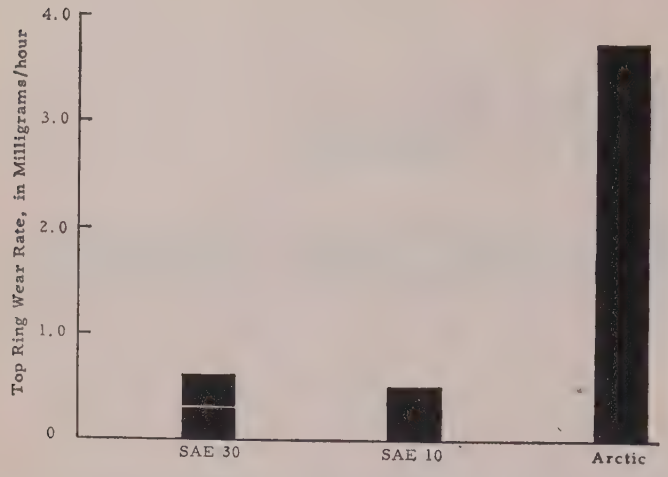


Fig. 1 – Wear of arctic versus conventional oils, using COT engine at 2400 rpm, with 280 F cylinder-wall temperature

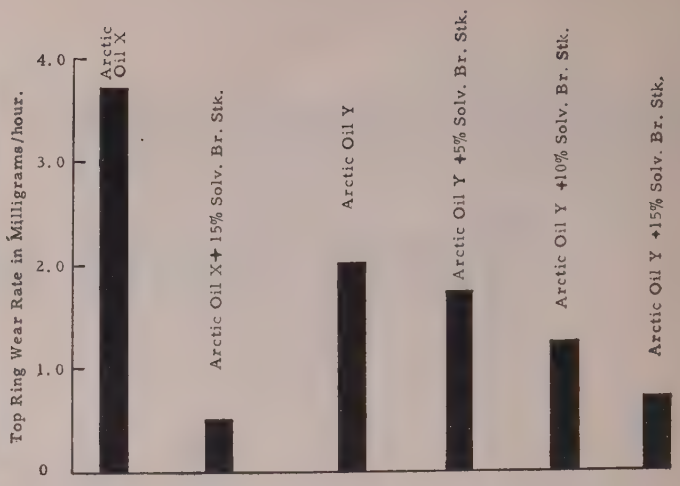


Fig. 2 – Effect of bright stock on wear of arctic oils, using COT engine at 2400 rpm, with 280 F cylinder-wall temperature

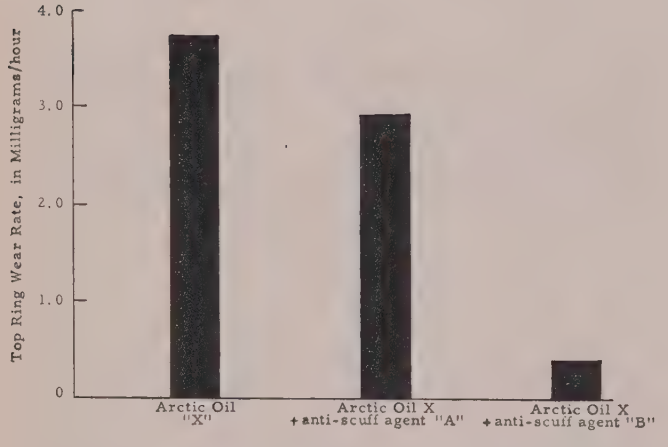


Fig. 3 – Effect of antiscuff additives on wear of arctic oils, using COT engine at 2400 rpm, with 280 F cylinder-wall temperature

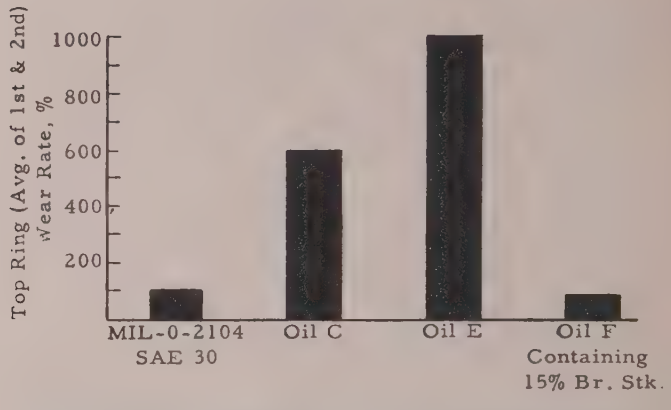


Fig. 4 – Full-scale test results with 2-stroke cycle multicyl diesel engine – 10-min idling at 400-450 rpm, and 10-min full load at 2000 rpm

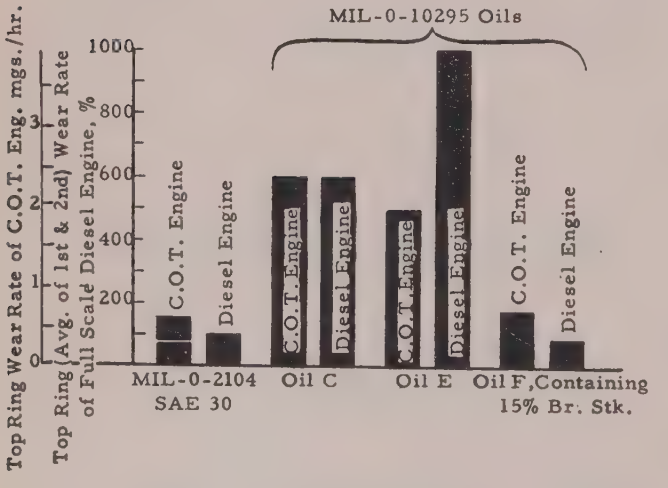


Fig. 5 – Correlation of COT tracer tests and full-scale diesel tests

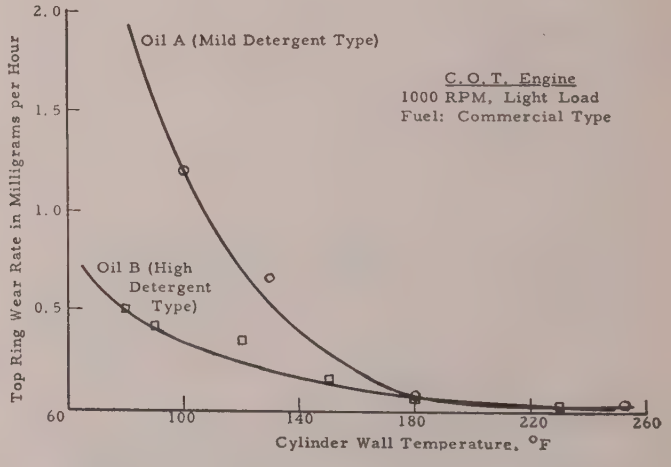


Fig. 6 – Relative wear of oil A versus oil B in laboratory engine with varying cylinder-wall temperatures, using COT engine at 1000 rpm, light load, and fuel of commercial type

bright stock, or certain antiscuff agents, reduced the wear to normal levels. These data are summarized in Figs. 1, 2, and 3. The white lines on these and other bar graphs represent duplicate tests. Fig. 1 shows that wear is about seven times higher with an arctic oil than with oils of conventional viscosity characteristics. It may be seen in Fig. 2 that the addition of 15% bright stock reduced the wear of the Arctic oils to a normal level.

Fig. 3 shows that the use of certain antiscuff agents, chemical additives which react with the metal wearing surfaces at high temperatures to form low shear films, can also be quite effective.

On the basis of these data obtained from the COT engine, the GMC Detroit Diesel Engine Division tested several arctic oils including one containing 15% bright stock, in the full-scale multicyl 2-stroke diesel engine. The results of these tests are shown in Fig. 4, the relative wear rates being determined by ring-gap increase measurements.

The four oils shown in Fig. 4 were also run in the COT engine tracer tests. The results of both the full-scale and COT tests are shown superimposed in Fig. 5. It appears that the correlation between the two test methods is satisfactory in spite of the great differences between the two engines. The SAE 30 oil and oil F were considered satisfactory in both engines, and oils C and E constituted failures.

The second wear program to be discussed is of a less specialized nature, in that it involves the effect of high versus low detergent oils on wear in wintertime passenger-car service. The passenger-car test was run during the winter of 1951-1952 in Philadelphia, and was an uncontrolled test in that normal to-and-from work driving was employed. The cars on test represented various makes and ages, and were tested "as found," with the exception that the top front piston ring in each car was replaced with a radioactive ring for the purposes of wear measurements. Using this method of wear determination, it was found that wear could be accurately determined in as little as one week's operation. The two oils tested in this program were typical of those which are available in some service stations, oil A containing a relatively small amount of detergent, oil B containing a relatively high amount. Since both these oils had previously been evaluated in the COT cold-jacket tests, it is interesting to compare the results of the two tests.

Fig. 6 shows the relative protection of the two oils with varying cylinder-wall temperatures as determined by the COT engine tests. These tests show that the spread in wear between oil A and oil B varies with cylinder-wall temperature in the 80 F to 180 F range. These differences in wear are due to the difference in corrosion protection ability of the two oils.

Table 1 – Viscosity-Temperature Comparison

Oil	Viscosities, SUS			
	−40 F	100 F	210 F	300 F
MIL-O-2104, SAE 30	.....	532	66	41
MIL-O-2104, SAE 10	325,000	191	46	35
MIL-O-10295, Arctic	15,000	120	45	36

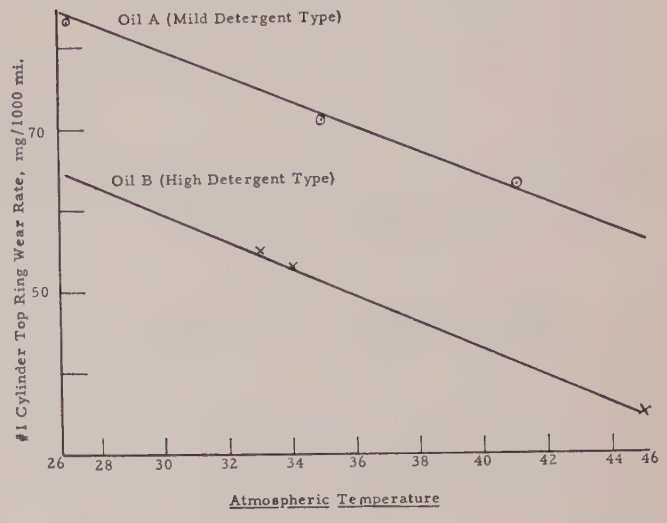


Fig. 7 – Relative wear of oil A versus oil B in service passenger car (No. 1) in the varying atmospheric temperature – average trip length 5 miles

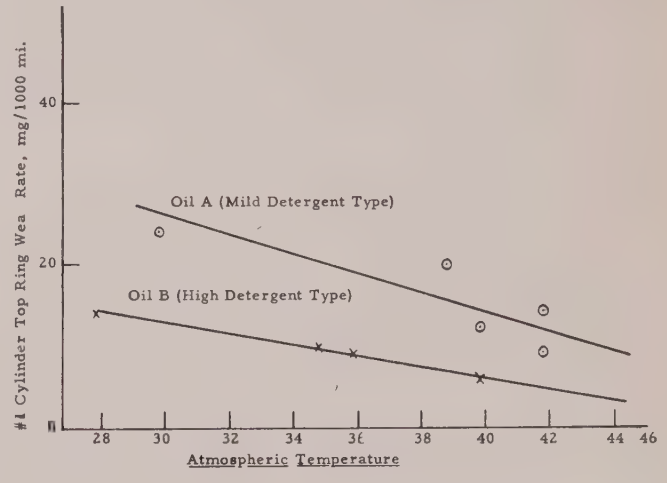


Fig. 8 – Relative wear of oil A versus oil B in service passenger car (No. 2) with varying atmospheric temperature – average trip length 7 miles

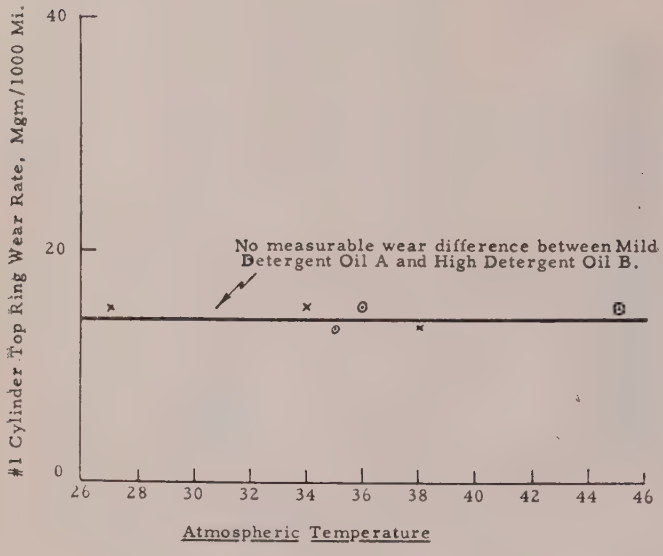


Fig. 9 – Relative wear of oil A versus oil B in service passenger car (No. 3) with varying atmospheric temperature – average trip length 19 miles, high thermostat, winter front

The performance of these same two oils in three of the passenger cars tested are shown in Figs. 7, 8, and 9. Fig. 7 is typical of most of the cars tested, with Figs. 8 and 9 representing the two extremes. It may be seen that higher wear was usually encountered during the coldest periods of operation, and the absolute wear differences between the oils were variable, depending on various factors of engine design, trip length, mechanical condition of the cars, thermostat settings, driving habits and many others. In Fig. 9 these factors combined to give a high proportion of warm-jacket operation, hence essentially no wear difference is found between the two oils.

In this test, it was shown that a high additive oil is effective in reducing engine wear.

In the arctic oil wear study, it was shown that high oil volatility may be the cause for lubrication failures, and that satisfactory performance could be obtained by the addition of antiscuff agents or heavy oil stocks.

In both of these test programs, the single-cyl laboratory engine tracer tests permitted the rapid development of wear-reducing oils whose performances were later confirmed in full-scale tests.

Acknowledgments

The author wishes to express his appreciation to H. C. Mougey and H. M. Gadebusch for their contributions to the arctic oil comparisons.

Radioactive Wear and Field Measurements

Continued from p. 232

relate with the wear rates of taxicab cylinders using the same oils.

Additional proof is illustrated by Fig. 3, which shows the oil consumption histories of taxicabs using two of the same oils. In preparing these curves, the data for heavy-duty motor oil B were available. However, for the premium motor oil only two points were established plus the total oil used by each of the taxicabs involved, which gave the area under the curve. This information, plus the assumption that all of the new engines were the same, and a knowledge of the usual shapes of oil mileage curves, allowed the curve for premium motor oil shown on Fig. 3 to be constructed. It will be noted that break-in was complete at about 6000 miles with the high-wear oil, and that the engines deteriorated rapidly in oil consumption after they had achieved peak performance. The engines using the low-wear oil took about the same mileage to break in, but the high mileage was extended for a much longer period. Therefore, the life of the engine was extended. The criterion, namely oil consumption, in Fig. 3 is the one normally used in service. That is, most engines are overhauled when the operator considers oil consumption excessive. At any degree of oil consumption after break-in, such as for example, 300 miles per qt, Fig. 3 shows that the mileage required to reach this oil consumption value was very close to being in accordance with the laboratory radioactive piston-ring wear results.

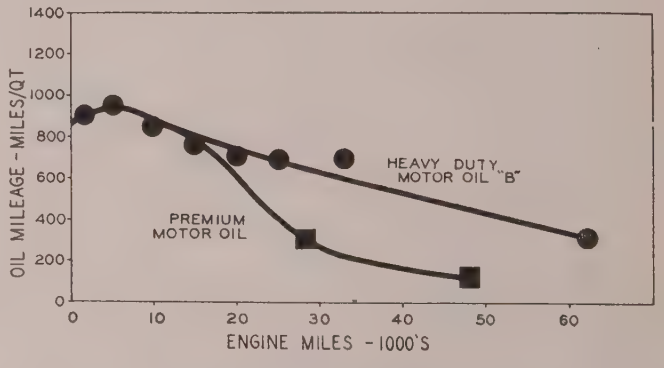


Fig. 3 – Effect of lubricating oil on engine life as shown by rate of change of oil mileage

Effect of Heavy-Duty Oils On Engine Wear in Typical Passenger-Car Service

J. L. Palmer, Lubrizol Corp.

This paper was presented at the SAE Summer Meeting, Atlantic City, June 4, 1952.

IN research work dealing with automotive lubricants, one of the most difficult phases of the work is the establishment of a significant background of test data obtained under actual field service conditions. This background is obviously required to provide for the correlative information necessary in the development of accelerated laboratory test methods.

In one very important type of automotive application, that of the passenger car, two factors cause the principal difficulties which must be handled in order to obtain the necessary field test background. These factors are:

1. Time – The average passenger car covers only about 10,000 miles per year. Therefore, in order to obtain a significant overall field service evaluation of a lubricant, such as a crankcase oil, it is usually necessary to conduct the test for a period of a year or more.

2. Wide range of operating conditions – The average passenger-car operation encompasses high-speed, low-speed, high-temperature, and low-temperature operation, and for the greatest part of its life the car sits around doing nothing but getting older. These conditions combine to eliminate the use of commercial passenger-car fleets or cars driven by full-time test drivers, as being representative of average or typical passenger-car service.

A scheme that could be used to obtain the desired results would be to run tests in the cars owned by company employees. This method would necessitate the use of a purely statistical approach in the interpretation of results because of differences in vehicle design and operating conditions. The test would, therefore, require the use of a large number of vehicles which, in turn, would result in costs and handling difficulties that are usually considered prohibitive.

THIS paper presents the results of a controlled field test of heavy-duty crankcase oils in a fleet of 15 Chevrolet cars. Operating conditions were selected to represent average or typical passenger-car service.

After 15 months of operation (approximately 14,000 miles on each car) it was found that, in comparison with the results obtained with a nonadditive oil, a heavy-duty oil of MIL-0-2104 performance level reduced piston-ring wear by 37%, reduced cylinder-bore wear by 42%, and significantly improved engine cleanliness.

The Author

J. L. PALMER (M '44), head of the Mechanical Testing Department of the Lubrizol Corp., has been associated with Lubrizol since 1947. A mechanical engineering graduate of Pennsylvania State College, Mr. Palmer also has an M.S. degree from the Chrysler Institute of Engineering. He remained with the Chrysler Corp. as a project engineer in the engine development laboratory until joining Lubrizol.

Table 1 – Piston-Ring Wear

(Test cars used in home-to-work service, equipped with 217 cu in. standard engines, in trips approximating 14,000 miles)

Car No.	Test Oil	Ring Position	Ring-Gap Increase, in. Piston No. 1	Piston No. 2	Piston No. 3	Piston No. 4	Piston No. 5	Piston No. 6	Averages
1	Base oil	Top	0.019	0.012	0.009	0.011	0.015	0.014	0.013
		Second	0.015	0.008	0.008	0.008	0.009	0.009	0.010
		Oil	0.018	0.010	0.008	0.009	0.009	0.010	0.011
3	Base oil	Top	0.015	0.011	0.011	0.011	0.011	0.011	0.012
		Second	0.009	0.008	0.008	0.007	0.008	0.008	0.008
		Oil	0.013	0.010	0.006	0.006	0.010	0.009	0.009
4	Base oil	Top	0.017	0.013	0.012	0.013	0.012	0.013	0.013
		Second	0.012	0.010	0.010	0.009	0.010	0.009	0.010
		Oil	0.014	0.010	0.007	0.009	0.007	0.009	0.009
5	MIL-0-2104	Top	0.009	0.010	0.007	0.008	0.009	0.007	0.008
		Second	0.006	0.006	0.005	0.005	0.004	0.006	0.005
		Oil	0.005	0.004	0.003	0.004	0.004	0.004	0.004
7	MIL-0-2104	Top	0.013	0.011	0.011	0.009	0.009	0.010	0.010
		Second	0.009	0.008	0.007	0.007	0.007	0.009	0.008
		Oil	0.011	0.007	0.007	0.007	0.007	0.008	0.008
8	MIL-0-2104	Top	0.012	0.011	0.009	0.011	0.010	0.012	0.011
		Second	0.009	0.009	0.009	0.008	0.008	0.008	0.009
		Oil	0.008	0.007	0.006	0.006	0.006	0.006	0.007

Table 2 – Cylinder-Bore Wear at Point of Maximum Wear

(Test cars used in home-to-work service, equipped with 217 cu in. standard engines, in trips approximating 14,000 miles)

Car No.	Test Oil	Bore Wear, in. Cylinder No. 1	Cylinder No. 2	Cylinder No. 3	Cylinder No. 4	Cylinder No. 5	Cylinder No. 6	Averages
1	Base oil	0.0037	0.0023	0.0021	0.0022	0.0029	0.0029	0.0027
3	Base oil	0.0042	0.0027	0.0026	0.0031	0.0014	0.0022	0.0027
4	Base oil	0.0036	0.0028	0.0019	0.0021	0.0021	0.0022	0.0024
5	MIL-0-2104	0.0023	0.0015	0.0011	0.0004	0.0013	0.0005	0.0012
7	MIL-0-2104	0.0031	0.0019	0.0016	0.0014	0.0014	0.0013	0.0018
8	MIL-0-2104	0.0020	0.0014	0.0017	0.0008	0.0017	0.0012	0.0015

Table 3 – Piston-Ring Wear

(Test cars used in home-to-work service, equipped with 236 cu in. Powerglide engines, in trips approximating 14,000 miles)

Car No.	Test Oil	Ring Position	Ring-Gap Increase, in. Piston No. 1	Piston No. 2	Piston No. 3	Piston No. 4	Piston No. 5	Piston No. 6	Averages
9	Base oil	Top	0.018	0.016	0.012	0.015	0.014	0.014	0.015
		Second	0.009	0.007	0.006	0.006	0.007	0.007	0.007
		Oil	0.019	0.015	0.014	0.015	0.013	0.013	0.015
10	Base oil	Top	0.019	0.017	0.016	0.015	0.017	0.017	0.017
		Second	0.013	0.008	0.004	0.004	0.006	0.006	0.007
		Oil	0.031	0.024	0.022	0.019	0.024	0.024	0.024
14	Base oil	Top	0.021	0.018	0.016	0.018	0.013	0.016	0.017
		Second	0.010	0.008	0.007	0.008	0.007	0.007	0.008
		Oil	0.027	0.026	0.023	0.026	0.017	0.024	0.024
11	MIL-0-2104	Top	0.012	0.011	0.009	0.011	0.009	0.010	0.010
		Second	0.005	0.005	0.004	0.005	0.003	0.002	0.004
		Oil	0.014	0.011	0.012	0.019	0.014	0.019	0.015
12	MIL-0-2104	Top	0.010	0.008	0.008	0.009	0.010	0.008	0.009
		Second	0.005	0.003	0.001	0.002	0.003	0.002	0.003
		Oil	0.015	0.010	0.011	0.010	0.013	0.012	0.012
13	MIL-0-2104	Top	0.012	0.012	0.009	0.010	0.009	0.013	0.011
		Second	0.005	0.004	0.004	0.004	0.004	0.003	0.004
		Oil	0.011	0.012	0.009	0.009	0.011	0.009	0.010
15	Experimental additive	Top	0.011	0.007	0.007	0.007	0.008	0.008	0.008
		Second	0.004	0.003	0.003	0.003	0.003	0.002	0.003
		Oil	0.014	0.014	0.012	0.011	0.012	0.013	0.013
16	Experimental additive	Top	0.009	0.007	0.008	0.008	0.009	0.008	0.008
		Second	0.003	0.003	0.002	0.002	0.002	0.002	0.002
		Oil	0.009	0.010	0.009	0.009	0.012	0.011	0.010
17	Experimental additive	Top	0.009	0.008	0.011	0.007	0.010	0.008	0.009
		Second	0.003	0.002	0.002	0.001	0.003	0.003	0.002
		Oil	0.011	0.011	0.020	0.012	0.014	0.017	0.014

Table 4 – Cylinder-Bore Wear at Point of Maximum Wear

(Test cars used in home-to-work service, equipped with 236 cu in. Powerglide engines, in trips approximating 14,000 miles)

Car No.	Test Oil	Bore Wear, in. Cylinder No. 1	Cylinder No. 2	Cylinder No. 3	Cylinder No. 4	Cylinder No. 5	Cylinder No. 6	Averages
9	Base oil	0.0053	0.0049	0.0045	0.0054	0.0048	0.0052	0.0050
10	Base oil	0.0042	0.0033	0.0049	0.0040	0.0047	0.0049	0.0045
14	Base oil	0.0051	0.0057	0.0040	0.0048	0.0046	0.0035	0.0046
11	MIL-0-2104	0.0031	0.0023	0.0024	0.0028	0.0026	0.0022	0.0026
12	MIL-0-2104	0.0041	0.0030	0.0027	0.0031	0.0026	0.0028	0.0031
13	MIL-0-2104	0.0044	0.0037	0.0020	0.0028	0.0022	0.0031	0.0030
15	Experimental additive	0.0035	0.0016	0.0024	0.0021	0.0022	0.0022	0.0023
16	Experimental additive	0.0033	0.0031	0.0017	0.0020	0.0017	0.0015	0.0022
17	Experimental additive	0.0032	0.0019	.0031	0.0021	0.0018	0.0021	0.0024

To build up the necessary background of test data, obtained in actual field service on its products, the Lubrizol Corp. established a fleet of 25 Chevrolet test cars, and a test method which is believed to reasonably represent typical passenger-car service. The essential element of the test method is that the cars were assigned to a group of company employees who were instructed to use the vehicles as they would their personal cars, except for a few minor restrictions.

The test results presented in this paper were obtained from 15 of these Chevrolet cars. Eight of the remaining 10 cars have not yet been torn down for inspection. The other two cars were run on a crankcase oil that is considerably different from current commercial products, and the results are not considered to be of sufficient general interest for presentation at this time.

Test Equipment and Procedure – Six of the 15 Chevrolet cars were 1950 coupes having a standard shift transmission and a 217 cu in. standard engine. The other nine cars were 1950 4-door sedans having a Powerglide transmission and the companion 236 cu in. hydraulic tappet engine. The reason for using two types of cars was to determine whether or not the rather different engine speed and load conditions encountered during accelerations in the Powerglide car would have a significant effect on engine deposits or wear. These were new cars, but before starting the tests, the engines were disassembled to permit measuring the cylinder bores and piston-ring gaps. A dial bore gage and ring gages corresponding to the nominal cylinder sizes were used to measure the bores and ring gaps, respectively. The standard black coating was stripped from the piston skirts and was replaced with tin plating in order to provide a more suitable background for observing piston-skirt deposits.

The essential operating conditions of the test are outlined as follows:

Test Equipment:

Six 1950 Chevrolet coupes, with a 217 cu in. standard engine.

Nine 1950 Chevrolet sedans, with a 236 cu in. Powerglide engine.

Type of Operation:

1. Home-to-work service.
2. Driven by company employees.
3. Driver sequence rotated every two weeks to equalize mileages and driver effects.
4. Out-of-town trip limit of 200 miles.

Oil Change Period: After the first 500 miles, and thereafter every 2000 miles.

Engine Tune-up: Every four weeks.

Oil Level Check: Weekly.

Test Gasoline: Premium grade commercial gasoline (leaded). All cars obtained gasoline from the same pump.

Engine Teardown and Inspection: After approximately 14,000 miles (15 months).

Test Oils – The oil hereinafter described as base

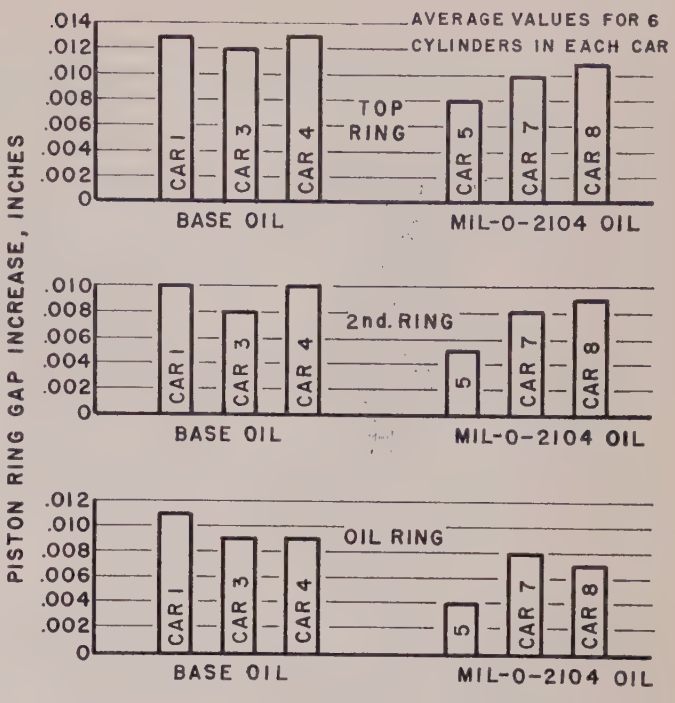


Fig. 1 – Piston-ring wear as affected by oil type in 1950 model cars used in home-to-work service covering approximately 14,000 miles, equipped with 217 cu in. standard engines

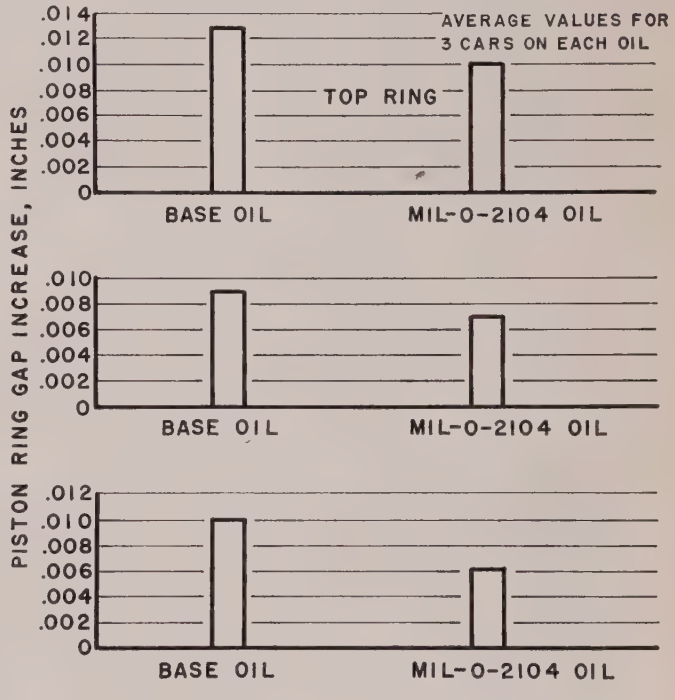


Fig. 2 – Average piston-ring wear as affected by oil type (see Fig. 1) in 1950 model cars used in home-to-work service covering approximately 14,000 miles, equipped with 217 cu in. standard engines

Table 5 – Engine-Deposit Merit Ratings

(Test cars used in home-to-work service, equipped with 217 cu in. standard engines, in trips approximating 14,000 miles)

	Base Oil			MIL-0-2104		
Engine Condition[a]	Car No. 1	Car No. 3	Car No. 4	Car No. 5	Car No. 7	Car No. 8
Varnish Deposits						
Piston Skirts	2.6	2.8	2.6	7.4	8.0	7.6
Rocker-Arm Cover Plate	6.0	6.5	7.0	7.5	8.5	8.5
Push-Rod Cover Plate	6.0	6.5	7.0	8.5	9.0	8.5
Cylinder Walls	2.8	3.3	3.0	7.8	7.8	7.7
Crankcase Oil Pan	6.0	6.0	6.0	7.0	8.5	8.0
Varnish Total	23.4	25.1	25.6	38.2	41.8	40.3
Sludge Deposits						
Valve Deck	8.0	8.5	8.5	8.5	9.0	8.5
Rocker-Arm Assembly	5.0	6.5	6.0	7.5	8.0	8.0
Rocker-Arm Cover Plate	6.0	6.5	7.0	7.0	8.0	7.5
Push-Rod Cover Plate	4.5	5.5	5.5	6.0	6.5	6.0
Oil Screen	2.0	5.5	5.0	9.5	10.0	10.0
Crankcase Oil Pan	6.5	6.0	7.0	8.0	8.0	8.0
Push-Rod Chamber	5.5	7.0	6.0	8.0	8.0	8.0
Oil Rings	8.6	9.4	9.2	9.7	9.5	9.7
Sludge Total	28.8	34.3	33.9	40.2	41.9	41.0
Combined Merit Rating[b]	52.2	59.4	59.5	78.4	83.7	81.3

[a] Based on CRC merit rating system (10 perfect).
[b] Combined merit rating (100 perfect) = varnish rating + 5/8 of sludge rating.

oil was a high V.I. solvent-extracted Mid-Continent material which contained no additives. The viscosity grade was SAE 10 W.

The oil described as MIL-0-2104 was made up of base oil plus the additive required to have the oil meet the military specification. In light of the present knowledge of the minimum additive requirements for MIL-0-2104 oils, it is estimated that this oil is overtreated by about 20% and must, therefore, be considered as being in the upper range of quality for oils of this type.

The oil described as experimental additive oil was made up of the base oil plus a strictly experimental additive material which had shown unusual promise in preliminary laboratory tests. The results obtained with this oil are included primarily to show additional evidence of the degree of test result repeatability obtained with this fleet of cars. This experimental oil could be considered as being about in the MIL-0-2104 quality range with regard to the amount of additive contained, and in its performance established to date. The distribution of the test oils in the 15 cars was as follows:

In the standard 217 cu in. engines, 3 cars on base oil, and 3 cars on MIL-0-2104 oil.

In the Powerglide 236 cu in. engines, 3 cars on base oil, 3 cars on MIL-0-2104 oil, and 3 cars on experimental additive oil.

Wear Results – The piston-ring and cylinder-bore wear data obtained are shown in tabular and chart form in Tables 1, 2, 3, and 4, and in Figs. 1, 2, 3, and 4.

Table 6 – Engine-Deposit Merit Ratings

(Test cars used in home-to-work service, equipped with 236 cu in. Powerglide engines, in trips approximating 14,000 miles)

	Base Oil			MIL-0-2104			Experimental Additive		
Engine Condition[a]	Car No. 9	Car No. 10	Car No. 14	Car No. 11	Car No. 12	Car No. 13	Car No. 15	Car No. 16	Car No. 17
Varnish Deposits									
Piston Skirts	2.2	1.5	1.5	5.8	4.9	6.3	6.8	5.9	7.2
Rocker-Arm Cover Plate	7.0	6.0	8.0	7.0	7.5	6.0	6.0	6.0	5.0
Push-Rod Cover Plate	7.0	6.0	6.0	9.0	8.0	9.0	9.0	8.5	9.0
Cylinder Walls	4.2	2.0	1.5	5.3	5.0	6.1	6.3	6.0	8.0
Crankcase Oil Pan	5.0	5.0	5.5	5.5	7.5	6.0	7.5	8.0	8.0
Varnish Total	25.4	20.5	22.5	32.6	32.9	33.4	35.6	34.4	37.2
Sludge Deposits									
Valve Deck	4.0	6.0	4.0	8.5	8.0	8.0	8.5	8.5	8.5
Rocker-Arm Assembly	4.0	7.0	7.5	8.5	8.0	8.0	8.0	8.0	8.0
Rocker-Arm Cover Plate	7.0	6.0	7.5	9.0	8.5	8.0	7.5	8.0	8.0
Push-Rod Cover Plate	4.0	3.0	3.0	7.0	7.0	7.0	6.5	6.5	6.5
Oil Screen	2.0	2.0	2.0	10.0	9.5	9.5	10.0	10.0	9.5
Crankcase Oil Pan	5.0	4.0	5.0	8.0	7.5	7.0	7.5	8.0	7.0
Push-Rod Chamber	4.0	6.5	4.0	9.0	8.0	8.5	8.5	9.0	8.0
Oil Rings	7.4	7.3	9.2	9.5	9.5	9.8	9.0	10.0	10.0
Sludge Total	23.3	26.1	26.4	43.4	41.2	41.1	40.9	42.5	40.9
Combined Merit Rating[b]	48.7	46.6	48.9	76.0	74.1	74.5	76.5	76.9	78.1

[a] Based on CRC merit rating system (10 perfect).
[b] Combined merit rating (100 perfect) = varnish rating + 5/8 of sludge rating.

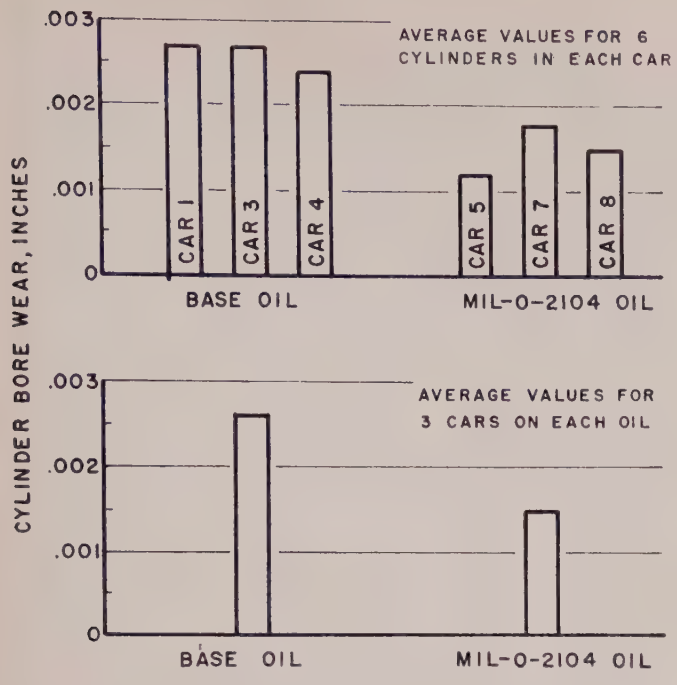


Fig. 3 – Cylinder-bore wear as affected by oil type, measured at point of maximum wear. Data obtained from 1950 model cars used in home-to-work service, covering approximately 14,000 miles, equipped with 217 cu in. standard engines

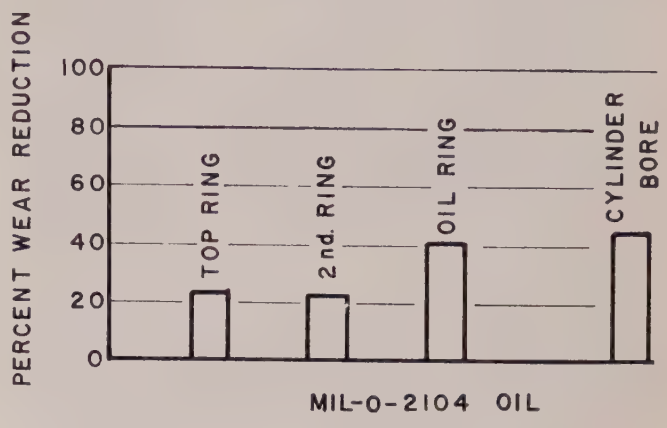


Fig. 4 – Per cent reduction in piston-ring and cylinder-bore wear, resulting from the use of heavy-duty oil in comparison with base oil. Data obtained from 1950 model cars used in home-to-work service, covering approximately 14,000 miles, and equipped with 217 cu in. standard engines

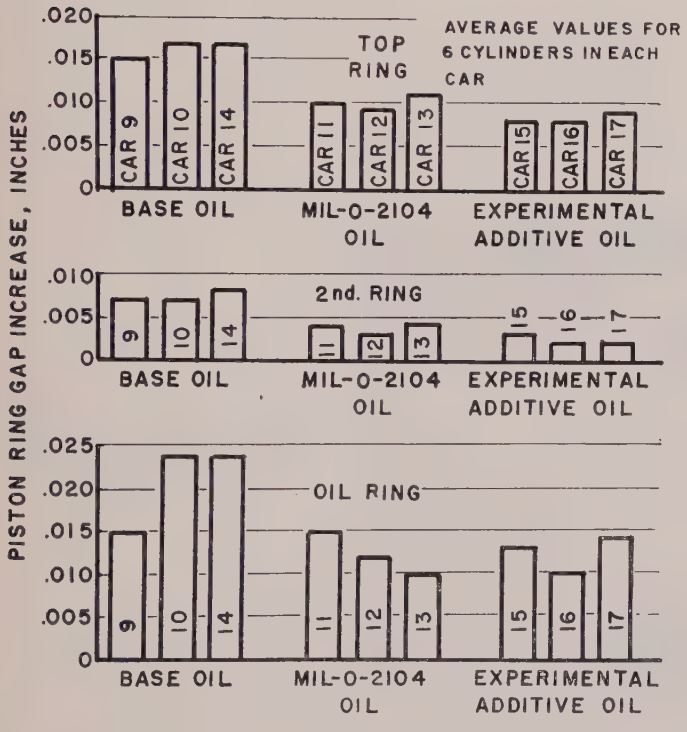


Fig. 5 – Piston-ring wear as affected by oil type in tests covering 14,000 miles of home-to-work travel. The 236 cu in. engines were 1950 Powerglide models

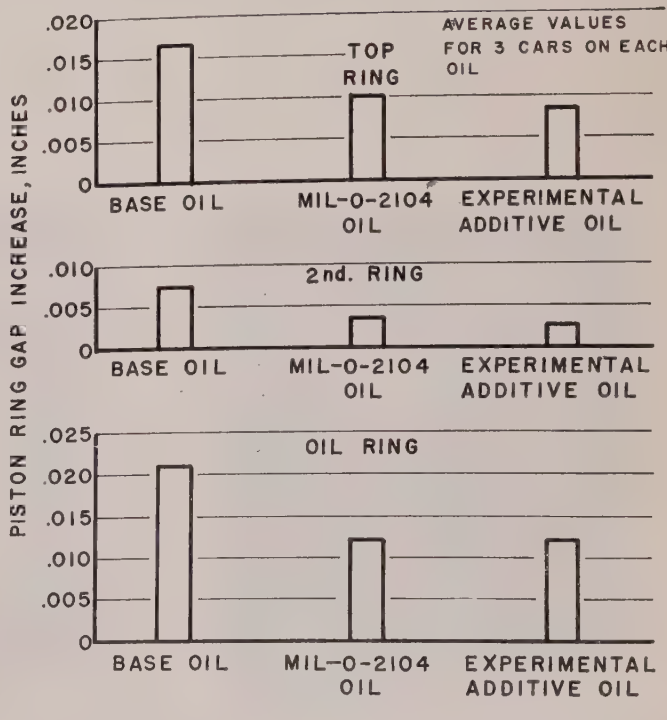


Fig. 6 – Average piston-ring wear as affected by oil type in tests covering 14,000 miles of home-to-work travel. The 236 cu in. engines were 1950 Powerglide models

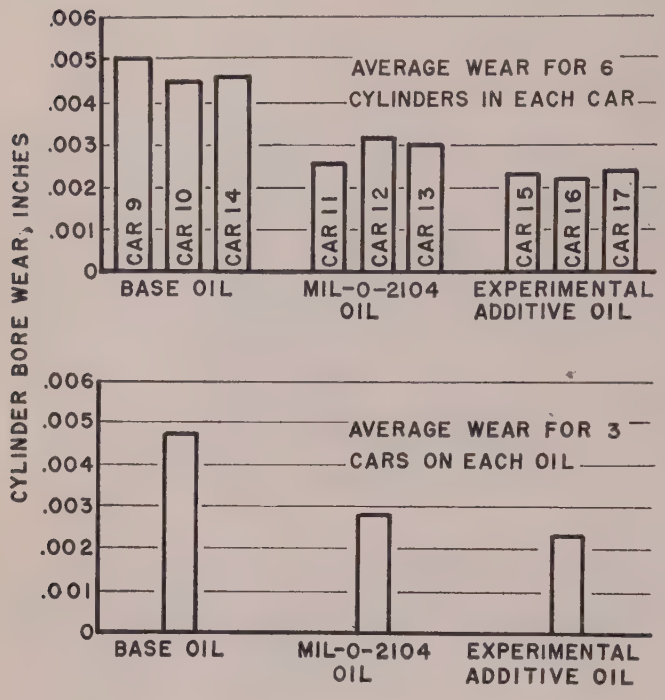


Fig. 7 – Cylinder-bore wear as affected by oil type, measured at point of maximum wear during 14,000 miles of home-to-work travel. The 1950 model engines used were the 236 cu in. Powerglide

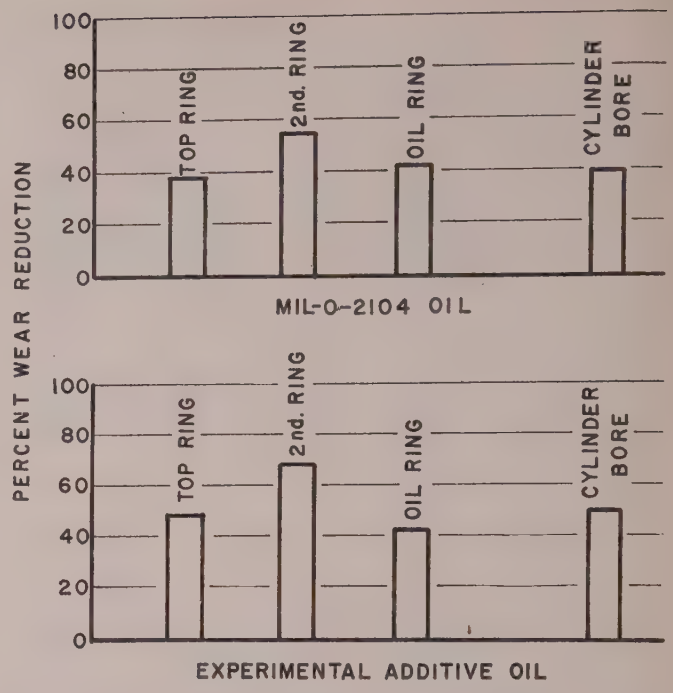


Fig. 8 – Per cent reduction in piston-ring and cylinder-bore wear resulting from use of heavy-duty oils, in comparison with base oil. Home-to-work travel for tests covered approximately 14,000 miles with 236 cu in. Powerglide engines, 1950 models

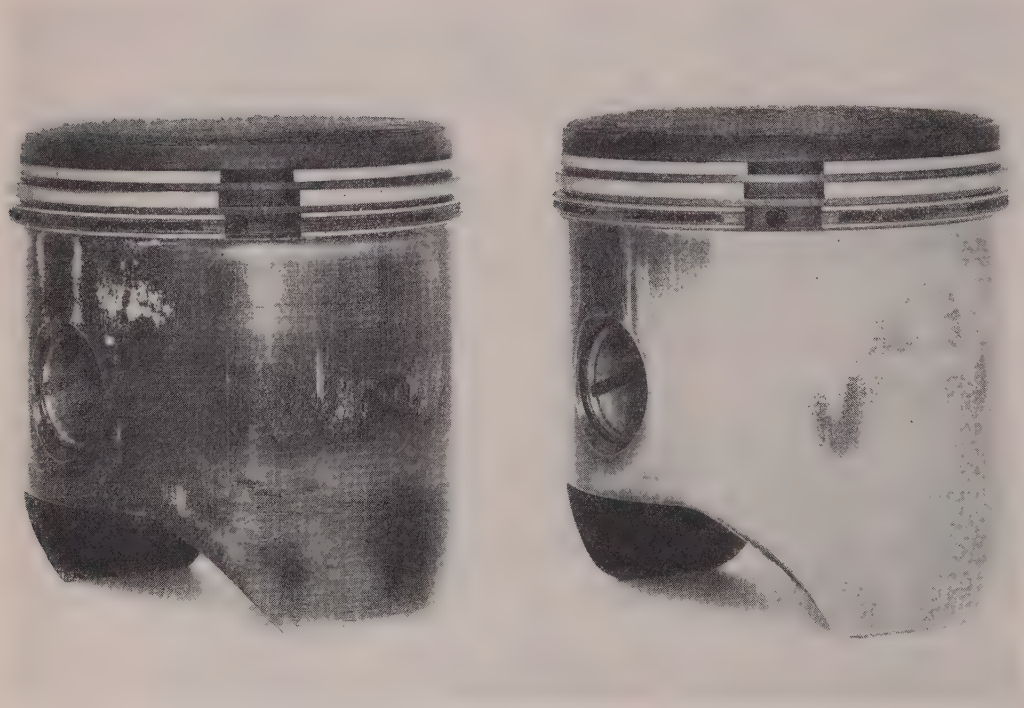


Fig. 9 – Comparison of deposits on Powerglide pistons after tests using base oil (left), and MIL-O-2104 oil (right)

Engine Deposits – The engine deposit results obtained in this test fleet are also presented in this paper as an associated matter of interest, because engine wear and engine deposits often appear to be interrelated.

Detailed engine-deposit data for the standard and Powerglide engines are shown in Tables 5 and 6. The data are expressed as merit ratings based upon the CRC rating system. Photographs of pistons, oil suction screens, and rocker-arm assemblies from two Powerglide engines are shown in Figs. 9 and 10. These photographs compare typical deposits obtained using the base oil with typical deposits obtained using the MIL-0-2104 oil.

Differences Shown Between Standard and Powerglide Types – Examination of the test results indicates that engine deposits and wear in the Powerglide cars are slightly more severe than in the standard cars. However, one factor in these tests not previously mentioned may account for this difference. This factor is that the two portions of the test fleet started on test at different times, so that in the 15 months of service the standard cars operated through only one winter season, while the Powerglide cars operated through the coldest parts of two winters. This situation eliminates the possibility of drawing positive conclusions as to the effect of the Powerglide engine-transmission combination on engine wear and deposits, because of the well-known effects of cold-weather operation on these factors.

Oil Consumption – Oil consumption rates obtained showed the poor degree of repeatability usually obtained on this item in road tests. Variation in the oil consumption rates of companion cars on any one oil showed ratios as high as 1½ to 1, and comparisons of the different oil types showed complete overlapping of the data. All of the cars had oil consumption rates that are generally considered to be in the normal range. It is expected that a much longer test period would be required before the differences in engine wear rate obtained with the different oil types would be reflected in a significant spread between the average consumption rates for each oil.

Cost of Operation – Compilation of the expenses incurred in this test during the 15-month operating period shows an average cost figure of about $100 per month per car. This figure includes the cost of depreciation, taxes and license fees, insurance, gasoline, car parts, wages of technician and mechanic, outside repair service, operating general overhead, and miscellaneous expenses.

Summary – Fifteen cars were tested for a period of 15 months, during which time approximately 14,000 miles were covered by each car. The operating schedule used was designed to represent average or typical passenger car service, and the principal factor investigated was the effect of heavy-duty oil on engine wear, and engine-deposit formation. From the results obtained in this test program the following observations are made:

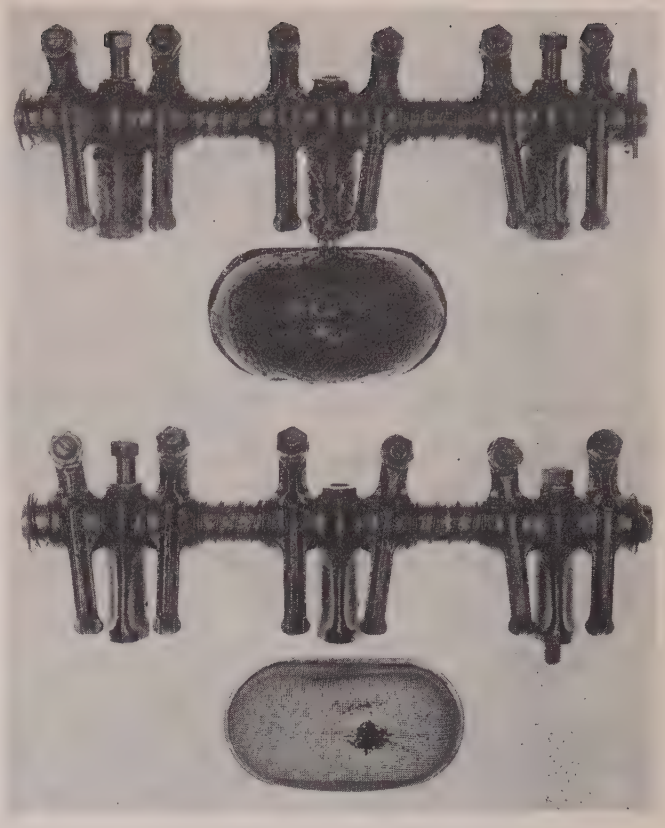


Fig. 10 – Comparison of typical deposits on Powerglide rocker-arm assemblies and oil suction screens, after tests using base oil (upper), and MIL-O-2104 (lower)

1. The use of a heavy-duty crankcase oil, in the MIL-0-2104 quality range, has the following effects on engine wear and engine deposits, when comparison is made with a nonadditive oil:

(a) Piston-ring wear is reduced, on the average, by 37%.

(b) Cylinder-bore wear is reduced, on the average, by 42%.

(c) Engine cleanliness is significantly improved. In terms of CRC merit ratings, the cleanliness is increased, on the average, from 53 to 78.

2. Triplicate tests on each oil showed good repeatability of engine wear and deposit results. For this reason the results are considered to be significant, and will adequately serve as a background of data upon which to base future laboratory research work.

3. The cost of operating a fleet of passenger cars in this manner is approximately $100 per month per car. This amount is not considered to be excessive when comparison is made with the cost of full-scale laboratory engine tests and similar research work conducted in this field.

Wear Prevention by

THE subject of wear is familiar to everyone. The expression, "worn out," is a common one whether applied to the clothes we wear, the soil we garden, the tools we work with, or the automobile we drive. These familiar examples of wear do occur, however, by a myriad of processes, and it would be difficult to consider them jointly. This, fortunately, is unnecessary for we shall be concerned with only one of these processes in this discussion and this in relation to but one wear phenomenon – the corrosion of cylinders and piston rings in internal-combustion engines.

An estimate of the magnitude of this cylinder wear is possible if data now on hand from field observations are inspected closely. Previous reports[1] show that cylinder-bore wear may average about 0.004 in. per 10,000 miles in intermittent service. If the average cylinder-wear profile is assumed to be equivalent geometrically to that found in laboratory studies, and if a few statistics

SEVEN HUNDRED tons of iron, estimated by the authors to be worn annually from the cylinder bores of American automotive engines, cause an annual engine repair bill believed to exceed $1,000,000,000. A large part of this wear is due to corrosion, particularly in severe service such as gasoline-powered delivery vehicles or stationary diesel installations.

Test results indicate that wear rates can be materially reduced by the use of crankcase lubricating oils containing high concentrations of alkalinity. The authors also present examples of radiochemistry research. Radiotracers are used to prove that oil consumed by the engine carries with it the iron debris which it contained at the moment of consumption.

Investigations conducted in the laboratory disclosed that corrosion in the cylinder zone is influenced by three factors of about equal importance: (1) high load can cause critical conditions of corrosion; (2) corrosive wear is rapidly accelerated under low cylinder temperatures; and (3) fuel in which the sulfur content has been raised to 1.0% is a wear promoter. But the tests also disclosed that high wear attributable to any or all of these conditions may be reduced by the use of lubricating oils possessing high concentrations of alkalinity.

The Authors

J. C. ELLIS (M '49) is lubricants group leader at the Martinez Research Laboratory of the Shell Oil Co. A graduate of Central Michigan College of Education, he joined Shell early in 1946, following five years of World War II service with the Army. A large portion of this service was with petroleum inspection laboratories in the South Pacific.

J. A. EDGAR (M '37) is chief research engineer of Shell's Martinez Research Laboratory. After receiving his M.E. degree from the University of California, he worked in aerodynamics and structures successively at Wright Field, Dayton, and with the Fokker Aircraft Co. in West Virginia. He joined Shell in 1933, and has been engaged in fuels and lubricants research and development for that company since then, with time out to serve in Naval Aviation for five years of the war.

Alkaline Lubricating Oils

J. C. Ellis and J. A. Edgar, Shell Oil Co.

This paper was presented at the SAE Summer Meeting, Atlantic City, June 4, 1952

of the automotive industry are used, it may be shown that each year some 700 tons of iron are lost from the cylinder bores of American automotive engines. This figure may or may not be startling, depending upon whether the reader is a private car owner who passes his worn engine on to a second buyer, or a commercial fleet operator who is constantly reminded by maintenance and overhaul costs how great wear may be. Certainly, 700 tons is not an important part of our annual steel output of 100,000,000 tons. But this is iron lost from places where it cannot be spared, and for every 2 oz which is corroded or worn from cylinder bores, an engine goes to drydock for extensive repairs. One billion dollars a year is a conservative figure for the cost of this type of engine wear. Furthermore, based upon annual scrappage estimates, some 3,000,000 vehicles are discarded annually. Although it would be presumptuous to assume this number is entirely the result of cylinder wear great enough to preclude further reboring, certainly a large portion of these 3,000,000 engines are worn out cylinderwise.

The questions that naturally arise from these considerations are whether the typical engine is wear-limited in the cylinder zone, and whether the average vehicle is engine-limited. With respect to the former, it must be established what manifestations of engine malfunctioning dictate the need for overhaul. It has been our experience that in the majority of cases, particularly for passenger cars and vehicles in door-to-door delivery service, excessive oil consumption is the major factor. Therefore, we must consider not only cylinder and ring wear, but oil-ring plugging as well. It is somewhat difficult to differentiate between the effects of these two factors, although some qualitative information is available. Shell Development Co. found that a newly broken-in Chevrolet engine ran 3,192 mpq at 55 mph. This economy was then decreased to 1,090 mpq by plugging the oil rings 100% with a mixture of graphite and molasses. On the other hand, Geniesse and McGrogan[2] found that a Chevrolet, 90,000 miles old, with worn but clean oil rings, ran 442 mpq as opposed to 72 mpq with worn and plugged rings. Thus, both wear and oil-ring plugging are important to consumption, but it appears that wear may be more limiting. As to whether the life of the average automobile is engine-limited requires a multiple answer, and one based on little more than personal experience. With respect to upholstery, tires, paint, and certain other components, the engine may not be limiting. With respect to such items as clutch, transmission, and final drive, the engine is limiting. In general, we are satisfied that the engine is the limiting factor to the useful life of the vehicle, and that the engine in turn is limited by cylinder and ring wear.

In general, wear may be classified as being either mechanical or chemical. Mechanical wear may result from the impact or impingement of a fluid on a solid, which is erosion, or it may be the result of two solids moving in contact, which is abrasion. Under all conditions of engine operation abrasion will contribute to some portion of cylinder and ring wear. However, when oil filter and air cleaner maintenance is good and the atmosphere is clean, this abrasive wear will be negligible. Nevertheless, under these conditions wear can still occur, and this wear is chemical or cor-

[1] See *SAE Journal*, Vol. 60, January, 1952, pp. 49-52: "High Additive Oils in the City, on the Long Lines, and Off the Road," by J. A. Edgar.
[2] See *Oil and Gas Journal*, Vol. 47, July 8, 1948, pp. 67-68 and 71-73: "Motor Oils – 1948," by J. C. Geniesse and J. F. McGrogan.

Table 1 – Rate of Cylinder-Bore Wear – Utilities Fleet

Engine Make	Operation with Heavy-Duty Oil: Mileage, Total	Operation with Heavy-Duty Oil: Wear Rate, in. per 10,000 miles	Operation with High-Alkalinity Oil: Mileage, Total	Operation with High-Alkalinity Oil: Wear Rate, in. per 10,000 miles	Operation with High-Alkalinity Oil: Average Final TBN-E of Oil
A	8,625	0.0034	13,514	0.0007	1.1
A	10,927	0.0050	17,966	0.0009	1.2
A	8,715	0.0039	8,972	0.0008	1.6
B	7,835	0.0046	17,343	0.0032	1.6
B	5,870	0.0058	12,786	0.0019	1.3
B	14,187	0.0040	18,217	0.0022	1.5
C	6,747	0.0050	15,532	0.0045	0.8
Fleet Average		0.0045		0.0020	

Table 2 – Rate of Cylinder-Bore Wear – Delivery Fleet

Engine Type	Operation with Premium Oil: Mileage, Total	Operation with Premium Oil: Wear Rate, in. per 10,000 miles	Operation with High-Alkalinity Oil: Mileage, Total	Operation with High-Alkalinity Oil: Wear Rate, in. per 10,000 miles
A	28,422	0.0045	9,549	0.0016
A	—	—	6,747	0.0005
A	—	—	9,716	0.0025
A	8,000	0.0064	—	—
D	25,828	0.0023	10,463	0.0018
D	16,786	0.0033	7,400	0.0013
E	37,555	0.0023	12,059	0.0004
F	—	—	11,082	0.0018
Fleet Average		0.0038		0.0014

rosive. That is, wide fluctuations in cylinder wear may be caused by changes to operating conditions which can effect corrosion only, and if the greatest part of corrosive wear is so eliminated, the remaining wear, which may be attributed to abrasion, is so small that it has no practical effect on the life of the engine. What, then, are the operating conditions which are conducive to this type of wear, and what action may be taken against it? If attention is initially focused on field experience it is found that high rates of wear universally occur in light-duty, intermittent, urban delivery type of service which is characterized by low operating temperatures, frequent stops, and a high proportion of cold starts. This service is rendered exclusively by gasoline-engine-powered equipment. Corrosive wear is less pronounced in heavy-duty service typified by long haul, highway transport usage, in which engine operation is best matched to conditions giving low wear. Both gasoline and diesel-powered equipment are found in this service. Heavy-duty service can be high-wear service in certain stationary engine installations in pumping and generating operations. This is particularly true when load factors are very high, and operating temperatures are low. Stationary powerplants may be gasoline or diesel, but wear is usually more severe in diesel engines where conditions may require the use of low-quality fuels.

From a consideration of this very brief summary of operations, it would appear that high wear could be nearly eliminated by changes in those engine operating conditions that are contributing. Such an approach, however, is unrealistic, if not impossible. For instance, it is difficult to conceive of a gasoline-powered door-to-door delivery vehicle operating under continuous high-output, high-temperature conditions, or a diesel pumping engine operating always at low load and high temperatures on the highest quality fuel. The question then arises, can high rates of wear be reduced by other means?

Two fleet tests have been conducted on the Pacific coast which provide insight to a means of wear-prevention.[1] The first fleet, in which seven engines were measured, was in a city utility service. The second, in which six engine models in the 200-240 cu in. class were represented, was a city delivery operation. These fleets totaled 59 vehicles, of which 17 were measured and inspected. The procedure was the same in both of these fleet tests; the engines were opened in the frames, and then reassembled without change or cleaning after the cylinders were measured. In Table 1 it will be seen that the vehicles had run on a heavy-duty oil from 5,870 to 14,187 miles (from new) before the first inspection. A highly alkaline oil was then put into the engines, and its use continued for additional mileages ranging from 8,792 to 18,827, when a final cylinder measurement was made. The oil change interval was 1,500 miles during both phases of the test. Results of tests in the second fleet are presented in Table 2. These tests in door-to-door delivery vehicles differed from the first fleet in that three units began the test on an oil of high alkalinity.

The information presented in both tables is revealing. Not only was the wear rate reduced

Table 3 – Rate of Cylinder-Bore Wear

Engine	Service	Operation with Heavy-Duty Oil: Total Time, hr	Operation with Heavy-Duty Oil: Wear Rate, in. per 1000 hr	Operation with High-Alkalinity Oil: Total Time, hr	Operation with High-Alkalinity Oil: Wear Rate, in. per 1000 hr
A	Pumping	1564	0.0036	3177	0.0085
B	Marine propulsion	1032	0.0070	1400	0.0005
C	Marine generator	1625	0.0029	1168	—[a]
				3633	0.0003
D	Marine generator	2043	0.0026	1016	—[a]
				3676	0.0002
E	Marine generator	4366	0.0029	640	0.0002
				1836	0.0003
F	Marine generator	3573	0.0028	1480	0.0003
				3117	0.0002
G	Drill rig	7,617	0.0026	1550	0.0006
H	Drill rig	7,006	0.0021	1373	0.0010
I	Pumping	3744	0.0012	1260	0.0004
J	Pumping	2829	0.0042	1532	0.0004
Average			0.0032		0.0004

[a] Wear too small to measure.

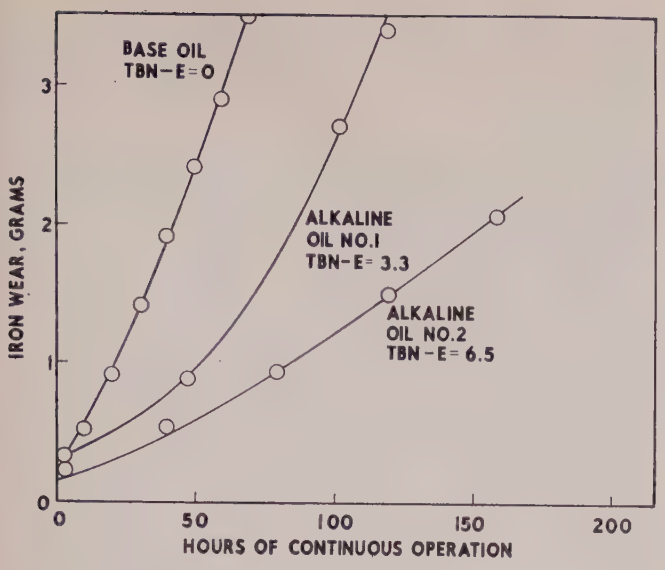


Fig. 1 – Comparative iron wear for base oils and alkaline oils tested in laboratory diesel engine with full load, 1.0% sulfur fuel, 100 F cylinder-jacket temperature

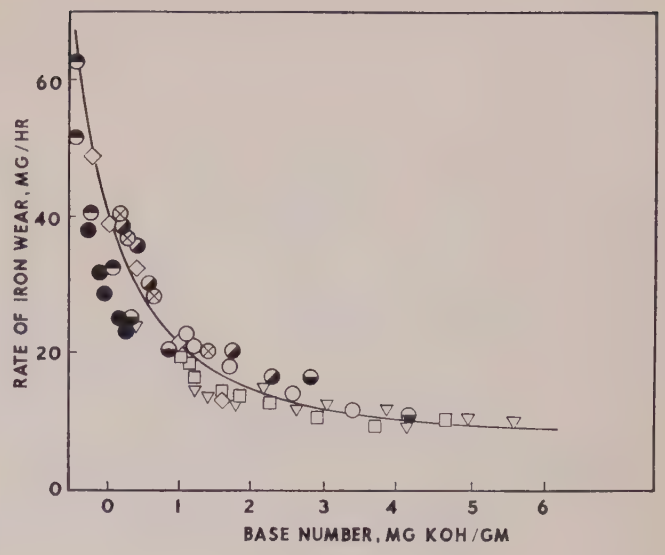


Fig. 2 – Wear rate versus base concentration for various lubricating oils tested in laboratory diesel engine, with full load, 1.0% sulfur fuel, 100 F cylinder-jacket temperature

a little more than 50% by the use of the highly alkaline oil in those instances where comparison is possible, but the wear rates with these lubricants are such that cylinder-bore life of 60,000 to more than 100,000 miles is indicated. This is exceptional for this type of service. A measure of oil alkalinity is available for the highly alkaline oils in the utilities fleet tests. It is seen here that a considerable concentration of base remains in the oil at the completion of the 1,500 miles of service prior to the oil change. These oils, when new, possess alkalinity equivalent to a total base number (TBN-E) of 6.0 or 7.0 as determined by ASTM Method D664-49. The average heavy-duty lubricant, if we exclude from consideration Series 2 oils or those meeting the old Supplemental List 1 revision of the MIL-2-104B Specification, show TBN-E values ranging from about 1.5 to 3.5 when new. We would expect that in these fleet tests the basicity of both the premium and heavy-duty oils would be zero at the end of 1,500 miles of such severe operation.

Before discussing further this observed relationship of cylinder-bore wear to oil alkalinity it will be of interest to compare performance in a service of an entirely different nature. A summary of wear results is presented in Table 3 for 10 different diesel engines of the same make, in various types of service ranging from an oil field drill rig powerplant to a marine propulsion unit. Fuel sulfur content ranged from 0.35% for engine B to as high as 1.2% for certain periods of operation in engines C, D, E, and F, and were equal for test intervals on both the heavy-duty and highly alkaline oils, except for engine I which ran on 0.15% sulfur fuel with the heavy-duty oil, but on 0.60% sulfur fuel with the highly alkaline oil. Load factors varied from intermittent to high, but were equivalent for comparable tests in the same engine. Oil and water temperatures were equivalent for all comparable tests except for engines I and J, which were operated at higher temperatures on the highly basic oil. Oil drain intervals for the high alkalinity oils were always equal to or longer than for the normal heavy-duty lubricants.

Again, although absolute values of base concentrations are not available, wear results are revealing. In each instance a reduction in bore wear has been experienced with the use of the high alkaline oil, and the average benefit is well over 50%.

These results indicate quite convincingly that cylinder-bore wear can be reduced in engines operating under severe conditions by a change in crankcase lubricating oil alone. It is indicated, furthermore, that a high level of alkalinity in the lubricant is the contributing factor to this wear reduction. This influence of lubricating oil alkalinity on wear rate has been studied further in extensive laboratory testing. The details of a large portion of this work have been reported previously,[8] and it is beyond the scope of this discussion to review them in particular here. It should suffice to summarize briefly our findings with regard to the wear versus alkalinity relationship, and to indicate the relative importance of those operating conditions which contribute to the corrosive-wear process.

The data reported here were supplied by a precisely controlled laboratory diesel engine. Operation was maintained with great regularity, and

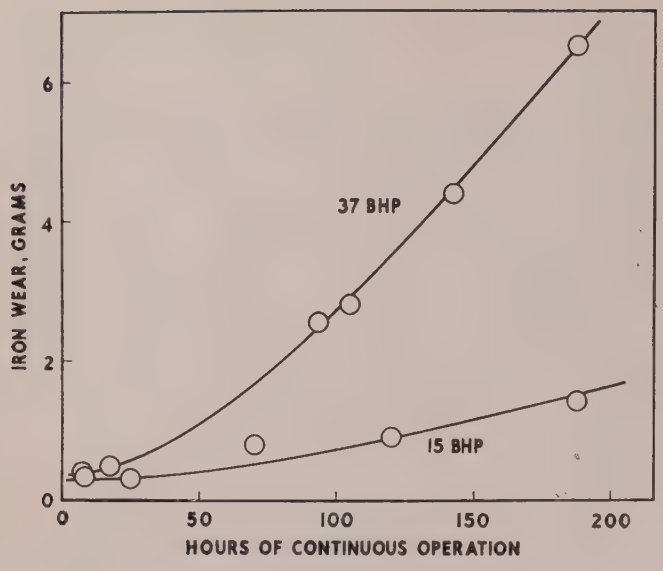


Fig. 3 – Effect of load on cylinder wear in laboratory diesel engine, with 1.0% sulfur fuel, 100 F cylinder-jacket temperature

lubricating oil alkalinity and iron wear were determined by periodic analyses of used oil samples. This technique for measuring iron wear will be discussed later. Fig. 1 presents typical wear curves for an uncompounded base oil and two oils of increasing alkalinity. These wear curves have been corrected for iron lost with oil consumption, and this consideration also will be discussed later. From curves such as those in Fig. 1 it is possible to determine instantaneous rates of iron wear as a function of the lubricating oil alkalinity at any moment. From a large number of wear determinations with many oils of different additive combinations the curve in Fig. 2 was constructed. The significance of this curve is arresting. Because of the close agreement of points for a large number of oils of diverse additive combinations, it indicates that base concentration, or alkalinity, is the property common to all of these oils which is related to wear prevention. These studies indicate that corrosive wear can be effectively countered by simple neutralization of the corrosive (acidic) materials which originate in the cylinder zone as a result of the combustion process. In other words, the film of oil on the cylinder surface will serve as a barrier to the penetrations of acidic molecules to the cylinder wall, as long as an effective concentration of alkaline molecules is present in the film. This means, of course, that the alkalinity in the oil must be depleted in providing wear protection, and that eventually the oil will lose its antiwear properties. The economics of oil change practices has been reported elsewhere[3] but it may be stated in summary that an extension in wear prevention may be effected in every instance by incorporating greater concentrations of alkalinity in the oil.

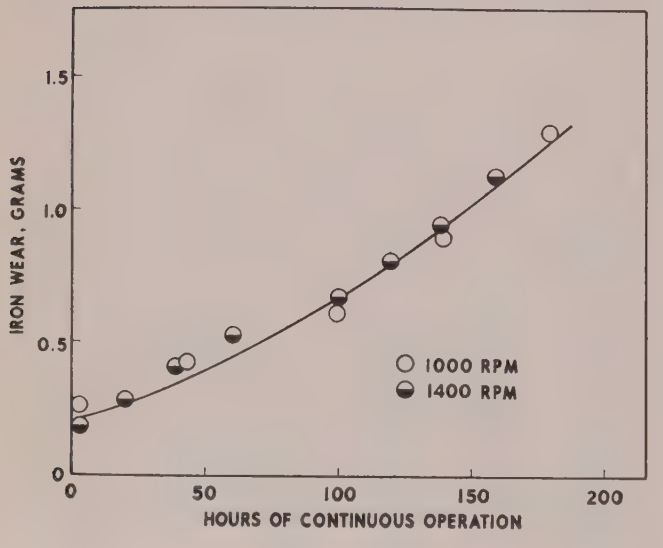


Fig. 4 – Effect of engine speed on cylinder wear in laboratory diesel engine, with 100 F cylinder-jacket temperature, 1.0% sulfur fuel, 68 psi bmep

This mechanism of wear prevention that has been assumed implies that any change in engine operating conditions which affects the forces which drive the acidic combustion products through the cylinder-wall oil film, will affect the wear rate. This has been found to be the case. As would be expected, these driving forces may be altered by changing the partial pressures of the combustion gases. This may be demonstrated by an increase in imep through added load. Results of one such experiment, utilizing the lubricant designated alkaline oil 1 in Fig. 1, are presented in Fig. 3. The effect of load reduction in decreasing wear is clearly evident. Engine speed, on the other hand, should have no effect on corrosive wear if imep is not changed. That is, the total length of time the oil film is exposed to the corrosive atmosphere is equal, regardless of engine speed. At high speeds the film is exposed many times for short intervals; at low speeds the reverse is true, and in both cases exposure durations are equal. This independence of wear and speed is exemplified in Fig. 4.

The migration of corrosive combustion products through the oil film under the driving forces involved, may be affected also by cylinder-wall temperatures, and any change in temperature conditions in the cylinder and ring zone should be manifested in the wear curve. This is illustrated in Fig. 5. The laboratory engine was operated for 240 hr at a cylinder jacket temperature of 100 F. The temperature was then raised to 160 F. The total wear was immediately reduced by a factor of more than four, as indicated by the sharp decrease in the slope of the wear curve.

In this brief discussion of the wear mechanism,

the effect of increasing the alkaline barrier in the oil film as well as the effect of changing engine operating conditions to alter the driving forces on the migrating corrosive particles, have been surveyed. These driving forces also can be raised by increasing the partial pressures of the corrosive particles in the combustion zone through an increase in their concentration.[3] A convenient means for doing this is to increase the fuel sulfur content. The results of such an experiment are presented in Fig. 6.

In summary, it may be sufficient to note from these observations that the three conditions which increase corrosion in the cylinder zone are about equally important: high-load factors can cause critical conditions of corrosion; corrosive wear is rapidly accelerated by low cylinder-wall temperatures; and fuel in which the sulfur content has been raised to 1%, is equal in importance to the above factors as a wear promoter in the severe laboratory test employed. Furthermore, the high wear attributable to any or all of these factors may be, in large part, alleviated by the use of lubricating oils possessing high concentrations of alkalinity. Before concluding, however, it is essential to discuss briefly the iron wear-measuring technique used in the laboratory studies, and to consider the validity of this method and the corrections which have been employed to account for iron losses through consumption.

It is probable that the experience of all investigators will indicate that physical wear measurements are difficult to obtain, subject to considerable error when tests are short and wear correspondingly low, poorly reproducible from test to test when engines have been disassembled between tests for the purpose of measuring wear and, in general, inadequate for a comprehensive study of wear as influenced by engine-operating variables in relation to fuels and lubricants. If test engines are not to be disturbed for cylinder measurements, or for gapping and weighing of rings, what means can be employed for a quantitative determination of the iron that is lost in the wear process? Two techniques immediately suggest themselves: (1) a chemical analysis of the crankcase oil for iron, or (2) the use of radioactive engine components. Moore[4] reported a successful application of the former technique in wear studies, and Pinnotti[5] has utilized radioisotopes successfully in similar investigations. Both approaches have been used in the studies reported here.

The use of crankcase oil iron analyses for determining the extent of corrosive wear in the

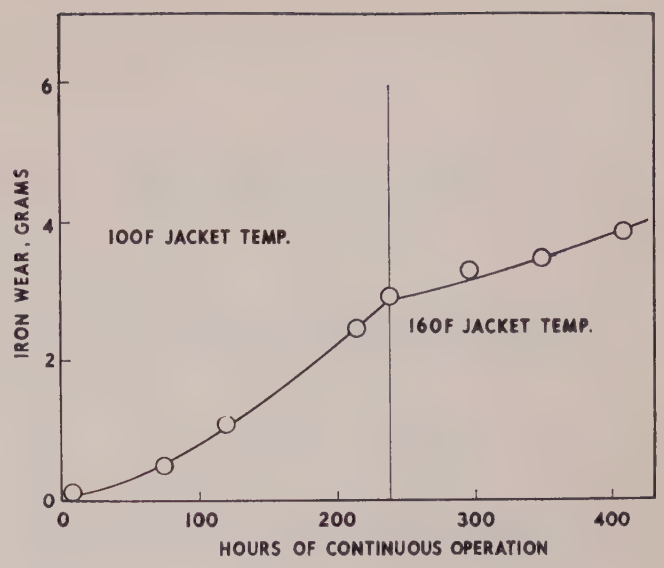


Fig. 5 – Effect of jacket temperature on cylinder wear in laboratory diesel engine with 0.3% sulfur fuel, full load

cylinder zone, which includes both bore and ring wear, presupposes (1) that all iron in the oil originates from this source, (2) that all oil lost by normal consumption carries with it an aliquot of iron proportional to the bulk crankcase oil concentration, and (3) that engine sludge deposits do not contain high iron concentrations.

The most direct and simple answer to (1) is to determine an iron balance, but to do this requires confirmation of (2). This was accomplished in the laboratory diesel engine by operating for 20

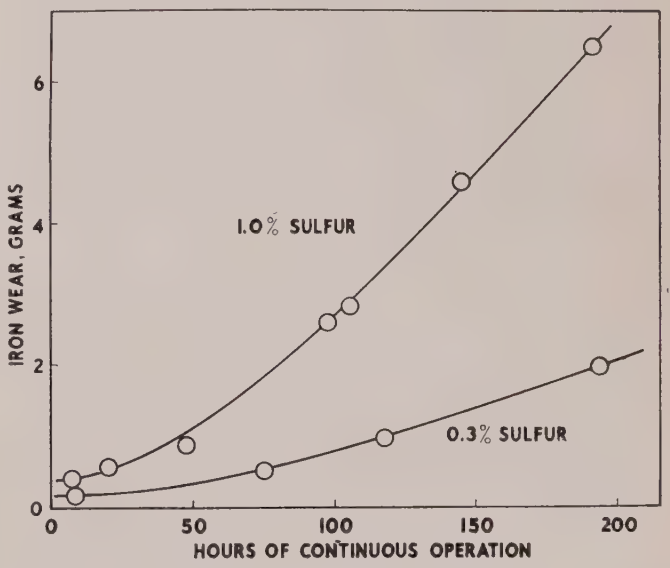


Fig. 6 – Effect of fuel sulfur content on wear in laboratory diesel engine, with 100 F cylinder-jacket temperature, full load

[3] See *SAE Journal*, Vol. 59, June, 1951, pp. 63-65: "Series 2 Oils Pay Their Way," by W. G. Brown, F. A. M. Buck, J. A. Edgar, F. E. Kronenberg, and J. M. Plantfeber.

[4] See SAE *Quarterly* Transactions, Vol 1, October, 1947, pp. 687-693: "Effect of Sulfur and Nitrogen Content of Fuels on Diesel-Engine Wear," by C. C. Moore and W. L. Kent.

[5] See SAE *Quarterly* Transactions, Vol. 3, October, 1949, pp. 634-638: "Application of Radioactive Tracers to Improvements of Fuels, Lubricants, and Engines," by P. L. Pinotti, D. E. Hull, and E. J. McLaughlin.

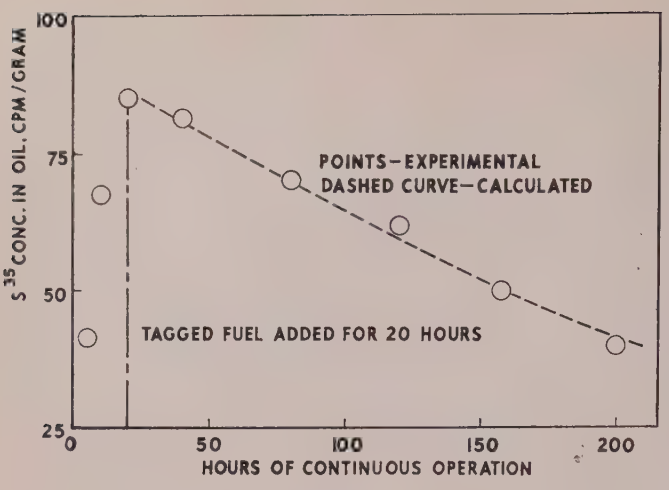


Fig. 7 – Predicted versus observed loss of fuel sulfur products from crankcase oil in laboratory diesel engine

hr on a fuel containing the radioactive sulfur isotope, S35. Any radioactive sulfur which is found subsequently in the crankcase oil then will have originated in this fuel. This sulfur may be unchanged from the organic form it had when added to the fuel, or be in an altered organic form as a result of reaction with fuel or oil hydrocarbons and oxygen, or combined with the metallic oil additives and with iron corroded from the cylinders. At the end of 20 hr, the radioactive diet was suspended, and the engine ran on for 180 hr more with normal fuel. The decline of radioactivity of the crankcase oil was carefully plotted during this 180-hr period because it represented exactly the rate of loss of corroded iron from the crankcase oil (as well as loss of other sulfur compounds which had been deposited during the first 20 hr). It transpired that the loss of these sulfur compounds was at a rate almost exactly equal to the product of the rate of oil consumption times the concentration of sulfur in the oil, as shown in Fig. 7. In this figure, the dashed line depicts what the loss of activity (counts per minute – cpm) would be if sulfur were lost in aliquot with oil; the circled points are the readings of the Geiger counter. This is considered conclusive proof that the oil lost by consumption processes from the crankcase carries with it all of the iron which existed in the oil at the instant it was consumed. Corrections to the measured iron content of the crankcase oil have been made accordingly. Using the measurements of accumulated iron in the crankcase oil, after making the corrections for iron lost through consumption, it was possible to make an iron balance. This was accomplished in the laboratory diesel engine by making, during overhaul and reconditioning, an accurate measurement of cylinder dimensions and piston-ring weights. Cylinder bores were measured, using a Caterpillar-type wear gage, and profiles were determined in four positions. The engine was then operated on the testing schedule for 2072 hr. All operation was at full load and constant speed, although fuel quality, lubricating oil, and cylinder jacket temperatures were varied during various portions of the total test. During these 2072 hr the engine was stopped nine times for reasons common to all such operations but, insofar as possible, operation was continuous, and all oil changes and flushings were made "on the fly." At the conclusion of the test, cylinder bores were remeasured and piston rings reweighed. Also a complete summation of all chemically determined iron, corrected for consumption losses, was made. These results are summarized in Table 4. This agreement is particularly good when possible errors inherent in such measuring techniques are considered. The conversion of bore measurements to iron loss by integration of four wear profiles is of course an approximation, although an excellent one, or so it is believed. It has been found in our studies that the chemical iron determination by the colorometric method[4] is applicable with good repeatability, but nonuniform sampling may cause errors from this source. To investigate this possibility, that is, that engine deposits may contain high concentrations of iron, samples of sludge deposits were collected from various sections of the engine and subjected to analysis. In no instances were iron contents of these samples appreciably different from those of the bulk crankcase oil at time of sampling.

These results provide considerable evidence to substantiate the assumptions made in these studies that:

1. All iron present in the crankcase oil samples originated from the cylinder zone.

2. All oil lost by processes of consumption carried with it an aliquot of iron proportional to the concentration of iron in the bulk crankcase oil.

3. Engine deposits did not contain high concentrations of iron.

In conclusion, it appears reasonable to state that corrosion of iron from the cylinder zone is an important wear mechanism; that this wear

Table 4 – Comparison of Physically Measured versus Chemically Determined Iron Wear

(Laboratory diesel engine, full load, constant speed, continuous operation)[a]

Cylinder No.	1	2	3	4	Total
Ring Wear, g	5.72	6.14	6.06	2.46	20.38
Bore Wear, g	14.50	14.40	15.85	12.05	56.80
Total	20.22	20.54	21.91	14.51	77.18

Summation of Chemically Determined Iron in Crankcase Oil, 73.1 g

[a] Except for required short shutdowns for routine maintenance.

may be alleviated by proper selection of fuel and engine operating conditions but, when these conditions cannot be imposed because of uncontrollable factors, crankcase lubricating oils possessing a high concentration of alkalinity will effectively reduce this corrosive wear.

DISCUSSION OF WEAR SYMPOSIUM PAPERS

Oil Improvement May Be Due to Many Factors

– George Round
Socony-Vacuum Oil Co.

IN Mr. Jackson's paper he presents data to show that the addition of 15% bright stock to arctic engine oil (MIL-O-10295) reduces the wear rate to a point comparable with that of SAE 30 engine oil or arctic oils containing selected antiscuff agents. He suggests that the high wear of the conventional arctic oils is due to high volatility rather than low viscosity.

While Mr. Jackson gives Saybolt viscosity data in Table 1 for the arctic oil without bright stock or additives, he does not show similar data to indicate how much the viscosity was increased by the addition of bright stock or whether the blend was cut back with additional light oil to maintain the 300 F viscosity of the original arctic oil. In the absence of such data, one cannot be certain whether the wear reduction of the bright stock blend is due to a viscosity increase, the effect of a volatility change in keeping an adequate film of oil on a hot surface, or to the known value of bright stock as a lubricant under boundary film conditions.

In dealing with such low viscosity fluids, Saybolt viscosities are misleading and the kinematic scale is preferable. Depending upon the viscosity of the bright stock used, 15% added to a 45 SUS at 210 F oil raises the viscosity to 49/52 SUS. At 300 F these figures change from 36 SUS to 37.3/37.5 SUS. On the kinematic scale 36 SUS at 300 is 2.88 centistokes, 37.5 SUS is 3.38 centistokes, a 17% increase in viscosity which is a substantial amount. Hence, we may well conclude that the improvement cited by Mr. Jackson may be due to all these factors – viscosity increase, reduced evaporation, and better lubricating properties under boundary film conditions, instead of to reduced evaporation alone as implied in Mr. Jackson's paper.

Chemical Wear Believed To Exceed Mechanical

– J. P. Stewart
Socony-Vacuum Oil Co.

TWO distinctly different mechanisms of wear have been presented – frictional wear and low-temperature corrosive wear; each has been set up by its proponents as the major cause of shortened engine life induced by cylinder and ring wear – or at least the several development programs appear to be directed toward one or the other, which amounts to the same thing. As a consequence the question then arises – which of these two mechanisms is mainly responsible for Mr. John Q. Public's car or truck engine's precocious and oversized appetite for lubricating oil?

Mr. Kunc and his co-authors, as the proponents of the friction type of wear, arrived at that viewpoint after examining the worn surfaces of piston rings from a large number of engines undergoing overhaul. Most of the other authors deal with corrosive wear induced by engine operation at low cylinder wall or jacket temperatures, a condition long recognized as the source of short engine life in so-called door-to-door delivery service.

If we confine ourselves to the gasoline engine, the predominant engine type, it is obvious that both mechanisms must occur in service, for engine operating temperatures significant to each are experienced alternately on the road – low cylinder wall temperatures immediately following the engine start, and high cylinder wall temperatures after warm-up. However, if we examine data from engine wear tests one is forced to the conclusion that the corrosion mechanism is a major, if not the major, source of ring and cylinder wear, even in engines of vehicles normally considered as being operated under heavy-duty or high-speed service.

Many have observed that engine wear tests over a long period exhibit abnormally low ring wear if the engine operation is at load and continuous, but if the engine is operated intermittently with cold starts, the ring wear is augmented considerably. Again in radioactive wear tests, emphasis is placed usually on the tremendous difference in ring wear between low cylinder and high cylinder temperatures. What seems to me of equal, if not greater, importance to average over-the-road operation, is the surprisingly long period of engine running required under noncorrosive conditions, when subsequent to a period of corrosive wear, before the normal low noncorrosive wear rate is reached. This period, usually of 5 hr or more duration, is presumably one during which the corroded surfaces are smoothed, corrosion products scrubbed away, or the ring surfaces abraded by the debris from such action. This phenomenon occurs with an increase in cylinder wall temperature, as well as with a change in the oil type, the comparison usually featured. Moreover, it is the consensus that the low-temperature corrosive wear begin simultaneously with the engine start. Is it then illogical to anticipate an analogous wear cycle in normal commercial engine operation – a high rate of corrosive wear immediately following a cold engine start and during early warm-up, followed by a "healing" period at higher than average wear for some time after normal load and operating temperatures have been reached? Current data do not permit an exact assessment of the extent to which this type of wear contributes to the total wear, but over an 8-hr operating period 50% would seem to be a reasonably conservative estimate, even though the

engine would be operating under optimum conditions some 95% of the time; under less favorable operating conditions the wear due to corrosive action would, of course, soar. And as a passing comment, would not the rings at the end of such an 8-hr operating period exhibit the wear pattern diagnosed by Mr. Kunc as the resultant of "frictional" wear?

Probably few appreciate the amount of moisture, that essential element to corrosion, that condenses on the cylinder wall during engine idling at low cylinder temperatures. Still very vivid over the years is the writer's recollection of an actual engine freeze-up – observed ice on the cylinder walls – brought about by intermittent idling during cold room starting tests – a 100 F jacket temperature solved that problem; or during longtime low temperature engine wear tests, the unorthodox but necessary and successful use of a Skinner rectifier to remove water condensate continuously from the crankcase oil, thus preventing the formation of heavy emulsions which up to that time had interfered with oil circulation. Cylinder wall "wear" during that test was of the order of 0.001 in. per 10 hr, and that under load. With such recollections of "barnyard" engineering results – confirmed by that offshoot of scientific atomic research, the radioactive wear test – can one be censured for believing that more automotive engines wear out chemically than mechanically?

Kunc-McArthur-Moody Closure to Discussion

THE excellent discussion on the paper "How Engines Wear" offered by Mr. Stewart was appreciated. We certainly agree that the principal effort must be concentrated on reducing the wear in Mr. John Q. Public's car. We are all working to this end. The discussion on wear really centers around how this can best be done.

There seems to be good agreement on the facts of wear, but there is disagreement on the explanation for the facts. For example, it is agreed that engines running in continuous operation exhibit abnormally low wear, and that low temperatures can increase wear. It is agreed also that a lot of engine start-ups increase wear. Wear during intermittent operation is much higher than wear during continuous operation.

There is no doubt that low-temperature corrosive wear can occur. However, we do not believe that it causes the bulk of the wear in the average field engine. Lubricants which are effective in reducing low-temperature corrosive wear in the laboratory are not always effective in reducing wear in the field. The particular field test in which this information was obtained was run under conditions which should be conducive to low-temperature corrosive wear. The operation was in low-temperature, stop-and-go type service in northern New Jersey, and the vehicles were parked outside night and day, both summer and winter, to insure severe low-temperature conditions. It is felt that if low-temperature corrosion was an important factor in this field test, a reduction in wear should have been obtained with the oils known to be effective for combatting corrosive wear in laboratory engine tests.

We think that a good part of the wear in the average car engine occurs immediately after the engine is started. Recent data obtained by the Esso Laboratories show that start-up wear is much higher than running wear. For example, the wear obtained in the first 15 min after start-up may be as great as that obtained in over 500 miles of operation under continuous running conditions. It is found that the start-up wear is high even if the engine is held at temperatures above the dewpoint (160 F) during the shutdown. It seems possible that at least part of the high start-up wear is due to the lack of lubricant during the first stroke or two of the piston. On the first stroke the dry surfaces are torn and roughened. Considerable running time is then required to heal the roughened surface. The wear rate drops as the metal wears smooth again until a well-lubricated surface is obtained.

The engine wear symposium has stirred up a lot of interest on the wear problem. This is a healthy sign. The many viewpoints presented should serve to stimulate additional research in this important field. If past history is any criterion, this research should lead to the development of even better oils from a wear resistance standpoint than those now available.

ORAL DISCUSSION

Question: Has any attention been paid to wearing qualities of steel or other metals?

Mr. Ellis: In my company they are paying attention to things other than steel and cast iron.

Mr. Kavanagh: We have used radioactive methods and found that technique reasonably effective.

Question: These three oils that you describe – A, B, and premium – how would they be classified by the new API classification system?

Mr. Kavanagh: Every test oil contains adequate materials to pass engine requirements of MIL-O-2104.

Question: If the water temperature goes up, but the oil temperature stays down, does that add to the wearing qualities?

Mr. Ellis: The reactions that take place occur in the cylinder zone. The pressures involved are required to be high, and can only be high under conditions of the compression process. Nothing happens to the oil itself in the crankcase.

H. G. Braendel, Wilkening Mfg. Co.: In this discussion, wear has been the most serious problem. We are limiting piston-ring wear and cylinder wear, and have been so successful that we are eliminating our replacement business. The sales of the replacement sets have been 30% lower during the past year. In another 5 years, discussions on cylinder and ring wear will be a thing of the past.

J. P. Stewart, Socony-Vacuum Laboratories: The type of wear which was found to cause overhauling of units was abrasive wear, and only 10% was corrosive. A good part of the test work was done in the low-temperature range. How much does the reducing wear shown in the illustrations affect that general situation?

Mr. McArthur: Our feeling is that the type of wear which occurs in the field is high-temperature wear, and field results seem to bear that out. The wear that is important is high-temperature wear.

Mr. Jackson: The two oils had the same base stocks and the same viscosity. There were no controls on how the cars were driven, and on trip length limits. The majority of our cars showed overhaul reduction which could only be explained on a corrosion basis.

Mr. Raviolo: There are three big steps. A large, substantial step is when you include a partial-flow filter in the lubricating system. The effect of a full-flow filter was four or five times as great as those, but there was no difference in oil, or oil change procedure.

Mr. Braendel: We have been eliminating temporary dry spots by making piston rings as conformable as possible.

Automobile Batteries— Selection, Service Life, New Developments

L. E. Wells, Electric Storage Battery Co.

This paper was presented at the SAE Summer Meeting, Atlantic City, June 3, 1952.

THE SAE has always been interested in the performance of storage batteries, and through its rating standards and life test requirements has continually sought to furnish the automotive user with a more suitable and dependable battery. Being cognizant of this fact, it occurred to us that readers would be interested in a presentation of several developments which the battery manufacturers have recently brought forth to improve the qualitative characteristics of the battery, meaning performance during life, and the quantitative characteristics or length of life.

This paper includes a brief discussion of cranking capacity improvements as effected by design changes, increased availability of active material, and increased separator efficiency. The major discussion will cover improved life characteristics as effected by new types of separators, newly developed grid alloys, and the use of an electrolyte of lower specific gravity.

The purpose in presenting this material is to illustrate a few facts concerning these new developments which may be of some help in the selection of a dependable automobile battery.

Cranking Capacity Improvement

(A) *Varying Plates in Same Cell Size* – The selection of an automobile battery is primarily based upon the starting requirements of the motor and, rightly so, since it is the most important job the battery has to do. It was for this reason that over the years the battery manufacturers have concentrated on building more and more cranking capacity into their product. This has been accomplished in several ways. First, changes were made in internal cell structure to provide as much plate

THE scope of extensive research devoted to storage battery improvements during recent years is emphasized in this paper, together with a comprehensive review of results achieved to date. In an effort to produce an optimum battery, research has been directed to the qualitative factors, leading to improved performance during the life of the battery, and to quantitative characteristics which would lengthen its life.

The author points out that manufacturers have concentrated on building more and more cranking capacity into their product, accomplishing this by decreasing plate thickness, increasing separator efficiency, and by the use of improved active material. He reviews the factors which limit life under cycling and overcharge conditions, and surveys the research which has led to the use of corrosion-resistant alloys to correct electrolysis.

The Author

L. E. WELLS (M '37) is chief engineer of the Electric Storage Battery Co. He attended the Case School of Applied Science, and began his career with the Williard Storage Battery Co. in 1917. During World War I he served with the United States Army Motor Transportation Corps in France.

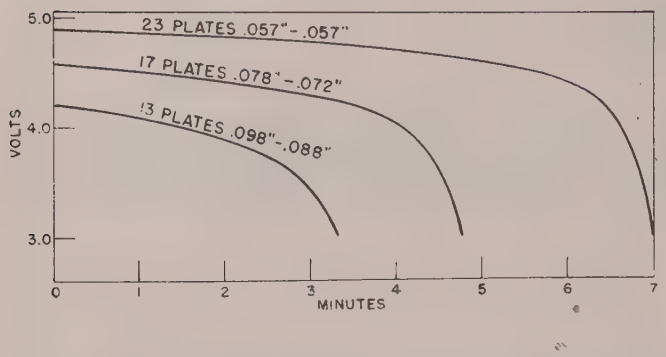


Fig. 1 – High-rate volt-time relation for varying plates in the same cell size

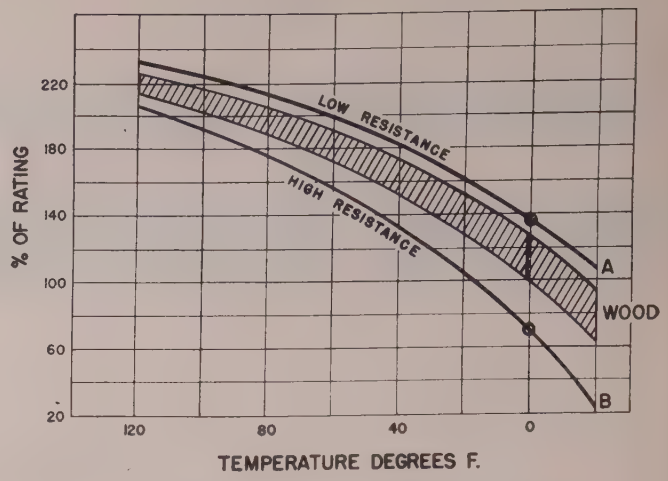


Fig. 4 – Influence of separators on time – 300 amp-min

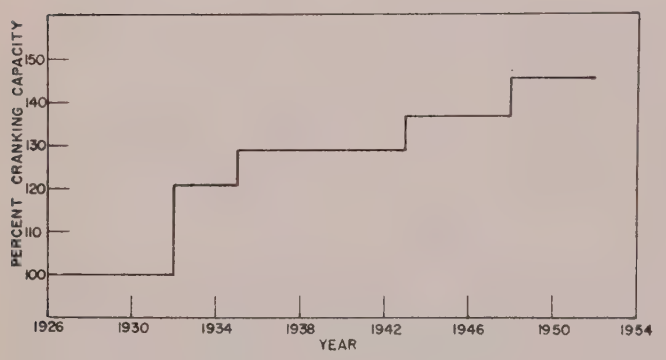


Fig. 2 – Increase in rated cranking capacity due to improved active material

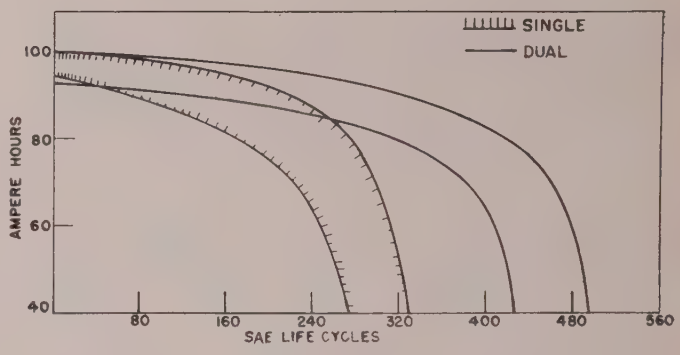


Fig. 5 – Single versus dual insulation – SAE life cycle test

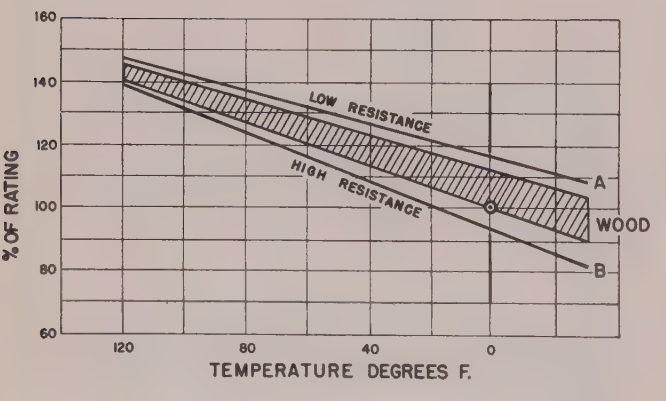


Fig. 3 – Influence of separators on voltage – 300 amp-5 sec voltage

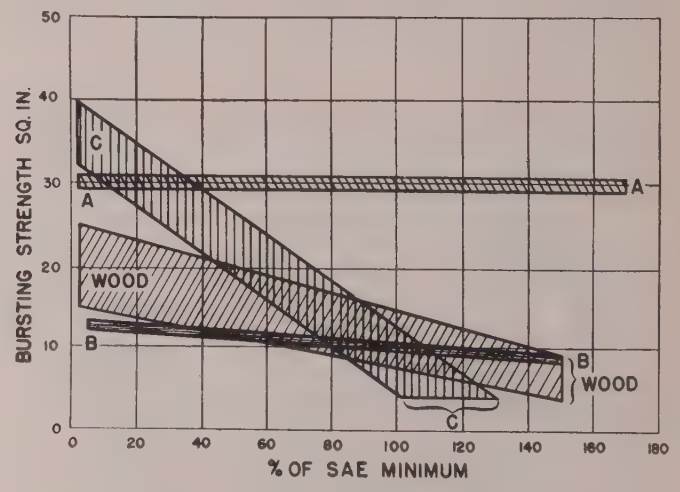


Fig. 6 – Bursting strength during life

area as possible. By decreasing the plate and separator thicknesses they have succeeded in putting a 23-plate element in the same space occupied by a 13-plate element. An example of the improvement in cranking capacity at the standardized 300-amp zero SAE test is shown in Fig. 1. These volt-time curves illustrate that it is possible by physical changes alone to increase cranking well over 50%.

(B) *Improved Active Material* – Improvement in paste formulation, which produced active material having greater low-temperature high-rate availability, has also had a marked effect on cranking capacity improvement. This improvement is illustrated in Fig. 2, which shows the increased rated cranking capacity developed in a 15-plate element during the last 20 years, while all physical dimensions were held practically constant. The total increase in rated capacity is 46%.

(C) *Increased Separator Efficiency* – There are a variety of newly developed separator materials coming on the market. Many of these are still in the laboratory test stage, while others have progressed in the field where a few appear extraordinarily good. Some of these new types of separators can be manufactured with a low-resistance value, thereby improving battery cranking characteristics. Fig. 3 gives a comparison of 300-amp, 5-sec voltage performance at various temperatures, of a low-resistance separator, one of the high-resistant type, and a wood separator. At 0 F the new low-resistant types might average 10% higher voltage.

Fig. 4 shows the advantage in minutes of cranking time obtained with a low-resistance separator when discharging at the 300-amp rate. It can be seen that the advantages of the low-resistant separator become greater as the temperature is lowered. It was only by the use of these new separators that we were able to build batteries to meet some of the severe low-temperature government requirements. This was also true in several cases at normal temperatures, where the discharge amperes were very high and the 5-sec voltage requirements difficult to meet.

Life Characteristic Improvement

(A) *Life-Limiting Factors* – Several of the techniques employed to furnish the improved cranking characteristics had an adverse effect on battery life performance. Thinner plates, for example, are less resistant to overcharge, and increased open active material offers more opportunity for shedding. Table 1 shows the outstanding factors that limit life under cycling and overcharge conditions.

(B) *Single versus Dual Insulation* – It is characteristic during cycling life that there is a tendency for the positive plate to lose its active material. One means of improving cycling life is the use of an inactive material retainer against the face of the positive plate, along with the normal separator. The effect of this dual insulation on SAE cycle life of a 100-amp-hr battery is shown

Table 1 – Life-Limiting Factors

Cycling	Overcharge
Positive material	Positive grid
Separator	High gravity
High gravity	Separator
Contamination	High temperature

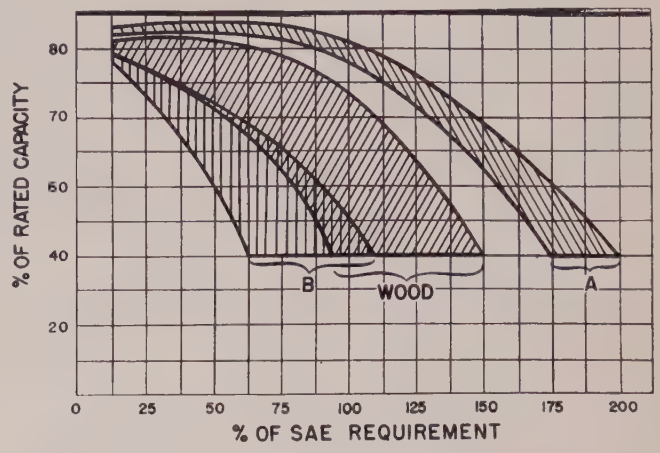


Fig. 7 – SAE cycle life

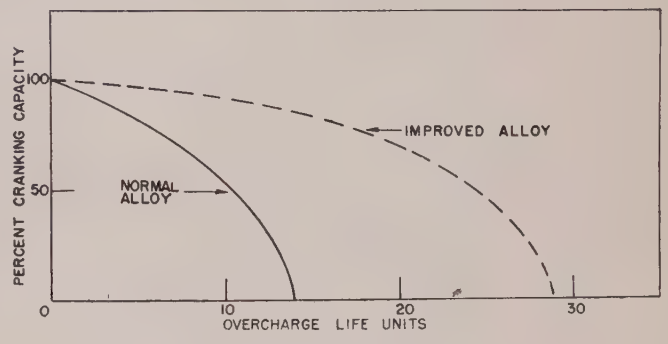


Fig. 8 – Effect of grid alloy on overcharge life

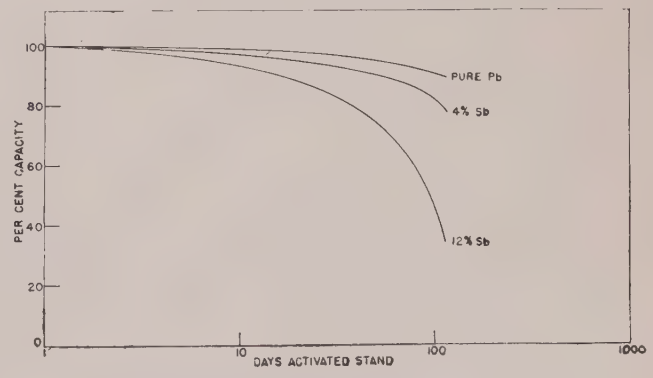


Fig. 9 – Performance versus stand time. Effect of grid alloy – 1.280 sp gr at temperature of 80 F

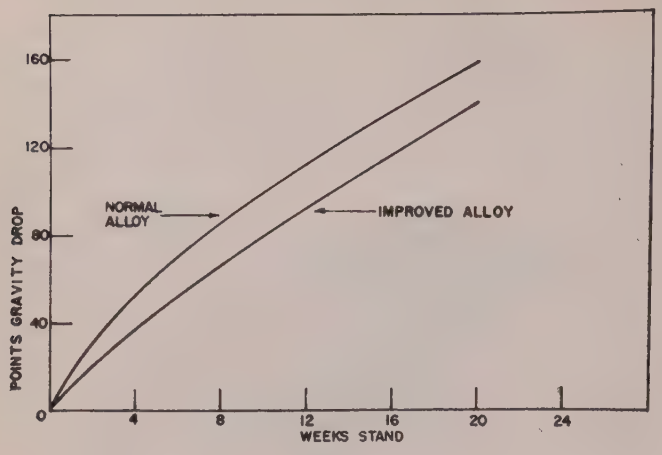


Fig. 10 – Comparative stand loss of normal versus improved alloy

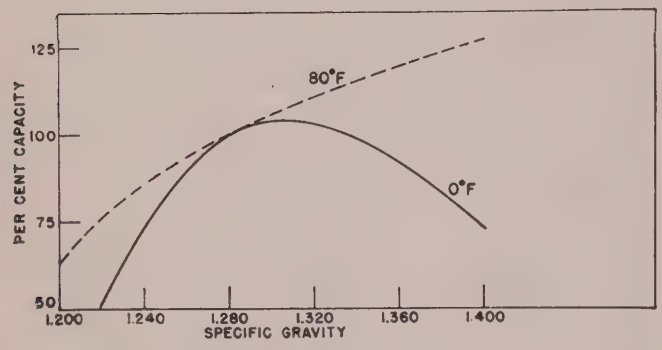


Fig. 11 – Cranking performance versus specific gravity

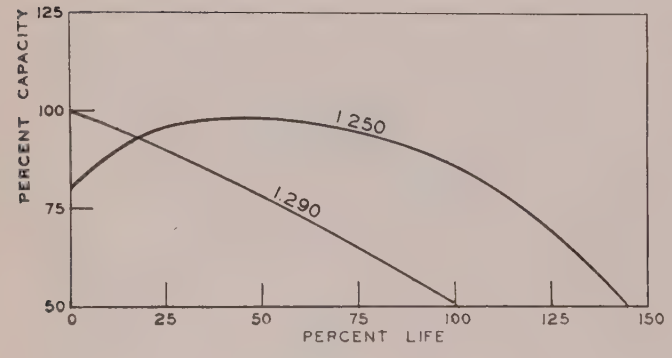


Fig. 12 – Effect of specific gravity on cranking capacity

in Fig. 5. The data shown in this graph indicate that we may get an increase of 30 to 50% in cycling life. It should perhaps be pointed out here that the added retainer will be of little value in overcharge life.

(C) *Improved Separator Life* – Most of the newly developed separators have as their primary aim longer life batteries. It remains to be seen whether this is true of all of them. Some of the new types are practically indestructible. Some are extremely resistant to acid and are flexible and tough throughout life.

Data relating to separator bursting strength during SAE life are shown in Fig. 6. From this it can be seen that type A, a plastic separator, maintains a uniformly high bursting strength throughout life, whereas type C deteriorates rapidly. Type B has a much lower wet strength than type A, but it does have uniformity throughout life.

It is believed that separator A would be the most satisfactory in vibration testing, where uniformly high mechanical strength is necessary.

Capacity during SAE life is shown in Fig. 7. Here again we see the life stability of separator A insuring better capacity performance throughout life. Wood, following standardized treatment, is not consistent in its ability to withstand the peroxidation and expansion brought about by cycling life.

This increased capacity shown during life by separator A is important evidence as to its chemical stability. Long immersion in acid at the operating temperatures encountered in the SAE life cycle have not released any components to chemically hinder normal plate functions.

Grid Alloy

(1) *Overcharge Performance* – A great deal of research has been done in grid alloys to improve their resistance to corrosion, that is, electrolysis due to overcharge. Batteries are now on the market containing these corrosion-resistant alloys. In order to demonstrate the value of this resistance to overcharge in terms of available cranking capacity during life, let us take a look at Fig. 8. Here we see that the cranking capacity of batteries with normal alloy grids begins to fall quickly, and within a period of 14 weeks is completely depleted. On the same type of overcharge test, batteries with the improved alloys give 29 weeks, or over twice the performance.

(2) *Stand Performance* – These newly developed improved alloys also have the property of reducing the self-discharge of the battery, that is to say, to improve its charge-retaining ability while standing idle. Fig. 9 is a generalized curve showing capacity loss on stand for three grids – one of pure lead, one with 4% antimony content, and one with 12% antimony. The purpose of these improved alloys is to approach the stand loss of pure lead and at the same time provide a grid metal of sufficient

mechanical strength for economic battery manufacturing.

(3) *Gravity Drop – Normal versus Improved Alloy Batteries* – Perhaps a better way to illustrate the advantages of the new alloy in reduction of self-discharge is to compare the gravity drops during stand of several batteries containing the normal alloy, with several containing the improved alloys, all in an electrolyte of 1.260 sp gr at a temperature of 95 F. This relationship is illustrated in Fig. 10. From this data it can be seen that batteries containing the improved alloy grids have considerably less gravity drop, and as a result will need less charge to keep them at good operating levels of charge. From Fig. 10 it can also be seen that after a normal 2-week stand period the batteries containing the improved alloy grids show 38% less gravity drop, after one month 31% less, and after 2 months 22% less.

Effect of Various Electrolytes

(A) *Cranking Performance versus Specific Gravity* – The automotive engineers have made marked improvements in engine starting characteristics, and the lubricating engineers have provided oils with much lower viscosity at cold temperatures, so that the actual starting job for a given horsepower engine is much easier today than it was 20 years ago. During this same period improvements were made that allowed better performance to be obtained from batteries of the same ampere-hour rating. It followed logically that the battery manufacturer could perhaps sacrifice a small amount of initial cranking capacity by a reduction in specific gravity of the electrolyte, and thereby gain in life performance.

Fig. 11 shows the cranking performance of various operating gravities. The optimum appears to be around 1.280, where low temperature is concerned.

(B) *Effect of Specific Gravity on Cranking Capacity During Life* – Even though there is a slight loss of initial cranking performance with 1.250 sp gr as compared to 1.290, it is significant that the 1.250 gravity battery improves its cranking performance during life, whereas in the 1.290 gravity battery the cranking ability deteriorates rapidly. This is illustrated in Fig. 12 where we see that after 25% of life the 1.250 battery begins to show better cranking ability. We also see that the 1.250 gravity battery will give up to 40% more life to a given cutout point. Widespread experiments are now being conducted in the field by different manufacturers that tend to verify these results.

(C) *Life versus Specific Gravity* – The relationship of life versus operating gravity for both cycling and overcharge life is shown in Fig. 13. The solid line represents the effect of operating gravity on cycling life and the broken line the effect on overcharge life. It will be observed that they both follow the same downward trend with

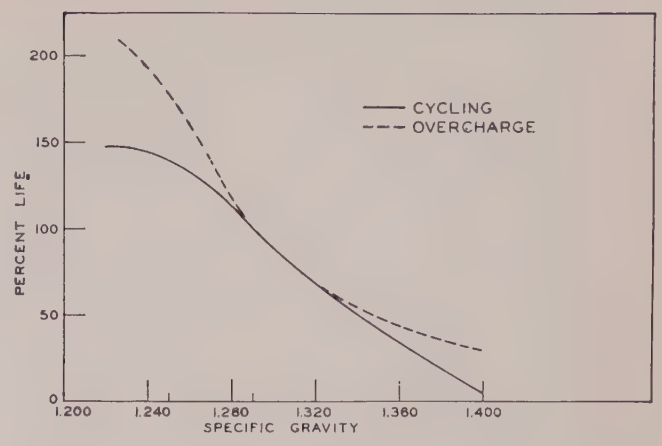


Fig. 13 – Life versus specific gravity

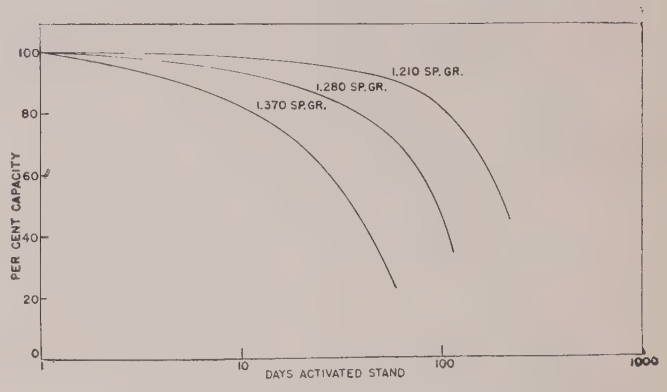


Fig. 14 – Performance after activated stand at temperature of 80 F

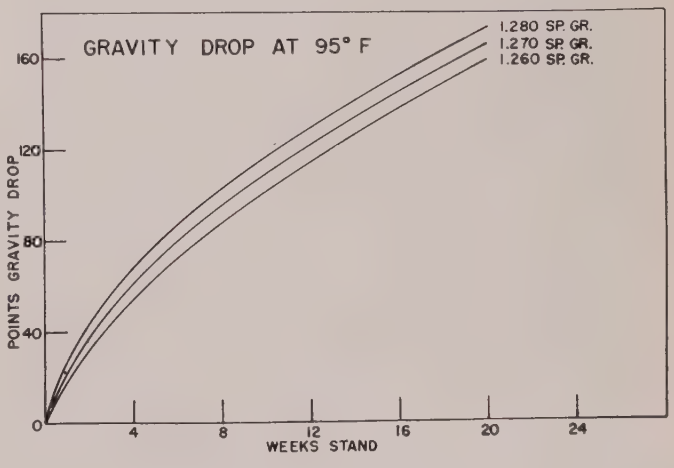


Fig. 15 – Gravity drop after activated stand

increasing gravity. This data indicates that maximum life cycle and overcharge life performance are obtained with 1.220 operating sp gr.

(D) *Performance after Activated Stand in Various Gravities* – Another important effect of specific gravity, or acid concentration, on the life performance of batteries is its function in the self-discharge characteristic. At any given temperature, the lower the specific gravity the less loss in capacity will be observed during idle or stand periods. This is shown in Fig. 14, with three widely separated specific gravities, each gravity being suitable for its own field of operation. The 1.370 is particularly suitable for special limited electrolyte volume construction and also one-shot operation.

(E) *Gravity Drop after Activated Stand in 1.280, 1.270, and 1.260 Gravities* – As a final note on the effects of gravity on standing loss, we would like to show the results of some recent experiments conducted on normal alloy batteries with three probable automotive operating gravities. These data (shown in Fig 15) indicate that for a drop of 20 points in gravity (1.280 to 1.260) we can expect a 25% improvement in stand loss characteristics at the standard 2-week period. After 1 month, 1.260 shows 20% less gravity drop than 1.280, and at 2 months, 15% less.

From the illustrations showing specific gravity-performance relationships, it is seen that a compromise must be made between the long-life characteristics of 1.220 gravity electrolyte and the maximum low-temperature cranking performance of 1.280. To date, all things considered, the most satisfactory compromise appears to be in the region of 1.250 to 1.260.

Summary

We have endeavored in this paper to remind you of several new battery developments and to illustrate in a small way just how they function to the advantage of battery users in regard to both qualitative and quantitative performance. Some of these improvements may have seemed small, but when you consider their cumulative value plus the fact that they work 24 hr a day for the battery user, you then realize the magnitude of the steps which battery manufacturers have taken to produce an optimum battery.

DISCUSSION

Improved Materials May Aid Trend to Thinner Plates

– L. J. Heine
Mack Mfg. Co.

AS a manufacturer of trucks and buses we use a large variety of engine sizes, both gasoline and diesel and, of course, we find it necessary to use a large variety of battery sizes to provide proper cranking performance. We have been very conscious and somewhat concerned about the trend to thinner battery plates shown by Mr. Wells in Fig. 1, since our field reports over the years have shown a consistent decline in battery life. This is particularly true in the postwar version of the 8D group where our experience indicates that at least for our applications too much life has been sacrificed to gain extra low-temperature cranking performance. Therefore Mr. Wells' paper has been very interesting to us since it indicates that the battery manufacturers have been conscious of this problem, and apparently have done considerable research which should soon be reflected in improved battery life.

Of course, we fully realize that the battery is only one unit of a vehicle's electrical system, and in very many cases of short battery life the blame must be put upon the voltage regulator or generator. If the voltage regulator performed satisfactorily under all conditions there should be no overcharge failures, and if the generator were optimum for the application there should be very little battery cycling.

In this connection we feel that there is much room for improvement in voltage regulator performance, particularly in the smaller types that we employ on some of our lighter trucks. It is now necessary to adjust these too often to compensate for changes in temperature and driving conditions. Since our experience shows that this procedure is often neglected, the result shows up in a complaint of inadequate battery life, practically always due to overcharge.

Since the war we have available at least for bus applications d-c generators and alternators that develop practically their full output at engine idling speed. This has greatly minimized the amount of cycling that the batteries on these vehicles were heretofore subjected to. Of course, we also find that the penalty for improper regulator adjustment on these vehicles is greater since the high current output of these machines can cause a battery to deteriorate quite quickly.

Another factor we have found that adversely affects battery life in some of our vehicles is the inherent resistance unbalance that exists in the 12/24-volt series-parallel circuit that we must employ to crank our larger diesel engines. In this circuit one battery is connected directly to the voltage regulator while the other battery has the resistance of the series-parallel switch contacts plus some extra wiring in series with it. This means that the first mentioned battery has a tendency toward short life due to overcharge. This condition has, of course, been aggravated by the thin plate batteries that have been available to us, and by faulty voltage regulator maintenance.

In conclusion, while battery life depends on many factors beyond the control of the battery manufacturer, we believe the trend toward thinner plates for commercial batteries has gone too far in sacrificing battery life, and Mr. Wells' paper has been highly gratifying to us since the improvements in materials that he describes should enable us to regain some of the ground that has been lost.

Plastic Suggested to Improve Battery Case

– H. G. Steigerwalt
Supplee-Wills-Jones Milk Co.

IN recent years we have all witnessed the increased demands made on the battery by:

1. Automotive manufacturers.
2. Fleet users.
3. Military needs.
4. Motoring public.

New items of electrical equipment such as window-lifts, air-conditioning blowers, and even refrigeration, added to the long list of accepted items such as radios, heaters, signaling devices, and numerous lights required on commercial vehicles, have all imposed severe loads on the battery and electrical system. Whether the battery manufacturer can continue to meet this demand, or whether other sources of auxiliary power such as hydraulic drives taken from the engine directly, will supply the power for all this auxiliary equipment, will depend upon the ability of the battery manufacturers to carry out practical application of their research findings.

In his paper Mr. Wells implied that all of the developments and improvements noted are being incorporated into standard battery production. Since the qualities of specific gravity, cycle life, service life, and output capacity are in varying conflict, and since service requirements vary considerably in the field, the question arises whether it might not be wise to supply a larger coverage of batteries designed to meet specific rather than average needs. Apparently some fleets have found it necessary to build their own batteries in order to obtain satisfactory service life.

Another consideration, although not touched upon by Mr. Wells, is the matter of case strength. Due to the high incidence of battery removal and handling in a delivery fleet, the percentage of repairs and replacements necessary due to broken cases, accounts for approximately half of the total battery cost. With the recent developments in the field of plastic chemistry, it would seem that a more durable case could be devised. I am thinking of either an acid-resistant plastic case, or a metal case suitably protected.

Batteries with Fewer Thick Plates Unobtainable

– F. K. Glynn
American Telephone & Telegraph Co.

EFFORTS to obtain batteries with fewer thick plates have not been successful since the war. Once we were able to obtain a battery with fewer plates, but found that its construction was the same as that of thin-plate batteries, except that some plates had been omitted. We do not need the starting flash that seems to appeal to the average passenger-car driver, but we would like longer life.

Quality of Wood in Separators Not Consistent

– T. L. Kendall
Delco-Remy Division, GMC

AN important reason for the widely varying life of batteries of identical design is to be found in the wood separators. Wood makes a good separator, but the quality is not as consistent as we might like.

Honesty of manufacture – that is, a real effort to keep batteries uniform, seems to be as important as design in determining battery life.

Praises Scope of Research On Battery Development

– B. F. Jones
Autocar Co.

FEW of us realize the tremendous amount of research and forward thinking that has been given to battery development; it takes a paper such as has been presented by Mr. Wells to bring forcibly home to us the true facts. As engineers we more or less take it for granted that the battery will do the job required, in spite of larger engines, heavier oils, and the multitude of lights and other electrical gadgets that go into the vehicle of today. At the same time we squawk about the amount of room required for the battery installation. I venture the guess that if battery development had stopped in the early '30's we would now have to carry twice as much battery weight, and utilize at least 2 to 2½ times the space now required.

At the same time it is certain that with all the advancement that has been accomplished up to the present time, men of Mr. Wells' type are still not satisfied with the present mark, but are pushing forward to obtain thinner plates, greater cranking capacity, lighter weight, and longer-lived plates and separators for their future product.

The lightweight batteries of the 24-v systems now being used in army vehicles represent some of this advanced thinking.

Future developments will surely be along increased battery life and reliability as part of the general advance in automotive progress.

Author's Closure To Discussion

THERE are several items, relative to battery life, that have been presented in the discussion of this paper. Mr. Heine's remarks are of particular interest in that they emphasize that the battery is only one unit of a vehicle's electrical system and the blame for short life can often be placed outside the battery. The comments of Mr. Steigerwalt concerning case strength are important. Considerable short life in the past, particularly during the years of acute material shortages, has been due to container failures. This problem has also been receiving attention from the battery manufacturers.

The author is in agreement with Mr. Kendall's point that uniformity in manufacture is an important factor in determining battery life. In connection with Mr. Glynn's comment, it is the hope of the battery manufacturers that these improvements will satisfy most of the long-life requirements so that it will not be necessary to revert to a less efficient starting design.

Mr. Jones has aptly presented the future trend of storage battery research. It will certainly be along the lines of providing batteries with thinner plates, greater cranking capacity, and lighter weight. It is also certain that any future endeavor to improve these qualities will be accompanied by a like effort to improve life characteristics.

Survey of Engine

THE young student contemplating graduation into the industrial world often asks the question, "What are the opportunities out in industry?" The questioner often shields his secret concern that all the problems, or nearly all, have already been solved by the vast research organizations operated by industry. He wonders, "Is there anything left to be done by the young cub engineer."

My purpose is to suggest to you a few unsolved problems concerned with the diesel engine. I hope I can stimulate your thinking into efforts which may bring solution to such problems. As a beginning I would direct your attention to the combustion system.

An insight into the position of the diesel engine combustion problems may be gained by referring to two leading authorities on the subject. Lewis and von Elbe of the Bureau of Mines, in a review of papers bearing on combustion subjects from 1948 to 1950, write[1] as follows with reference to the present status of the diesel engine combustion process:

"It is now widely recognized that the problems of the diesel engine concern the engineer rather than the chemist, and it is not surprising, therefore, to find a much smaller number of papers devoted

[1] See *Industrial and Engineering Chemistry*, Vol. 43, September, 1951, pp. 1925-1941: "Combustion," by Bernard Lewis and Guenther von Elbe.

THE author states that his purpose in presenting this paper before the Student Branch is to suggest a few unsolved problems concerning the diesel engine. His effort to stimulate student thinking into channels which may lead to acceptable solutions has been summarized in four pertinent points:

1. Stressing the importance of mixture formation to the end that each fuel particle, when properly prepared for the chemistry of combustion, may find its necessary equivalent of oxygen as quickly and conveniently as possible.

2. More definite controls for heat extraction of regions subject to thermal fatigue in combustion-chamber envelope.

3. Better knowledge of distortion control to give added life to critical parts required to channel heat or seal pressures.

4. Need to develop a real supercharger preeminently matched to diesel field operating conditions.

The Author

C. G. A. ROSEN (M '37) is consulting engineer, Administrative Department, Caterpillar Tractor Co., Peoria, Ill. He graduated from Cogswell Polytechnic College and the University of California with B.S. and M.E. degrees. Mr. Rosen was SAE vice-president representing the Diesel-Engine Activity in 1940, and is an active member of the Fuels and Lubricants Activity Committee. He was also a member of the SAE War Engineering Board.

Combustion-Chamber Envelope

C. G. A. Rosen, Caterpillar Tractor Co.

Paper presented at SAE Student Branch, Purdue University, Lafayette, Ind., March 20, 1952.

to diesel-engine combustion by comparison with spark-ignition combustion. Papers that have appeared within the period covered by this review are concerned with survey and testing of currently available fuels for high-speed diesels, studies of factors affecting self-ignition, such as chemical composition, influence of inert gases, the effect of ethyl nitrate, the relative merits of additives, such as nitrates, peroxides, ethers, and hydrocarbons versus pilot injection, particularly for cold starting, the practice of pilot ignition, and the effect of combustion on depth of penetration of liquid fuel droplets, measurements of temperature and radiation by means of pyrometers are reported; pressure indicators are described that serve to detect speedily any disturbance in the operation of the engine such as uneven loading or piston leakage."

On the other hand, Martin A. Elliott, also of the Bureau of Mines, has this to say:[2]

"In the early stages of the development of the diesel engine the chemical phase of the process did not receive as much attention as the physical phases because, after all, it was possible to build successful diesel engines without a detailed knowledge of the chemistry of the process of combustion. At the present stage of the development, the law of diminishing returns seems to be functioning, and the problems to be solved are becoming progressively more difficult. We should, therefore, bring to bear on these problems, not only physical or mechanical techniques, but also chemical techniques."

[2] See paper presented at 1947 National Oil and Gas Power Conference by Martin A. Elliott. Published by ASME in "Diesel Fuel Oils," 1948.

So even among the experts there are differences of opinion concerning the unsolved problems in diesel combustion.

It is not my purpose to make this discussion a combustion symposium, but rather to follow the strict phraseology of the subject announced. I merely wish to select a few pertinent factors which are influenced by the combustion-chamber envelope.

A. As a chemical retort.

B. As a heat exchanger.

C. As a sealed pressure vessel.

When I speak of the combustion-chamber envelope I mean the composite of the following elements:

1. The lower contour of the cylinder head, with its valves, pockets, and chambers facing the combustion gases.
2. The inner cylinder surface and its jacket.
3. The piston crown and its sealing rings.

An engineer is significantly interested in the combustion-chamber envelope as a means to promote mixture formation. When we speak of mixture formation, we might mean either the formation of a suitable fuel-air mixture before ignition, or the setting up of conditions favorable to mixing fuel or products of thermal decomposition with air after ignition has occurred. These two phases of mixture formation are being given considerable attention at the present time. In some developments it is significant that the first phase has been given maximum attention, namely, the formation of a suitable fuel-air mixture before ignition with all fuel delivered before ignition, and then from there on to the control of the nucleus of ignition, in order to provide systematic combustion reac-

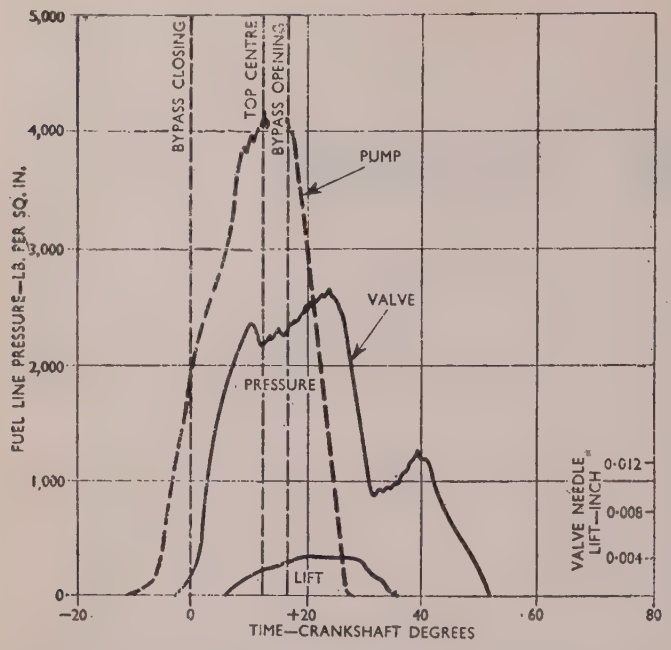


Fig. 1 – Injection characteristics designed for fuel system to provide suitable spray pattern, with flat check-type valve in fuel pump and open-nozzle fuel valve

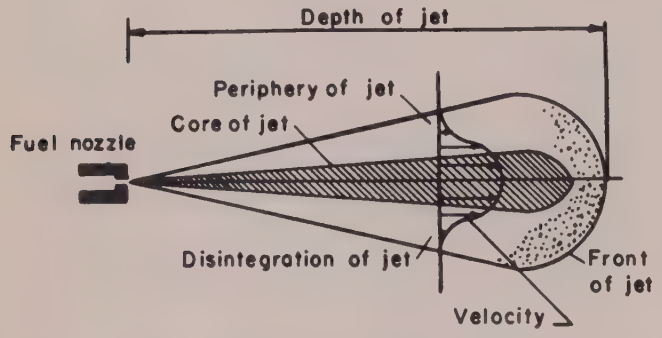


Fig. 2 – Schematic view of fuel spray pattern

tions to yield a smooth-running and efficient engine. We all have a qualitative appreciation of the difficulties of properly mixing fuel and air in the short time available in the high-speed diesel engine. However, the air requirements for different paraffin hydrocarbons can be cited as an example. In the case of the complete combustion of cetane, which is a 16-carbon-atom paraffin, about 117 volumes of air are required for each volume of hydrocarbon vapor. We can now begin to see why the mixing problem is difficult in the diesel engine. If we carry this computation a step further we will find that an individual drop of liquid cetane must be mixed with about 736 times its volume of air in an engine under conditions existing at the end of compression, in order to complete its combustion. These very elementary calculations, based only on the stoichiometry of the combustion reactions and the perfect gas laws, furnish ample evidence of the difficulties we might expect in mixing diesel fuel and air.

The passage of fuel through the injection system of a diesel engine is the first step along the road to combustion. This is an important step, and the design and operation of the injection system, Fig. 1, to yield correct spray patterns is a profound influence on combustion and on subsequent behavior of the fuel after it leaves the nozzle. The first contact between fuel and air is made by a continuous jet of liquid fuel moving at high velocity. The disintegration or atomization of this jet is the first stage of mixing fuel and air.

The more important fundamental factors affecting the disintegration of this continuous jet into drops, are the velocity of the jet, the frictional

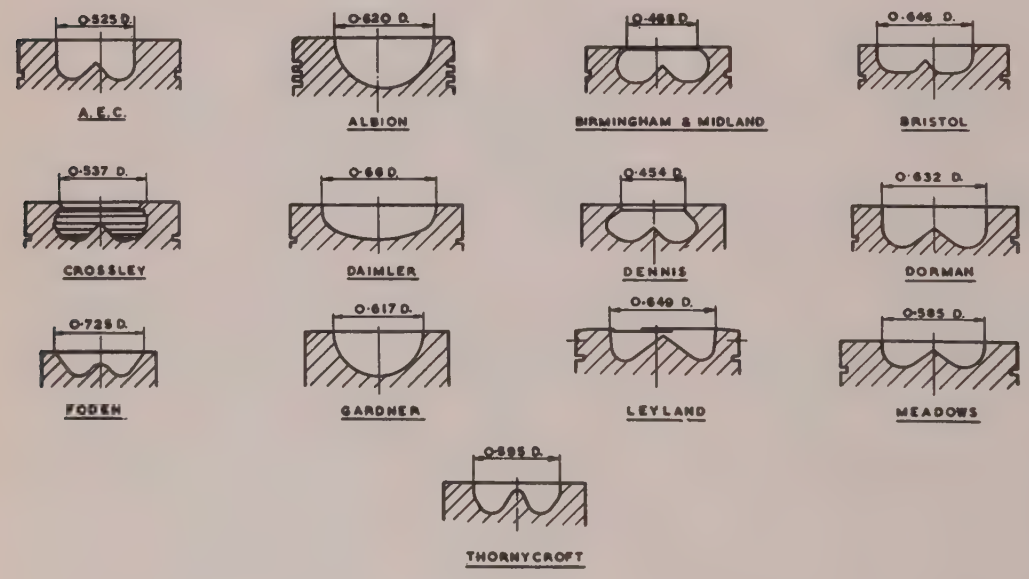


Fig. 3 – Direct injection combustion chamber with toroidal type of contour in piston

drag at the interface between the jet and the air, the density of the air, and the surface tension of the fuel. Analysis shows, Fig. 2, that the core of the jet consists of fuel moving at high velocity. At 4000 psi injection pressure, the mean spray velocity is about 800 fps. Around this core is a zone or mantle containing drops of fuel mixed with air. The velocity is highest at the center and decreases to zero at the interface between the zone of disintegration and the ambient air. In the vicinity of this interface where the velocity of drops relative to air is low, further mixture of fuel and air is accomplished.

Before such mixing occurs, concentrations of fuel in air in the disintegrating jet, range from zero at the periphery to 100% at the center. The disintegration of the fuel jet accomplishes the first mixing of liquid fuel and air. The final mixing of fuel vapor and air is accomplished by vaporization of drops having significant velocities relative to the ambient by reason of organized or general turbulence created by the design of the engine combustion chamber and by indiscriminate turbulence occurring in localized regions.

In making a brief summary of the physical process of mixture formation, we can state that a jet of liquid fuel traveling at velocities of several hundred fps is broken up into drops of the average order of 10 to 20 microns in diameter. Under the conditions in the diesel engine, drops of this size will vaporize in about one to two milliseconds. The mixture of fuel vapor and air occurs as a result of vaporizing drops moving relative to the air and as a result of the turbulence in the combustion chamber. This is of interest in considering the influential factors which the combustion-chamber contour has on mixture formation. In the British vehicle diesel engine using direct injection with the deep toroidal type of combustion chamber in the piston crown, some points of interest are worthy of discussion.

Fig. 3 shows a series of these combustion chambers. Generally the combustion-chamber diameter in the piston crown is from 50 to 60% of the cylinder bore. The depth of the center is about 25% of the cylinder bore. In this scheme the air introduced through the inlet valve is given a rotating movement by suitable shrouding of the valve or the port at right angles to the axis of the cylinder, that is, the center of the helix of the airflow from the inlet valve is coincident with the cylinder centerline. On the compression stroke, the air is forced into the cup as the piston reaches the top center, and the velocities are multiplied in inverse proportion to the diameter of the combustion chamber. Theoretically, the transformation of the air from a rotating mass having a diameter equal to that of the cylinder, into a mass of smaller diameter equal to that of the combustion chamber, should result in velocities proportional to the inverse of the square of the two diameters. However,

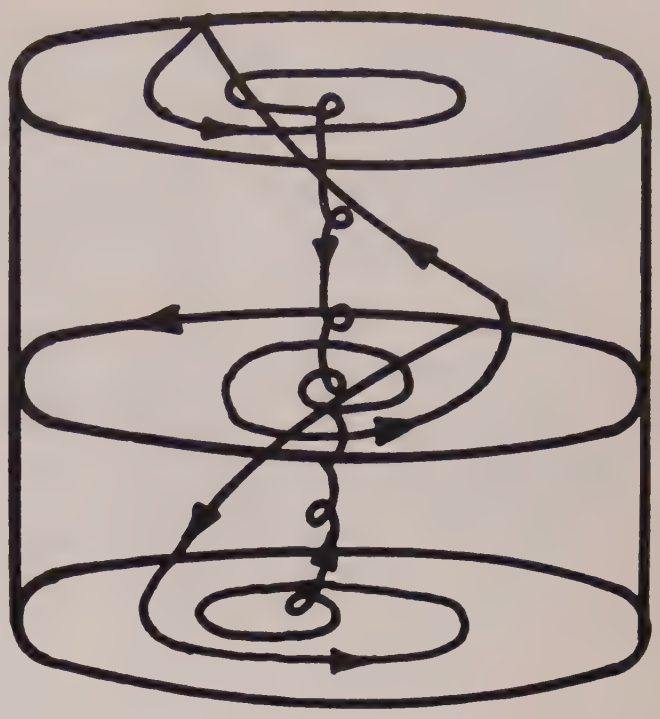


Fig. 4 – Turbulence pattern in direct injection toroidal type of combustion chamber

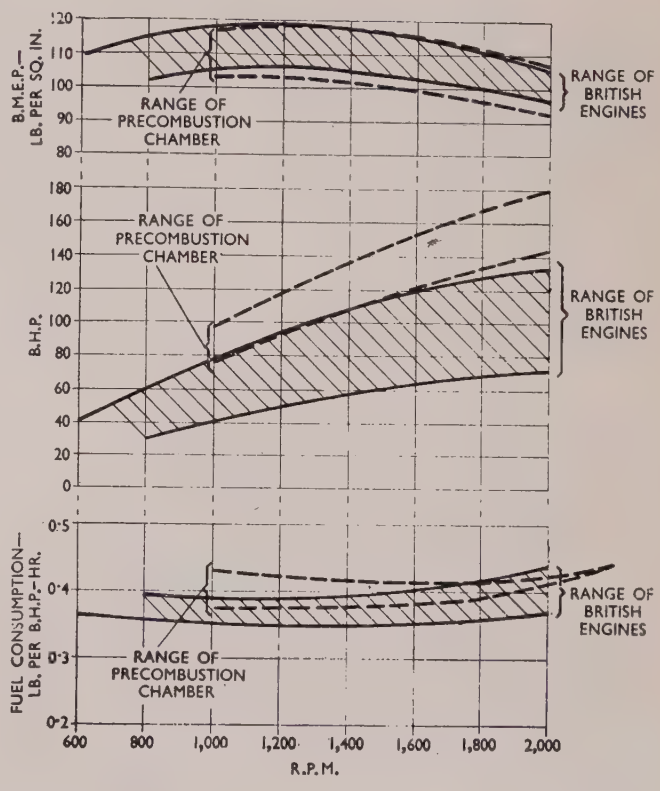


Fig. 5 – Comparative performance curves of direct injection and precombustion chamber engines

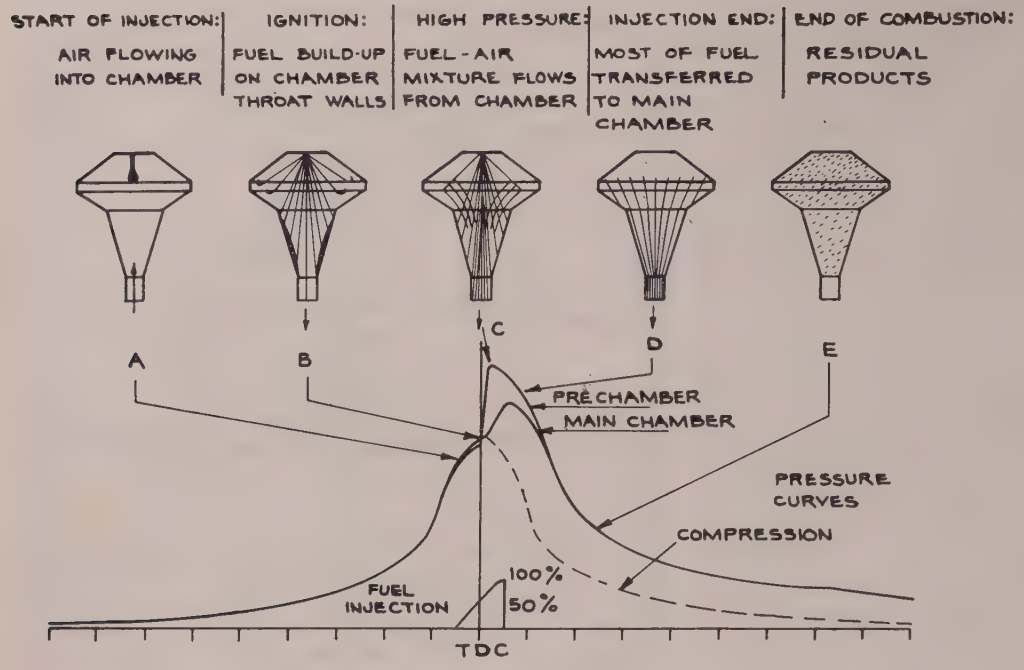


Fig. 6 – Scheme of combustion in precombustion chamber engines

in the engine considerable losses accompany the transfer, and the increase in rotational speed is much less than the theoretical. The actual speed is more nearly a direct relationship to the inverse of the two diameters.

The piston is moved to a minimum clearance between the cylinder head and the piston which is about 0.040 in. This final action of squeezing the air out from between the piston and the cylinder head has been called squish. The path of the air traveling into the combustion chamber is thereby changed by a superimposed current of air moving out from the space between the upper projection of the piston crown and the cylinder head. In addition (Fig. 4) to the radial movement of the air there is a superimposed secondary movement which creates a double spiral within the combustion chamber. The face of the cylinder head and the face of the piston during the compression stroke offer friction to the movement of the air adjacent to such faces, and thereby slows down the centrifugal force of the rotating gases. The center of the bowl of the combustion chamber, however, retains essentially its maximum velocity. The combination, therefore, of the slowing effects on the two faces, one at the bottom of the piston bowl and the other at the face of the cylinder head, is such as to cause a movement of air from the center of the bowl outwardly and down to the lower face of the bowl; this then flows inwardly and moves upward toward the center of the axis of the bowl. For this reason the center of the bowl of the combustion chamber is usually peaked to assist in the inward flow of the turbulent moving air into the center vortex of the cone of rotation.

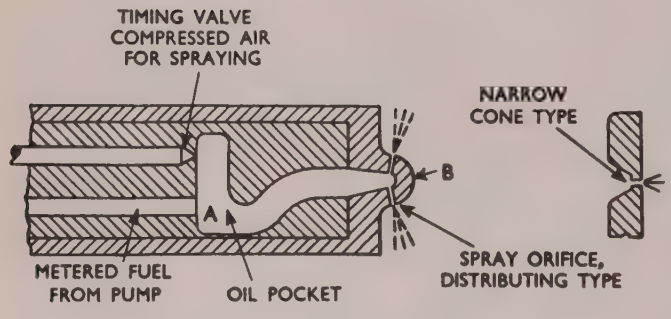


Fig. 7 – Open-nozzle type air-injection fuel valve

The action of the vortex movement is to turn the mass of air inside out in a smooth and orderly manner, and brings within the range of the fuel sprays, air which otherwise would not have been reached by them.

The efficiency of this method of mixture formation is clearly revealed in Fig. 5, which shows the

range of British vehicle engines in their capacity to produce peak bmeps of 120 psi, and fuel consumptions of 0.35 to 0.36 lb per bhp-hr at the lowest point in the consumption curve, also a very flat fuel consumption curve with colorless exhaust.

By comparison, the modern type of precombustion-chamber engine has equal capacity to produce high output by another effective means of mixture formation. The fuel consumption, however, is not as good as the open-chamber type of engine, as a fluid friction penalty is paid to retain the ability of a prechamber engine to idle smoothly and continuously under low temperature, low load, and low speed operation.

The dynamics of combustion in a precombustion-chamber engine is illustrated in Fig. 6. Here, for the sake of simplification the pressure cards are idealized.

At point A fuel injection into the precombustion chamber starts, and at point B approximately one-half the fuel is injected at full-load operating conditions. At this point also, the fuel ignites by virtue of the elevated temperature of the compressed air. Fuel injected before ignition will be distributed in the air of the precombustion chamber and some of it deposited on the throat walls of the chamber. The throat, therefore, acts as a collector for the heavy fuel spray during the early stages of injection. Later this fuel is scrubbed off and dispersed by the high velocity resulting from the ignition in the precombustion chamber. This situation is quite analogous to that of the open-nozzle type of air-injection system. After flow has been established from the precombustion chamber through the throat area, the remainder of the fuel is shredded off the conical surface and passes out into the main combustion chamber through the medium of a high-velocity gas stream.

The construction of the precombustion chamber is such that the highly atomized fuel is forced into a region where ignition takes place. The ignition of the fuel in the burner-tube portion of the precombustion chamber establishes a higher pressure within the chamber. A pressure differential between 100 and 200 psi then develops between the precombustion chamber and the main chamber, depending upon engine-operating conditions. High-speed Schlieren photography has indicated that the velocity of the issuing stream from the burner tube is initially 500 to 750 fps, depending upon the injection characteristics and timing. Peak velocities may be as high as 1000 to 1200 fps, when the pressure difference reaches 200 psi. This situation is illustrated at C. At point D the pressure differential still carries air and remaining fuel out of the precombustion chamber. At point E the pressures in the two chambers are equalized and very little material remains in the precombustion chamber. Occasionally, from D to E, pressure surging is observed under some conditions. Effective combustion in the main chamber is completed at the crank-angle position illustrated at point E. Remaining combustion is mostly a cleanup or exhaustion of earlier combustion byproducts. The next, Fig. 7, shows the close similarity of the precombustion-chamber process to air injection.

From the foregoing, it is clear that the rate of fuel injection, ignition delay of the fuel, precom-

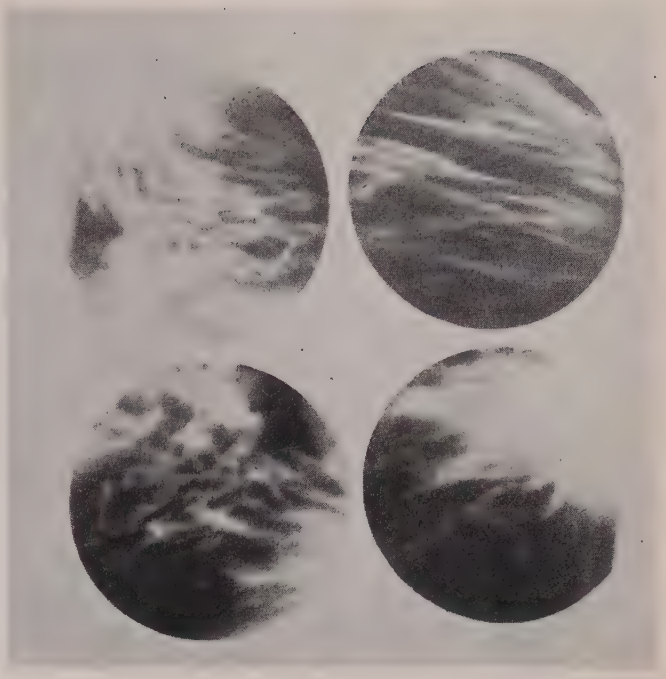


Fig. 8 – Photograph of burning fuel droplets shortly after initiation of combustion in prechamber

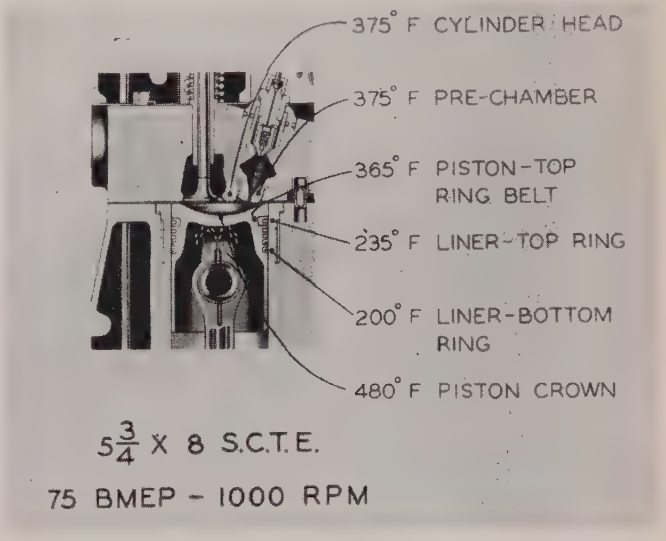


Fig. 9 – Survey of temperatures in combustion-chamber envelope of precombustion-chamber engine

Fig. 10 – Photograph of fuel being burned on initial combustion in prechamber and main chamber

bustion-chamber geometry, and orifice characteristics, are all instrumental in delivering the fuel into the main combustion chamber. The rate of combustion reactions occurring in the prechamber determines the fuel-air mixture which emerges from the precombustion chamber.

Some idea can be gained regarding the state of affairs within the precombustion chamber at the initiation of ignition. Fig. 8 by Landen shows that the burning fuel droplets are sometimes well-

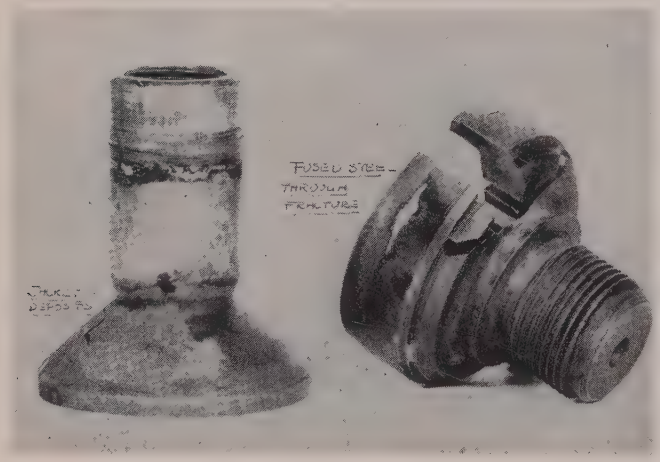


Fig. 11 – Burner tube ruptured by shock waves in precombustion chamber

organized in definite channel patterns, whereas, at other times confusion results from the high turbulence of air currents emanating from the narrow orifice in counterflow to the movement of the fuel droplets in the precombustion chamber.

The physical aspects of evaporation and mixing the fuel with the air have been evaluated by Wentzel. The rate of heating and evaporation and time of evaporation, given in Table 1, are all important. A droplet with an initial radius of 0.002 in. in an atmosphere of 752 F at a pressure of 34 atmospheres will reach 625 F in about 0.00105 sec, or a drop of 127 deg in air temperature, while in an atmosphere of 1022 F at a pressure of 27.8 atmospheres, it will reach 735 F, or a drop of 287 deg in 0.00057 sec. The gas oil used in these calculations had a mid-boiling point of 460 F at atmospheric pressure. NACA experiments showed that fuel injected into heated gases under pressure resulted in a pressure and temperature drop to the air. In the precombustion-chamber engine the average temperature drop of the entire air of the prechamber is about 20 deg F depending upon the amount of fuel evaporated before reaching the throat of the precombustion chamber.

This is an important point which can be ascertained by referring to Fig. 9. Here a single-cylinder test engine indicates the average critical temperature points of the piston, cylinder heads, rings, and so forth. It can be seen that temperature in the throat area of the prechamber is exactly the same as that of the cylinder head (375 F).

It is often expected as an average assumption that the prechamber throat area would achieve much higher temperatures. This is true in some engines, but is not the case of the Caterpillar precombustion-chamber engine. The reason for good idling at low ambient temperature and light load is the fact that the average temperature drop of the small volume of air in the precombustion chamber, due to the introduction of fuel, is about 20 deg, as mentioned above. In the open-chamber type of engine the introduction of fuel under idling or light-load conditions when the cylinder wall temperatures are low, causes a respectable temperature drop (as indicated previously) of the ambient air, many times that reached in the smaller confined volume of the precombustion chamber. In this way it is often possible to obtain missing or ragged idling in open-chamber types of engines, because of the large volume of air which is affected in the exchange of heat to the fuel particles engaged in surface evaporation in the combustion chamber. The opportunity for the droplets to be properly ignited and, even more important, to sustain burning under such critical operating conditions, is rendered difficult.

The flow of fuel from the precombustion chamber into the main chamber is illustrated in Fig. 10. Here a small volume of fuel has been burned in the prechamber and the initial entry of fuel has been made

into the main chamber, showing a small spurt of flame. The significant feature in this photograph is that the stream into the main chamber shows a definite shock wave. This shock wave is of considerable importance in the study of diesel fuel characteristics. A prechamber engine might well be a very valuable piece of instrumentation for the further study of those factors which create shock waves and ultimately cause distress of piston rings and piston-ring grooves. An example of the destructive effect of the shock wave is shown in the next illustration, Fig. 11. Here a burner tube has been ruptured which subsequently permitted the flow of high-temperature gases which melted the ruptured surface. The initial stress causing this rupture must have been of the order of 30,000 psi or over. At the maximum peak pressures measurable in the precombustion chamber the greatest combined stress which can be computed is that of 9000 psi. The superimposed shock waves must have created the pressures necessary to create this rupture, which is of the order of 3⅓ times that of the maximum gas pressure attained.

In pursuing this study even further, it can be shown that the comparison of the pressure-time diagram of combustion with the curve denoting rate of pressure rise is of value in determining the presence of shock waves. Fig. 12 shows the means of computing an arbitrary shock wave factor. Values above 1.5 have been chosen from experience as those producing undesirable shock waves by their proved detrimental influence on piston-ring breakage and ring-groove wear in the engine. Normal good operating conditions show a shock factor of about 0.5. This method appears to offer better possibilities of determining combustion roughness than is now obtained by comparing rates of pressure rise per deg of crank angle.

The flow of fuel from the precombustion chamber into the main chamber, when treated as an air-injection phenomena, indicates some intriguing possibilities in obtaining high output. Combustion studies by Dr. Landen illustrate the sequence of events (shown in Fig. 13) of the flow of fuel and air from the orifice of the precombustion chamber into the main chamber, with maximum temperatures of 3800 F. These Schlieren photographs, taken at speeds of 8000 frames per sec, show the emergence of the fuel at the central exposure of A. The similarities which prevail between air injection and the precombustion-chamber engine are significant.

This is true as far as the combustion phase is concerned in the main combustion chamber. The particular combustion chamber shown in this figure is not ideal for the purpose of obtaining maximum utilization of the air of combustion in providing intimate contact of air with all the fuel particles presented from the spray emerging from the precombustion chamber. In all the photographs, the upper sectors of each side of the spray cone show voids where the fuel particles are not mixed to the extent necessary to promote the maximum utilization of air in the combustion chamber. The photographs were taken in an apparatus which permitted the taking of good Schlieren photography as illustrated in Fig. 14. This is a one-shot combustion engine and is so arranged as to provide the means of obtaining the factors essential to establishing a more thorough knowledge of the sequence of combustion.

Mention has been made of the influence of the combustion chamber on shock waves as related to fuel characteristics. Of further significance is the influence of specific combustion-chamber design and its profile on top ring belt temperatures in the

Table 1 – Time Factors in Sequence of Combustion Reactions – Initial Portion of Cycle

Beginning of	Crank Angle, deg btc	Time, sec
Injection	9	0.000
Initial combustion	5	0.00056
Luminous combustion of carbon formation	3	0.00084

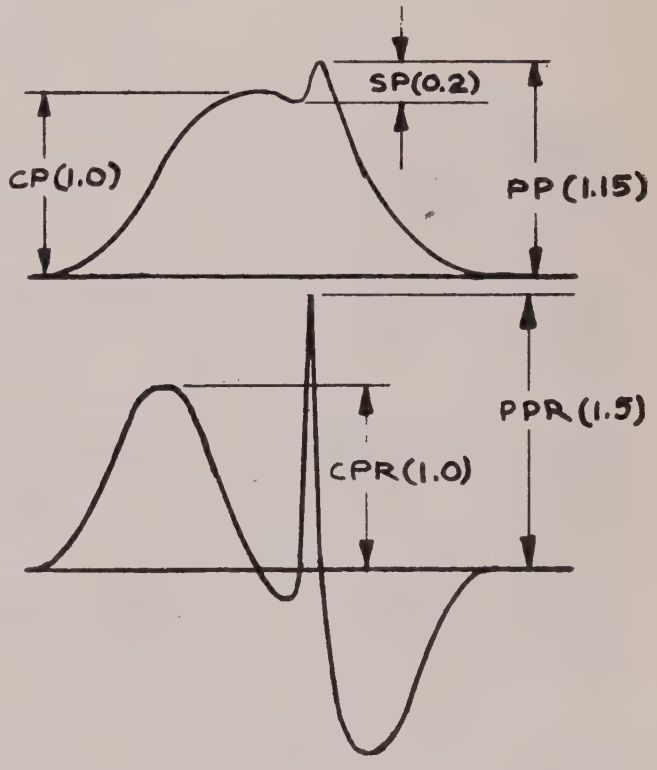


Fig. 12 – Analysis of combustion shock phenomena

Table 2 – MAN Diesel Developments

Description	Medium Bore, Low Speed	Small Bore, High Speed	Medium Bore, Medium Speed, High Supercharge
1. Bore	12	3½	12
2. Stroke	18	4¼	18
3. Speed	325	1800	428
4. Compression Ratio	15/1	18/1	13/1
5. Bmep	80	80	213
6. Maximum Cylinder Pressure	850	1060	1700
7. Excess Air Ratio	2.00	2.00	2.50
8. Fuel Consumption	0.365	0.375	0.308
9. Effective Efficiency, %	37	36	44.5
Total Losses Include the Following:			
10. Loss Due to Combustion Process Different from Theoretical, %	2.5	3	3.7
11. Loss Due to Heat Transfer through Cylinder Walls, %	5	6.3	2
12. Loss through Mechanical Friction	8	9.2	5
13. Loss through Pressure Drop	1.5	2.5	1.7

piston. Fig. 15 illustrates five different designs of combustion chambers and their attending characteristics of top ring belt temperature in relation to the quantity of fuel injected. Here it can be seen that curve A for a swirl chamber design is of a high-temperature magnitude, particularly at 2000 rpm. This indicates that the speed of the engine or the number of times the combustion period is repeated per min has a marked influence on ring belt temperature. The combustion chamber in B illustrates a lower temperature characteristic than A, but even higher than the precombustion chamber illustrated in C. Inasmuch as all of these engines are of specific design they do not reflect general practices for the combustion system illustrated, which may vary considerably from those shown in comparable combustion systems.

Engine D shows far less change in maximum top ring belt temperature with speed, whereas a 2-cycle engine in E, fitted with oil jet cooling is practically constant as to top ring belt temperature regardless of whether the speed is 1000 rpm or 2000 rpm. This piston is a malleable iron piston and is oil-cooled, whereas all of the previous pistons are of aluminum and are not liquid-cooled. These data were taken from information submitted by the Delft Laboratories of the Shell Oil Co.

One more thought with reference to the rapidity of mixture formation. In some experiments conducted by the Swiss Locomotive Works on an engine of the type shown at B, the period of time in a nitrogen-filled chamber when the full-blossomed pattern of the spray cone is changed by the squish at the end of the piston to a completely dispersed mist, in the combustion chamber, was only a quarter of a millisec or 3 deg of crank angle at 1800 rpm.

Combustion-Chamber Envelope as Heat Exchanger – In the development of high-output diesel engines the major or principal attention has been directed to the ways and means by which more air is made available to consume more fuel in given cylinder displacement. On the whole, far too little

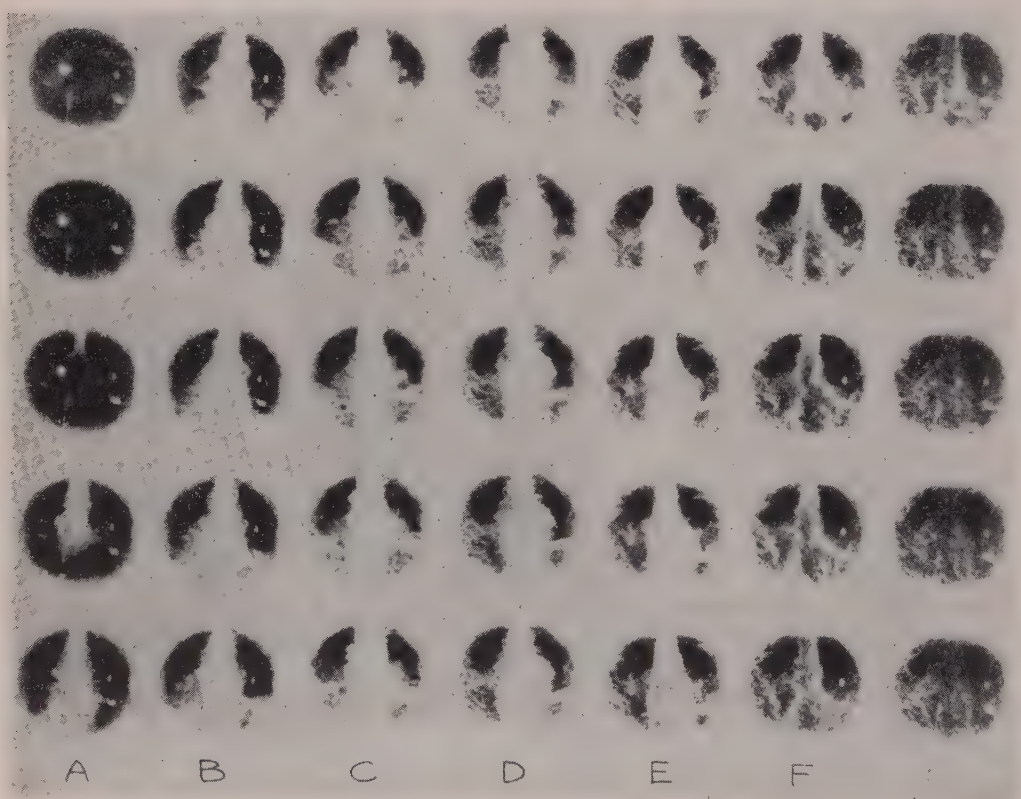


Fig. 13 – Sequence of events in combustion of fuel from the prechamber to the main chamber

concern has been given to effective means of heat extraction to critical parts, and to the lowering of the inherent losses of the piston engine system.

A most significant recent development is that of the MAN high supercharged engine reported at the International Internal-Combustion Engine Congress in Paris in May, 1951. Experiments on this unit have shown a 300% increase in output with only a 30% increase in heat flow through the material of the combustion chamber and the piston rings. A comparison of statistics, Table 2, may prove interesting, showing pertinent data on three engines of current design by MAN. Note in the last column the fuel consumption of 0.308 lb per bhp-hr at a bmep of 213 psi. Also the effective efficiency of 44.5% with the substantial reductions in friction and thermal losses.

It may be of interest to note that the friction losses vary almost directly as the square of the compression ratio. This has been found to be true in several designs of high-output engines.

The new MAN engine has some interesting features. It is supercharged by a turboblower with 5 stages in an axial-flow exhaust gas turbine and a 9-stage axial-flow compressor discharging into an intercooler before the final radial-flow stage after which the air is aftercooled as shown in Fig. 16.

High efficiency of supercharge is obtained and maximum weight of air is delivered to each cylinder at the relatively low temperature of 80 F. This is extremely important in supercharging diesel engines. Supercharging is a natural for the diesel as a diesel thrives on excess air delivered at low temperature.

However, progress in its general adoption has been somewhat hampered by the inefficiency of the blower and the difficulties in matching properly the supercharger with the demands of the engine for air in proportion to the fuel delivered.

The temperature of the air delivered to the cylinder at the end of the suction or intake stroke influences the mean cyclic temperature considerably. The terminal temperature of compression is raised 3 to 3½ times the inlet air temperature depending upon the compression ratio used. There is a broad difference in engine performance and maintenance if air is delivered to the cylinder at 80 F or at 270 F (typical cases in present-day engines). The former results in successful operation, but the latter is encumbered with thermal difficulties with pistons, rings, and valves.

The industry is in great need of something new in pressure-charging or vast improvements in present superchargers, to yield the higher outputs which customer demands will foster.

An efficiently blown diesel engine should yield a minimum of at least 100% increase in output over a normal atmospherically-aspirated engine. The case of the MAN engine is an exceptional example, but it goes without saying that no automotive diesel will tolerate a 5-stage turbine, a 10-stage compressor, and two inter-coolers, on a supercharged installation.

In engines the heat flow from gas to cooling water is mainly governed by the high resistance offered by the gas-wall interface. This is borne out in Fig. 17, which shows the effect of increasing cooling water temperature on the temperature of the liner interior and of the first ring groove. It appears that each of these temperatures rises by the same amount as the cooling water temperature, so that the temperature difference between wall or groove and water remains the same. In other words, the heat flow passing the measuring point remains the same, the heat resistance of the path from this spot to the cooling water not having changed. With higher wall temperatures heat flow from gases to wall should become slightly smaller, but the decreas is too small to be detected and for all practical purposes can be neglected.

With a given heat flow as governed by the gas-wall interface the temperature drop through a wall will be proportional to the thickness of that wall, and this constitutes by far the main reason why temperatures of cylinder walls are higher in big engines than in small. In big and medium-sized engines the temperature gradient through the cylin-

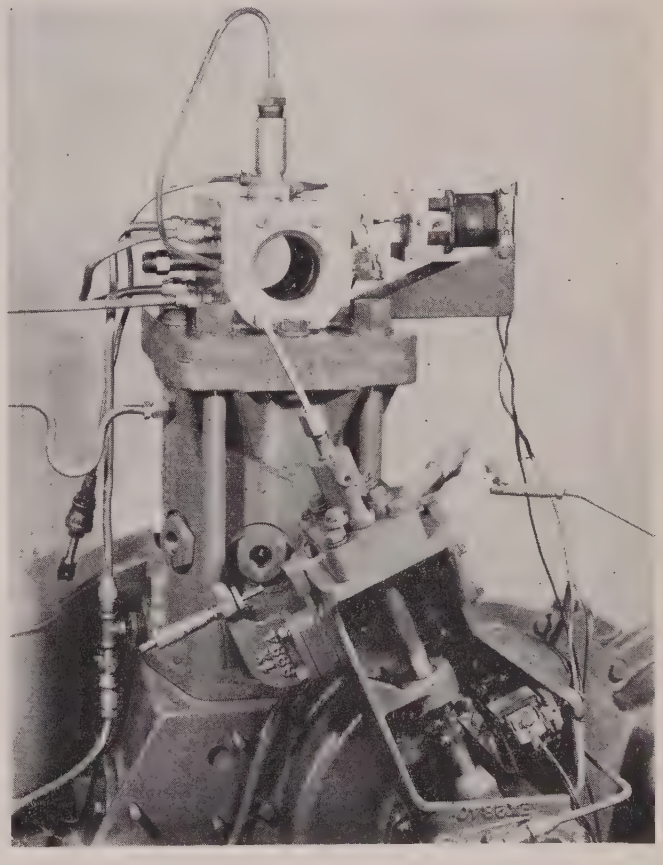


Fig. 14 – Experimental engine for obtaining Schlieren photography

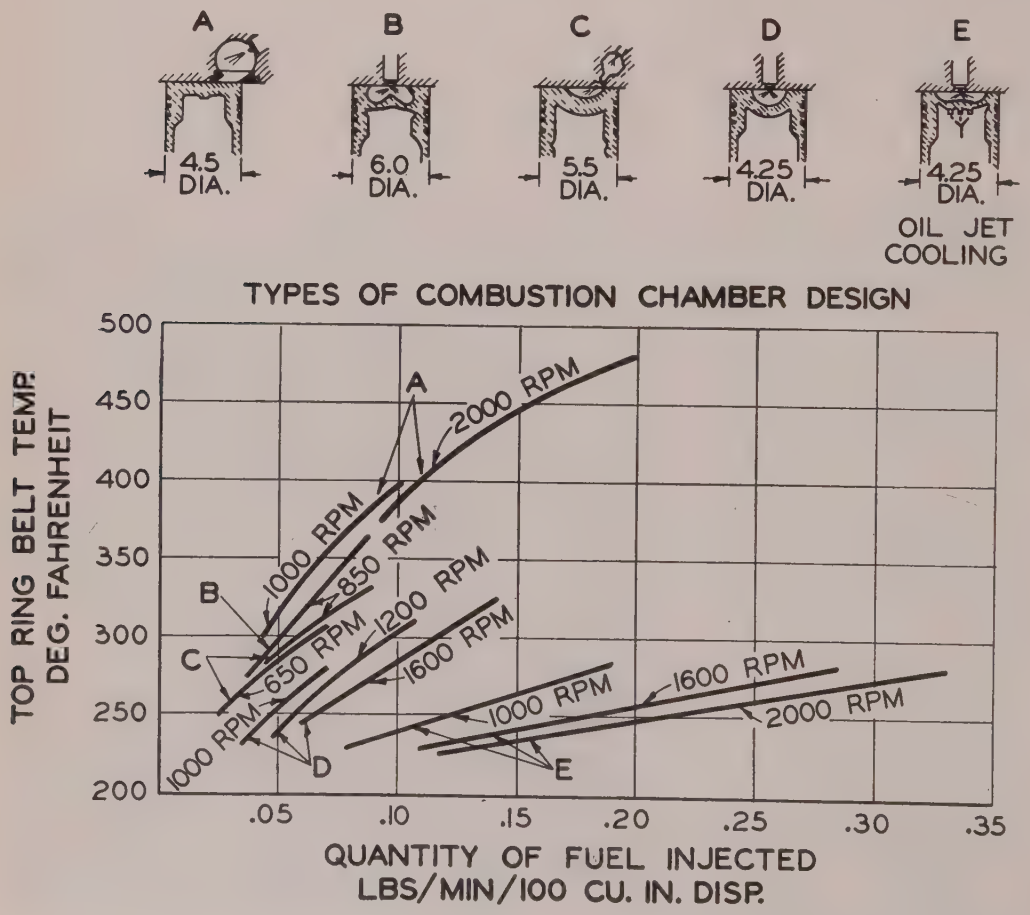
der wall can be measured by fitting thermocouples into the liner at different depths. It has been found to be of the order of 110 F per in. for fuel inputs of 0.0182 lb per cu in. per stroke.

In most engines it is hardly feasible to reduce the temperature drop through walls by making the walls thinner. However, at a given heat flow the drop can be made smaller by using a material having a higher heat conductivity. In replacing a cast-iron liner by a chromium-plated aluminum liner of equal thickness, the effect can be appreciable. The aluminum liner has been found to allow some 30% greater output for the same ring-groove temperature. Such experiments show that the temperature difference between inner surface of liner and cooling water is about halved. This scheme makes it possible to raise the cooling water temperature without the risk that wall temperatures at full load will become too high.

During the exhaust period the exhaust valve is essentially in a situation similar to that of the exhaust-gas thermometer. Its greater temperature increase, however, must be ascribed to the greater heat input during other periods, that is, during combustion and expansion as a result of the charge density. Supercharged diesel engines, however, frequently have more difficulty with inlet valve seats than with exhaust valve seats.

Piston temperatures increase with load for a given speed (gas velocities) and fuel/air ratio (gas temperatures), the heat transfer being nearly proportional to density. Heat transfer increases with load in the diesel for the reason that fuel/air ratio or gas temperature is chiefly responsible.

Supercharging – The limit to the amount of supercharging of engines is determined by piston temperature and/or rate of wear. Piston temperature more frequently is liable to become the limiting cause because its temperature is proportional to the density of the charge (Fig. 18). Reducing gas temperatures by applying intercooling or by reducing the fuel/air ratio (large air excess) then are means of reducing heat input to the piston. Avoidance of excessive gas movement to reduce piston temperature may be desirable; however, the combustion of greater quantities of fuel in air of greater density



Courtesy of Delft Laboratories, Shell Oil Co.

Fig. 15 – Influence of combustion-chamber design on piston temperatures

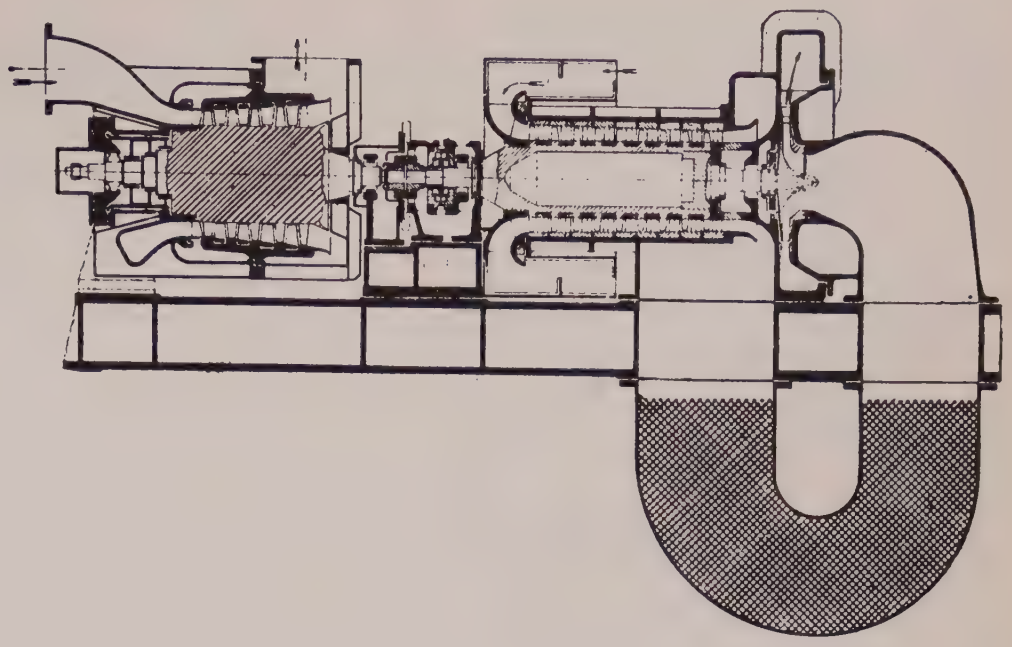


Fig. 16 – MAN multistage supercharger provided with intercooler and aftercooler

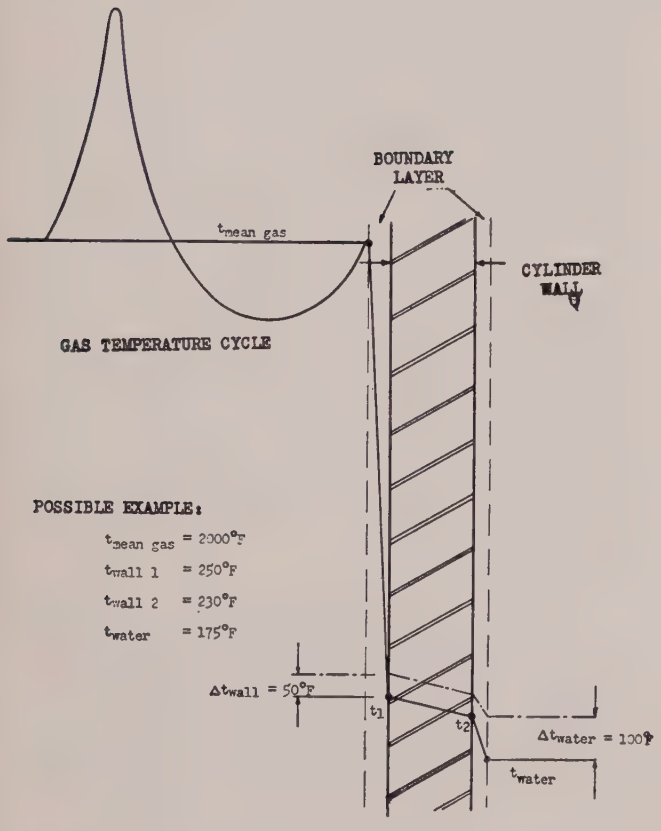
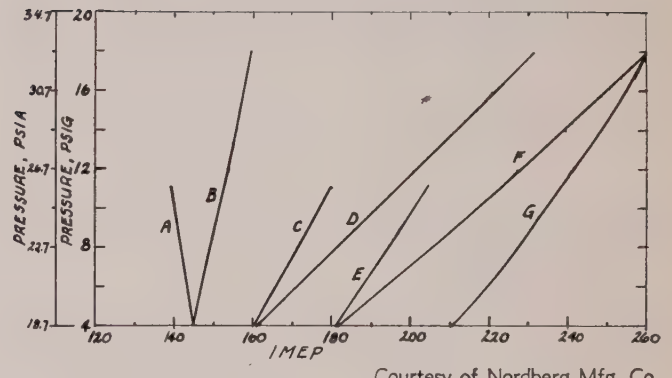
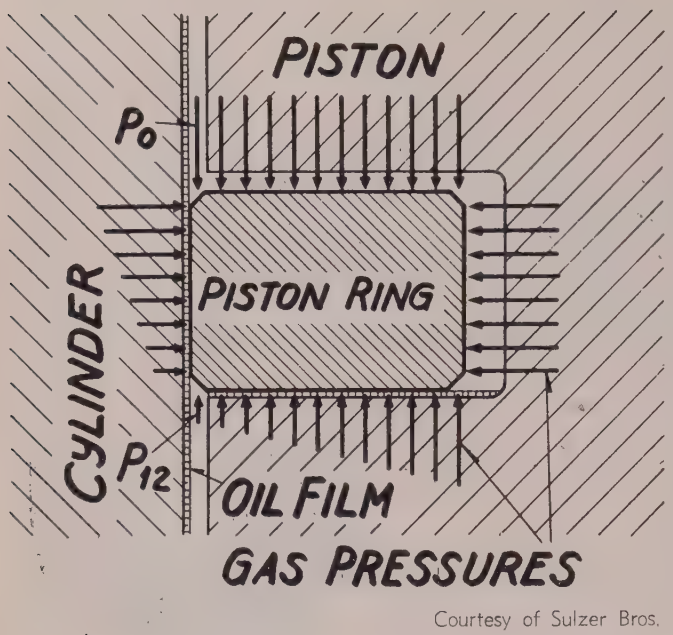


Fig. 17 – Diagram indicating relative temperatures of combustion gas cylinder wall surfaces, and cooling water



Courtesy of Nordberg Mfg. Co.

Fig. 18 – Analysis of supercharging systems relating to imep versus supercharging pressure, P_1. All curves are based on constant heat transmission to water jackets. (A) nonintercooled Buchi system $P_3 = 800$ psig; (B) nonintercooled Buchi system, $P_3 = P_1 \times 44$; (C) Buchi system intercooled to 100 F, $P_3 = 800$ psig; (D) Buchi system intercooled to 100 F, $P_3 = P_1 \times 44$; (E) Miller system intercooled to 100 F, $P_3 = 800$ psig; (F) Miller system intercooled to 100 F, $P_3 = P_1 \times 44$; (G) Miller system intercooled to 100 F, $P_3 = 1425$ psig

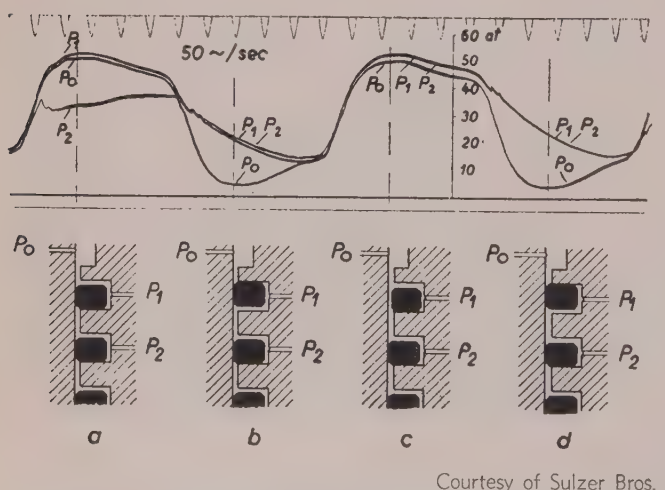


Courtesy of Sulzer Bros.

Fig. 19 – Ideal state of piston ring

calls for more gas movement than with natural aspiration.

Combustion-Chamber Envelope as a Pressure-Sealing Vessel – For brevity and simplicity I will confine my remarks to one phenomena of the piston ring. Fig. 19 shows the ideal state of a piston ring. The dynamics of piston-ring movement, as observed by Mr. Aue of Sulzer Bros., is expressed by Fig. 20.



Courtesy of Sulzer Bros.

Fig. 20 – Unstable pressure conditions as related to piston-ring position

The top trace shows the timing pips. The cylinder pressure trace (in a 2-cycle engine) is P_0. The pressure behind the top ring is P_1 and the pressure behind the second ring is P_2. At the end of compression and the beginning of combustion the left-hand diagram shows the position of the three compression rings. The top compression ring is sealing the groove on its bottom and against the cylinder wall, the two effective sealing surfaces, thus P_1 is substantially equal to P_0. Inasmuch as the top ring seal is effective, P_2 is lower than P_1.

As the gas pressure P_0 drops, due to expansion of the burning fuel, P_1 also is lowered; however P_2 may continue to maintain its pressure until its value intersects P_0 and P_1 as the traces indicate. As P_0 and P_1 continue to reduce due to cylinder gas expansion, the gas trapped between the top and second ring will expand against the pressure P_0 and ultimately cause the top ring to lift and strike the top of its groove as shown in the center diagram at B. When pressures P_0 and P_1 are balanced, the top ring is neutralized and may leave the cylinder wall.

As P_0 increases in the next compression and firing stroke the top ring will be dropped on its lower face in the groove, but P_0 may be on the cylinder face equal to P_1 behind the top ring and in this position unable to seal the combustion space, placing the burden of sealing the combustion space on the second compression ring.

On the subsequent stroke the rings may assume the relative positions as shown at C and equilibrium conditions will be once more established with the top compression ring functioning as is shown.

It is of interest to evaluate this phenomena and contemplate the state of affairs which could exist under the influence of combustion shock waves as shown in Fig. 10. If a burner tube can suffer failure under such high fluctuating pressure waves – what could be the fate of the top piston ring. These factors are of much interest in further study of combustion and the labyrinth packing provided by the piston ring setup.

In conclusion I wish to emphasize that I have attempted to stimulate thinking by referring to some of the unsolved problems of the diesel engine. I have only suggested a few pertinent ones for your attention:

1. We should stress the importance of mixture formation to the end that each fuel particle, when properly prepared for the chemistry of combustion, may find its necessary equivalent of oxygen as quickly and as conveniently as possible.

2. We need more definite controls for heat extraction of regions subject to thermal fatigue in the combustion-chamber envelope.

3. We need better knowledge of distortion control to give added life to critical parts required to channel heat or seal pressures.

4. We need to develop a real supercharger preeminently matched to diesel field operating conditions.

Fleet Management: A Job Evaluation

T. L. Preble, Tide Water Associated Oil Co.

This paper was presented at SAE National Transportation Meeting, Pittsburgh, Oct. 22, 1952.

LAST winter, colleagues in the Transportation and Maintenance Activity asked me to write a piece, tentatively titled, "Just What Is a Motor-Vehicle Superintendent." As a helpful hint they stated that it should cover "education, ability, personality, duties, relations, studies, responsibilities, standing, and so forth."

I replied that I would undertake the job. Privately, I thought that committing some ideas on the subject to cold, unretractable type might be akin to a sharp and introspective look in the mirror.

It appeared desirable to give this paper the title, "Fleet Management – A Job Evaluation." For many years, both as a representative of truck manufacturers and as a fleet operator, the writer has had the uneasy feeling that some fleet supervisors have been living in a bit of a vacuum – a negative which nature is reputed to abhor, and which operators should strive to eliminate by selling their provable good works. In great part the vacuum, if any, exists only by reason of:

First, lack of understanding by senior management as to the intricacies of, and the importance of fleet management from a standpoint both of cost and public relations, and the resultant need of employing professionals to direct the job.

Second, inability of some highly competent operators to impress management with the importance of their functions.

As a target, therefore, this paper must retain a management point of view. And as a paper appropriate to this Society, it must embody recognition of the fact that, as Past-President Dale Roeder underlined last year – "Our working orders tell us to promote the arts and sciences and standards and engineering practices connected with the design, construction, and utilization of automotive apparatus . . ."

The theme here rests largely on the word *utilization,* recognized in Chapter 8 of our constitution which, in defining qualifications for members, states in part "shall be composed of persons who, by previous technical training or experience or by present occupation, are qualified to take charge of automotive engineering work, including operation and maintenance."

The Society's bibliography on the Transportation and Maintenance Activity is extensive, and

SINCE there are nearly 22,000 fleets of trucks, comprising a total of 1,260,000 vehicles, operating in the nation, this author presents a review of management factors that should have wide appeal. Within the scope of fleet management he considers reduction of capital expenditure and operating expenses, and facilitation of executive supervision.

Equipment selection and retirement, maintenance, and vehicle operating techniques are among the topics discussed. He emphasizes the need for better understanding and cooperation between the fleet owner and the truck manufacturer, thus correlating actual field service with design and production engineering problems.

The Author

T. L. PREBLE (M '26) has been supervisor of Automotive Transportation for the Tide Water Associated Oil Co. since 1934. Mr. Preble graduated from the University of California, and saw active service overseas in World War I. His work with Tide Water was interrupted by a four-year period of service in World War II, during which he served as consultant to the War Department and the ODT and also saw active duty as a Colonel in the Ordnance Department.

replete with take-home pay. However, it contains few attempts to summarize fleet executive responsibilities from either a management or an engineering point of view. So this writer must forego the pleasant process of thumbing an intellectual hitchhike and hasten to the point as he sees it.

Management Factors

Let's take management first.

The Automobile Manufacturers Association advises us that there are about 21,700 fleets averaging 58 trucks each, a total of nearly 1,260,000 trucks – let alone passenger car, bus, and airborne fleets. Evaluating at $4000 present average truck cost, we have a total replacement investment, before depreciation, of over 5 billion, together with an out-of-pocket operating cost approximating an annual outlay of 1 billion dollars.

The above average of 58 trucks per fleet obviously does not connote the employment in all instances of highly trained supervisory personnel. Rather, it emphasizes the obligation of many (though not all) truck suppliers to supplement their amorphous advertising and across-the-desk claims of "selling transportation" by providing their customers with a more finite service in respect to prepurchase job analysis, leading to the selection of a minimum number of properly qualified pieces of equipment. This is a story in itself. Time does not permit its exploration in this paper. We must, of necessity, deal with the executive personnel of fleets of hundreds, and of thousands.

A friend and writer on economic subjects, Richard H. Lansburgh, once defined management as "the correlation of the details of operation of an enterprise so that it will work as a harmonious whole toward the desired goal." He subdivided the definition as follows: first, the establishment of major policies; second, the planning for and setting up of an organization to carry out these policies; third, the operation of the enterprise through this organization.

It is obvious that a fleet manager should be chosen with the above simple, important fundamentals in clear perspective, as an important supplement to his technical qualifications.

What should be equally obvious, but unfortunately is not in some instances, is the law of responsibility and authority expounded years ago (although certainly not discovered) by a prominent industrial writer, editor, and mechanical engineer, L. P. Alford. He said that "responsibility for execution of work must be accompanied by the authority to control and direct the means for doing the work."

This, as indicated previously, may be the fault both of top management and of the fleet operator. Clear delineation of the precise scope of the whole job of fleet management has not been achieved in some companies.

Scope of Fleet Management

It should be made crystal-clear that the scope encompasses:

1. Equipment selection and retirement, following field job analysis and cost accounting scrutiny.
2. Maintenance.
3. Vehicle operating techniques designed to reduce both out-of-pocket and investment outlay.

Advantages of Centralized Control or Coordination – That is the whole package. It must be handed to the right man and he, as a member of the management team, must be given authority commensurate with responsibility, modified in some cases, it must be admitted, by the corporation's organizational structure. Even though decentralization may be the corporation policy, as it often is, the benefits of uniform automotive policies and procedures, and free interchange of ideas may not be denied. This can be approximated, whether the automotive man's status is that of staff only, or line and staff.

The following summary presents some of the more important advantages of centralized control (or coordination) and implies, therefore, a definite procedure for the realization of such control.

Reduction of capital expenditure by:

1. Centralized purchase contracts, based on consolidated commitments and savings through quantity purchases.
2. Standardization of types of equipment for specific duties, and a consequent reduction of number of truck sizes and types used.
3. Reasonable standardization of make of equipment and consequent reduction of number of makes used.
4. Ability of central control to interchange equipment between companies or components thereof, reduce idle time of equipment, and exert better control of assignment of equipment to meet requirements of various localities.
5. Possibility of pooling transportation and maintenance activities in specific territories through coordination of activities which can best be achieved through centralized control.

Reduction of operating expenses by:

1. Elimination of superfluous overhead.
2. Districting of equipment by make of truck to simplify maintenance and operation.
3. Standardization of service policies and shop procedures.

Facilitation of executive supervision by:

1. More efficient handling of motor-transportation problems. They have become such a large, technical, and complex factor in the operation of many industries, as to demand the same type of centralized control (or coordination) that is exercised over other important departments of the business, such as merchandising and production.
2. Elimination of interorganization friction, intrigues, and controversies, through codified standard practice and clearly defined authorities.

3. Educational activities to overcome antiquated and expensive practices.

4. Standardization of cost accounting, permitting the experience of one unit or subsidiary to be cited for the benefit of other units without costly experimentation by the other units.

5. Exercise of closer scrutiny and improved judgment in replacement of worn-out or unsuitable equipment, because of broader perspective furnished by uniform cost accounting and centralized control. This ever-present problem is one demanding keen analysis and a high degree of coordination if wasteful practices are to be avoided.

6. An improved spirit of cooperation among employees due to the existence of clearer job standards and corresponding ability to compare the performance records of employees.

Confusion and, therefore, lack of clarity as to scope, and degree of control, are even indicated by the various titles given to operators; for example, equipment superintendent, transportation supervisor, automotive engineer, maintenance manager, manager automotive department, and so forth. I am indebted to Lt.-Col. W. A. McCutcheon for a letter on this subject which stated in part:

"It occurs to me, that as applied to top-flight motor vehicle superintendents, the fundamental title, Automotive Engineer, should be broadened to Automotive and Transportation Engineer, for in many cases, both transportation and maintenance are responsibilities of one person. Frankly, I have wondered why we didn't 'move in' to the transportation field long ago."

J. M. Orr of the Duquesne Light Co. has said:

"I dislike the term 'transportation superintendent'; it always implies a man who is charged with the responsibility of operating a plant selected by someone else, and who has no latitude or opportunity to put into practice beneficial measures he has learned from his previous experience to improve applications and conditions. Much psychological good would result from use of the term 'transportation manager' in its stead."

It is currently the opinion of many industrial analysts that the word *economy* soon may emerge from its uneasy retirement and arise to challenge us once again. So with that as an additional prod, let us dwell briefly upon the previously cited three elements of the "whole package" of fleet management.

Equipment Selection and Retirement

Selection should avoid rule-of-thumb procedures. Even though a like-for-like vehicle replacement is indicated, today's changed operating conditions, legal and otherwise, should be scrutinized carefully. This is especially important with respect to cargo vehicles intended for largest possible carrying capacity, where the owner's aim is optimum economy – or in the case of for-hire carriers, earning power. Such vehicles are often so "tailored" in respect to dimensional and axle specifications as to inhibit maximum efficiency.

Careful preselection job studies not only will clarify the above problem but also will result in the most effective compromises (and I believe that word necessarily applies in many instances) in respect to horsepower, axle ratios, tare weight, and other elements of importance.

Skilled vehicle selection has received increasing and very beneficial study in recent years – but one goal has received too little comment, namely, that of reduction of capital investment, a matter of high importance to corporation executives. Improper selection, as well as unintelligent utilization, contribute to an imprisonment of capital which well could show a greater investment return elsewhere in the corporate picture.

Similarly, vehicle retirement deserves intensive study. It is a difficult problem, loaded with variables, which I doubt will ever be solved completely either by educated guess or by mathematics. The operator must use his soundest judgment in retirement studies (as in vehicle transfers) with overall economy in mind. That somewhat vague term, overall economy, refers specifically here to: (a) operating expense; (b) investment account; and (c) to a constant realization that both (a) and (b) are cash out of the corporate till.

One last word on this subject. Total transportation cost is, of course, vehicle cost (controllable and fixed) plus driver cost. I believe that in many instances since World War II driver cost is rising at a greater rate than vehicle cost. Analysis may disclose individual instances where total costs may be somewhat reduced by the purchase or assignment of a delivery vehicle larger than formerly was deemed necessary. The idle time of such vehicle will be increased, but the driver may return to his base much sooner, there to devote his time to other work, such as warehousing, and so forth.

Maintenance

That detailed attention to the ever-changing art of maintenance, and to a maintenance organizational structure properly designed for the individual fleet, scattered or otherwise, will contribute to economy is self-evident.

It must suffice here to emphasize the necessity of establishing suitable standards relative to personnel and equipment and their utilization, and to underscore the benefits to owners of maintenance research on a variety of items. Manufacturers, also, will be benefited if they will examine fleet records of failure frequency, and consult with men who earn their living keeping fleets rolling with a minimum of cost and of down-time.

Vehicle Operating Techniques

You will recall that Col. McCutcheon said, "I have wondered why we didn't move in to the transportation field long ago." I believe he had in mind

that whereas maintenance is an established art, and whereas increasingly effective thought is being given to equipment selection, that much remains to be developed by way of acceptable practice in vehicle utilization. In this I concur.

I believe much remains to be done in the collection of an adequate amount of operating data, in correlating such data and establishing standards thereby to the extent possible, covering loading time, running time, unloading rates, clerical time of vehicle operators, and the like. Such studies can be of great practical use in reduction of operating expense and of capital investment.

Analyses of this nature, supplemented with development of routing and dispatching procedures, are particularly valuable in delivery operations, where solicitation of the consignee's cooperation is successful. Reasonable maximum of hours of operation, the reduction of peak loads, and the hauling of a maximum load in terms of time and of distance are goals usually subject to comparative attainment. The effect on fleet size and on cost per ton-mile can be dramatic.

Also these studies can be helpful in some of the elements of equipment selection. For example, the effect of traffic congestion sometimes can be measured in terms of driver cost, and vehicles with more appropriate power and gear ratio supplied.

No discussion of vehicle operation can be complete unless reference is made to rather recently developed techniques of driver selection and training. Some of them are exceedingly effective. Such a program should be high on the agenda of fleet management objectives. Results definitely to be anticipated are:

1. Lower maintenance expense through reduction of accidents, and insurance premium cost resulting therefrom.

2. Notable reduction of the so-called "hidden costs" which are caused by driver ineptitude.

3. Greatly enhanced driver morale. The development of pride of attainment of genuine professional status as a driver is not only a highly desirable goal of management-employee relations, but will achieve such eminently practical results as faster round-trip time due to reduced roadside consumption of coffee.

4. Improved union relations, largely because of employment of drivers of superior personal and job qualifications. The percentage of rejection of driver candidates is some measure of the effectiveness of a sound program of driver selection.

I bow with real appreciation to Amos E. Neyhart, of Pennsylvania State College, and others, who some day will be acclaimed as pioneers in the development of driver selection and training practices of great and provable benefit.

Management Qualifications of Fleet Executives

You will recall from my preliminary remarks that I was requested to dwell a bit on such qualities as education, ability, personality, duties, relations, studies, responsibilities, and so forth.

In at least partial answer, let me quote in part from Thomas H. Nelson of Rogers, Slade, and Hill:

"Today 50% or more of the specifications (of a manager) have to do with the ability to plan, organize, build morale and teamwork, establish good public relations, and build for continuity – matters beyond his technical specializations.

"The specifications put increasing emphasis on his ability to get results through others. That, I believe, is the very essence of being a manager.

"Another characteristic of this new type of manager is that he thinks in terms of functions rather than merely in terms of specific operations, and specific methods, which should be the responsibilities of subordinates.

"There are five basic functions: (1) planning, (2) organizing, (3) directing, (4) coordinating, and (5) controlling."

Genuine executive ability, and all the term implies, is a management "must" in the fleet operator. I know many of these men, hold them in high regard, and have some knowledge of their problems, both internal and external.

However, there is a marginal fringe composed of men living complacently in the smug shadow of past experience, lacking the intellectual drive to create new experience, and the desire to listen to others. Past-President C. F. Kettering once said, "When a man gets to a point where no one can tell him anything, the only thing to do with him is to bury him." Such men will do well to remedy their swivel-chair inertia and their home-office complex by getting out in the field more frequently, and right down to the lowest echelons of automotive operation and maintenance. This should develop a constant urge for betterment.

Fleet Manager's Superior Officers

Of course, there are examples of higher company executives who are lacking in automotive knowledge and therefore in understanding of the proper scope and responsibilities of the job. In such instances the fleet operator must realize that ample thought-provoking data easily may be developed with which to impress the boss with his automotive facts-of-life.

As for the higher executive who has not been close to his motor-vehicle problems, I suggest that he answer the following self-questionnaire:

1. What is my total motor-vehicle investment?

2. What is my total annual expense for motor-vehicle operation, maintenance, fixed charges, and the like?

3. How many units does my company own?

4. How many units does my company rent? Why?

5. Have I an adequate cost system for control of automotive activities?

6. How does my expense per ton-mile, or other

basis of measurement, compare with that of other companies in my industry. How are my net profits affected thereby?

7. Are the activities and responsibilities of my automotive personnel properly organized, coordinated, and charted?

8. Is my fleet manager properly qualified, technically, and as a business executive?

9. Have I given him sufficient authority to enable him to exercise adequate control?

10. Do I consult with him frequently enough?

The company executive, if he will but spend intensively the right amount of time on this complex problem, cannot help but gain the complete understanding which is necessary if he is to reap the benefits which can be achieved only as a result of his final authority.

Technical Qualifications of Fleet Operators

In respect to technical qualifications and duties, the fleet operator, as L. V. Newton said years ago, should:

"1. Have a good education, preferably engineering training.

"2. Preferably have a good working knowledge of motor-vehicle repair work gained through actual experience in automotive repair shop."

The value to his organization (and to the industry) of the man charged with fleet management is, of course, greatly enhanced by his ability, thorough technical training, and/or experience to understand intimately the construction, proper usage, and maintenance of the complicated, expensive equipment which he commands. As F. K. Glynn of AT&T once said, he must be able to "determine the most efficient type, make, and size of chassis, body, accessories, and auxiliary equipment for the job."

He must be thoroughly proficient and precise in coming to conclusions as to minimum field operating requirements. This includes necessary horsepower to achieve maximum overall economy (including driver cost), dimensional fundamentals such as wheel base, cab-to-axle distance, king-pin offset, suitable axle ratios, proper transmission, design and selection of bodies, and many other items.

He must be capable of making staff studies of a multitude of problems, including maintenance research, unusual designs to fulfill particular cases, diesel versus gasoline power as applied to his operation, cause of failure of parts or components, and so on.

He must be keenly and constantly aware of the fact that the major target of his training and experience is that of applied science – the production of something useful; in his case, economy of automotive capital investment as well as operating expense.

He need not be qualified to specify the proper physical, chemical, and design characteristics of a crankshaft, but he should understand thoroughly the fundamentals of overall vehicle design, and detailed specification requirements in terms of predetermination of the hauling job the equipment is intended to satisfy.

He need not be a qualified fuels and lubricants man, but he should be basically grounded in this complex subject. Indeed, he will do well to have much more than a nodding acquaintance with such other SAE professional activities as Truck and Bus, and Diesel Engine, plus others which may be within the scope of his operation.

I do not imply that he must be a "jack of all trades," but he must energetically study technical matters relevant to his responsibilities. Certainly the amount and quality of his know-how will be inverse to his static inertia.

He should shun "study for study's sake" and get into the arena of real contribution, not only to his company, but to the automotive and to other industries related to his work. The transportation and maintenance bibliography contains many excellent examples of such contributions, such as of recent date, cooperation with other members in the compilation of excellent recommended practices concerned with vehicle ability.

More such practical, usable contributions are needed in order to insure retention of satisfactory professional standing by the Transportation and Maintenance Activity. I am sure plenty will be forthcoming.

Owner-Manufacturer Cooperation

I believe that I would be remiss if I did not at this point emphasize a fundamental matter of long standing importance which I think demands less of perfunctory concurrence and more of shirt-sleeved attention. The fleet operator is eager for improvement of designs and materials. So is the vehicle manufacturer. In regard to collection of factual and useful frequency-of-failure data, in consultation as to needs with respect to maintenance cost reduction through better accessibility, and in many other matters, the fleet owner could be of much more use to the manufacturer than he is. May I be indulged if I quote from remarks I made years ago before the Ninth National Transportation Meeting:

"One of the most important relationships which can exist between the transportation executive and the manufacturer has to do with trends in motor-vehicle design. The importance of this is self-evident. The manufacturer who more accurately can forecast design trends so that he may be in a more favorable position to know just what to build is fortunate indeed.

"I firmly believe that considerable of the guesswork which must be undertaken under present conditions can be eliminated. Buyers and sellers should, by all means, draw closer together in matters of transportation and design research. For one

thing, it is incumbent upon the manufacturer to insist that his engineers spend more personal time with buyers. The inadequacy of such contacts in the past cannot properly be explained. It is true that the engineer who calls upon a great number of buyers in many different vocations will receive many conflicting and contradictory statements, many of them of no significance. He will, nevertheless, observe trends and receive some splendid suggestions from operators. Too often does the manufacturer's designing engineer cloister himself in his own office and depend upon his examination of competitors' specifications and conversations with his own personnel, instead of personally exposing himself to the customer – the man who ultimately pays his salary."

Isn't the above still true, in some measure?

More personal contact between transportation executives and the design and production engineers, who bring into being the equipment they buy and operate, should have a wholesome effect. It is gratifying to learn that one heavy-duty manufacturer initiated effective steps in that direction only last April. The sales department is usually ineffectual in its endeavor to be a catalytic agent between the operator and the design engineer.

Conclusion

In my attempt to provide in preceding remarks a reasonable job evaluation of fleet management, for reasons of time and space I had to refrain from exploration of many interesting corollaries and deductions. I have tried largely to retain, as twin threads of the story, the management and the technical aspects of the job. I hope the paper may on occasion prove useful in respect to recommended practices, to fleet managers, and to the senior managements of both users and producers.

As a salutary closing, I can think of nothing more beneficial than the following words of Past-President H. L. Horning:

"The cost of turnover among technical men is negligible compared with the possibilities that lie dormant in most of them. The major causes of turnover are the failure of management to exercise sufficient care in choosing men for employment and to surround them with conditions which stimulate them to produce their best results. The secondary cause lies in the unwillingness of the men to sacrifice time and effort and make the constant drive toward perfecting themselves that is the price of success. Management receives its greatest encouragement to help men who are willing to help themselves. The man who gives untiring effort outside of his working hours to perfecting himself has the best chance of achievement, and if the efforts are well and persistently directed, they will always produce results."

Surely, these, in eloquent simplicity, are fundamental truths, both for top management and for the fleet executive.

DISCUSSION

Trouble Report Form Suggested Driver Aid

– F. K. Glynn

American Telephone & Telegraph Co.

I WOULD like to add a few more words on one angle of the problem – mutual cooperation between the fleet operator and vehicle manufacturers' engineering, production, and service.

Such cooperation is not successful as a haphazard procedure. Therefore, let me start at the beginning to build up a routine plan of orderly procedure in a fleet.

The first man to sense a fault in a properly selected motor vehicle in regular service is the driver. Give him an easy, but sure way, to report this trouble and keep everlastingly after him (in a nice way, you understand!) to report all troubles at the end of his working day. I could throw in the idea here that the driver's trouble report form have a detachable portion which can be filled in by the motor-vehicle people and placed in the cab to advise the driver what corrections were or will be made, and to thank him for his cooperation.

The next man exposed to the fault or trouble is the garage foreman or the mechanic who cares for the driver's report. He can correct the same trouble in the same model time after time with complete futility, or he can become an important link in the chain to reach upwards in the fleet organization and outwards to the manufacturer. Isolated or nonrepeating troubles are inconsequential in the program. It is the repeaters and important prematures that should be reported up the line after study by the foreman with the mechanics.

In a scattered fleet, there may be district and division levels. At the next collective level above the garage reporting the trouble, inquiry should be made as to prevalence of the trouble at other garages. Any sidelights, remedies and extent of the trouble should be added to the report and the report forwarded to area home office. At any level where personnel are competent, engineering and designwise, an analysis should be added to the report.

When the report built up as suggested reaches the Superintendent of Motor Equipment, it is indisputable evidence of real value to the manufacturer. Fleets can usually realize and prove troubles far ahead of dealer, branch, and factory service organizations.

The superintendent has a problem of immediate field compromise or cure of the trouble. At this point the factory zone, district, or branch service manager should be of assistance. At least he will relay the trouble to the factory.

If the superintendent devises an effective field prevention or a cure, it should be promptly transmitted through organization to the mechanics. This is just getting out of current trouble at a cost. Foremen and mechanics will read the "practice" written at the home office, I hope, regardless of the stiff and stilted English in which it may be written. They will enjoy reading it, however, if you'll write the same facts in language as you would talk to them face to face.

At this point in my discussion we have established and may have cured the trouble. What about the future? The factory probably knows about the trouble from the service representative you called in but few (very few) manufacturers will pick up and carry on unless the fleet superintendent shows an interest in correction of engineering design or production procedure at manufacturing level.

This interest is best manifested by the Fleet Superintendent asking for a conference with factory personnel. This is mutual cooperation to clear current troubles and produce better motor vehicles in the future.

Temperature Gradients and Heat Stresses in Brake Drums

G. A. G. Fazekas, Polytechnic Institute of Brooklyn

This paper was presented at the SAE Summer Meeting, Atlantic City, N. J., June 6, 1952.

UNTIL recent years, friction fade and/or the concomitant excessive lining wear had been the main factors limiting the size and weight of brakes. Lately, however, the position underwent a radical change; World War II provided an imperative impetus on account of aircraft brake requirements. As a result, friction materials were so much improved that they can operate now up to a *nominal* 1500 F, at 500 to 1200 psi loading,[1] they may sustain 10 hp/in.[2] energy absorption[2] and even further improvements seem to be within reach.

This change passed the onus of being the weakest link in the chain onto the brake drums. As is well known, when their temperature exceeds a certain amount, they tend to develop heat checks, causing in turn brake shudder that can assume alarming proportions. This is further accompanied or followed by crazing and/or heavy scoring, leading eventually to outright fracture. A significant aspect of such thermal failures is that almost invariably the drum seems to be eaten up piecemeal.

Concerning the mechanism of thermal failure,

[1] See Proceedings of the Institution of Mechanical Engineers (England), Vol. 156, 1947, p. 297: Discussion by E. W. Sisman of paper, "Aircraft Wheel Brakes," by D. A. L. Robson.
[2] Private communication from R. C. Parker.

THE main purpose of this paper is to show that the phenomena classed under the generic thermal failure are not due to excessive temperature alone, but to a combination of high stresses and high temperature; other factors of importance are brake design, geometry, physical properties of brake lining, and those of the drum.

Concerning stresses, it will be shown that temperature gradients alone can cause them to reach well into the plastic state, even in a comparatively cool drum. Such heat stresses are known to give rise to crazing of the track, but in addition it is explained here that they are also responsible for heavy scoring, heat checks, or outright fracture.

A detailed analysis indicates why and how certain purely mechanical design aspects lower stresses and heat concentrations, thereby mitigating or even preventing the onset of thermal failure. Included is also an attempt to provide a rational basis for appraising the suitability of drum materials and of plating, in terms of simple physical properties.

Lastly, design charts and procedures are explained and illustrated in a few examples from the author's practice.

The Author

G. A. G. FAZEKAS is assistant professor of mechanical engineering at the Polytechnic Institute of Brooklyn. He received his mechanical engineering degree from the Federal Institute of Technology in Zurich, Switzerland, and worked for many years in England. He has been associated with the automotive industry since 1942, principally working on the design and development of brakes, clutches, and high-pressure hydraulics.

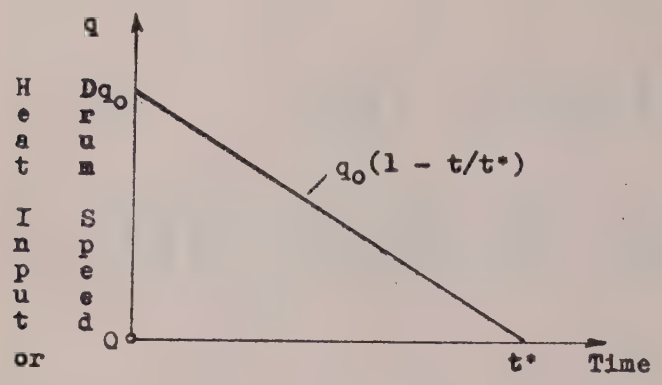


Fig. 1 – Assumed drum speed and heat input, versus time

so far one aspect only, crazing, seems to have been clarified. Crosby and Timmons[3] attributed it to excessive skin stresses, generated by temperature gradients that arise on severe braking. For during actual braking the compressive heat stresses tend to pucker the skin, the high spots thus generated are then rubbed harder, receive more heat, which makes them grow, so that a vicious circle develops. The skin being overstrained will then set, so that on cooling it will be in tension which may be easily great enough to tear it: hence the origin of crazing. This does not explain, however, the origin of heat checks, or the cause of fracture when no other thermal damage is visible, or the cause of heavy scoring without any tangible clue.

Hitherto the connection between scoring and other than abrasive stresses appeared unknown, while the more obvious thermal failures were approached from two sides only; (1) vehicle manufacturers attempted to provide adequate cooling, but with little success, owing to prevalent fashions in wheel design which passed the buck to (2) metallurgists who achieved a very notable but unfortunately limited improvement.[3, 4] From the short-term point of view the writer was thus forced to seek further improvements by experimenting with the mechanical design alone. It soon transpired that three further approaches were open:

(a) Certain aspects of brake design and geometry were found to inhibit the onset of heat checks.

(b) Some clearly definable physical properties of friction materials sufficed to overcome borderline cases.

(c) Very considerable improvement could be achieved by, what for want of a better word may be termed as strainlining the brake drum, the effect of which is cumulative to (a) and (b).

The present paper deals with the brake drum side alone, the effects of (a) and (b) will be touched upon only insofar as seems necessary to provide a physical picture of the phenomena known under the generic of thermal failure. It is hoped that qualitatively at least, and so far as the drum is concerned, the picture presented is nevertheless complete, the main findings being:

1. Quite apart from and in addition to the weakening effect of high temperature, excessive skin stresses of other than abrasive origin, promote heavy scoring as well. Because once a material is heavily strained, its resistance against additional stress or strain – that is, against abrasion – tends to evanesce. The extent of abrasive damage depends, therefore, also upon factors extraneous to the surface.

2. Skin stresses of purely surface origin are superimposed upon those emanating from the bulk, the major part of which are usually due to temperature gradients, and not to mechanical agencies.

3. Bulk stresses of whatever origin are, of course, very much influenced by the shape and dimensions of the drum.

4. They are also influenced by the physical properties of the materials, including those of the surface (plating). In this connection it was possible to find both an analytical proof for, and an extension of, Oberle's Modell concept.[5]

5. Some of the bulk stresses were found to be of a nature capable of inducing a circumferential elastic instability (incipient buckling) which would explain the fundamental origin of (symmetrically spaced) heat checks.

Whereas chemical and/or allotropic changes are brought about by temperature alone, various mechanical effects are induced by stresses, the magnitude of which depends primarily upon temperature gradients. Therefore, the absolute level of temperature is of concern here only insofar as it affects the elastic, and plastic, properties of the material. And within a large temperature range these effects will be shown to be smaller than one would expect.

The subject matter itself refers primarily to cylindrical drums, and is divided into four parts: the first one treats temperature fields and their level, the second one deals with the heat stresses induced, the third one with some simple stresses of mechanical origin, and the fourth with the more important factors governing practical design considerations.

Application of developments to both aircraft and motor car brakes resulted in somewhat different design rules, as service conditions and requirements are not the same for the two. Bulk temperature of aircraft drums will be shown to depend both on the quantity of heat and on the rate of heat generation. Temperature gradients are roughly propor-

[3] See *Foundry*, Vol. 68, July, 1940, pp. 32-34, 83-84: "Metallurgical Aspects of Brake Drums for Heavy-Duty Service," by V. A. Crosby and G. A. Timmons.

[4] See *Metals and Alloys*, Vol. 15, February, 1942, pp. 256-262; March, 1942, pp. 405-411; April, 1942, pp. 595-598: "Brake Drums," by Oliver Smalley.

[5] See *Journal of Metals*, Vol. 3, June, 1951, pp. 438-439G: "Properties Influencing Wear of Metals," by T. L. Oberle.

tional to thickness, they amount to several hundred deg C, and as the thickness of the drum is increased, its power to absorb a thermal shock will be shown to deteriorate rather rapidly, as the saturation limit is approached.

With motor vehicle drums the picture is somewhat different, owing to unavoidable residual temperature. For if the cooling surface remains the same, a thick section cools more slowly and collects more heat in repeated use. But with increasing thickness the temperature rise per application decreases, so that there is an optimum thickness indicated by a simple formula.

In both cases however, the choice of thickness is further influenced by heat stresses which were found to be proportional to $\sqrt{\text{thickness}}$. Other factors remaining the same, thin drums therefore tolerate higher temperatures than thick ones. With conventional motor car drums bending stresses (and deflection) warrant a heavy section, so that again an optimum condition arises.

With any type of drum, a considerable part of the stress is, of course, confined to the surface skin of the track, and is proportional to flash temperature. To minimize it, perhaps the most important single rule is to spread the contact evenly, which means that heat distortions must be put under control by strainlining. Even so, on moderately loaded drums heat stresses alone may well extend into the plastic state. For this reason a purely analytical treatment must be regarded as an approximation only, which in the absence of anything better, did serve as a rather reliable guide for experimental work. The latter took place over several years, but in a rather *ad hoc* manner, as and when trouble arose, so that only the gist of the more salient results can be given here.

Part I – Drum Temperature

Surface temperature (T_s) of the drum is made up of four components superimposed one on another:

(a) Temperature of the drum at the commencement of braking (T')

(b) Average temperature rise (T_a) due to energy absorbed

(c) Temperature rise ($T_m - T_a$) due to gradients across the drum

(d) Flash temperature (T_c) due to imperfect contact.

Of these, T' is not very different from the ambient temperature T_o, with aircraft or very lightly loaded motor vehicle brakes. In heavily loaded and/or poorly cooled motor vehicle brakes, however, the phenomena are more involved and will be treated first; (b) and (c) will be jointly considered next, followed by (d).

I. 1. Effect of Cooling on Mean Drum Temperature

Effect of Cooling on Mean Drum Temperature – The way motor vehicle brakes are used can be statistically visualized as consisting of two types of applications: (1) normal cruising on English roads corresponds to about ⅝ maximum vehicle speed with a $40\%g$ stop about every 2 miles; (2) interspersed are occasional bursts of heavy braking, which may be taken for the present purpose as consisting of about 6 consecutive stops, at $65\%g$ deceleration, at about ½-mile intervals, from ⅞ maximum speed. Current road-testing procedure is conducted about on these lines.[6]

Let us consider now a solid of weight (W), of specific heat (c), having an exposed surface (A) which is to cool slowly from an initial temperature (T''). If the coefficient of heat transfer (h) is constant, it follows from the Newtonian law of cooling that the temperature (T) at a time (t) is given by:

$$T - T_o = (T'' - T_o)\, e^{-bt} \qquad (1)$$

where:

$$b = \overline{A}h/Wc \qquad (2)$$

As the brakes are applied at regular intervals (t_o), receiving every time the same heat input (Q), the corresponding mean temperature rise per stop is:

$$T'' - T' = \overline{\Delta T} = Q/Wc = Qb/h\overline{A} \qquad (3)$$

(For reasons shown below, it is usually justifiable to neglect cooling during the actual process of braking.) The mean drum temperature is thus:

First stop

before $\quad T_1' = T_o$

after $\quad T_1'' = T_o + \overline{\Delta T}$

Second stop

before $\quad T_2' = T_o + \overline{\Delta T}\, e^{-bt_o}$

after $\quad T_2'' = T_o + \overline{\Delta T}\,(1 + e^{-bt_o})$

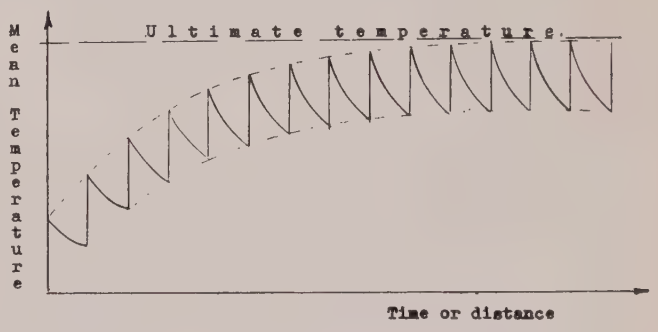


Fig. 2 – Mean temperature level on cruising

[6] See *Automobile Engineer*, Vol. 41, May, 1951, pp. 185-191: "Brake Torque – Some Notes on the Technique of Evaluation and Measurement," by G. A. G. Fazekas.

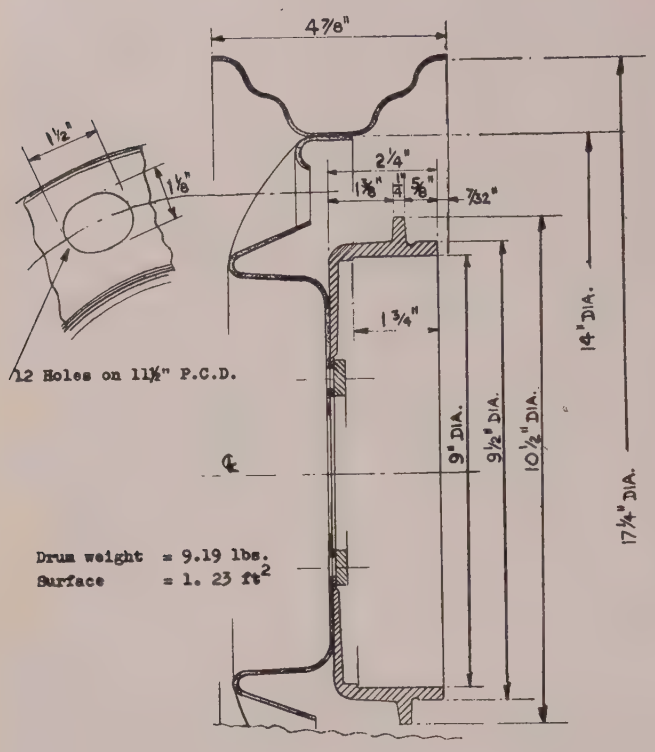


Fig. 3 – Details of wheel and drum used for cooling tests in Figs. 4 and 5

Third stop

before $T_3' = T_o + \overline{\Delta T}\,(e^{-bt_o} + e^{-2bt_o})$

after $T_3'' = T_o + \overline{\Delta T}\,(1 + e^{-bt_o} + e^{-2bt_o})$

It follows thus by induction that:

nth stop

$$\left.\begin{aligned} &\text{before} && T_n' = T_n'' - \overline{\Delta T} \\ &\text{after} && T_n'' = T_o + \overline{\Delta T}\,\frac{1 - e^{-nbt_o}}{1 - e^{-bt_o}} \end{aligned}\right\} \quad (4)$$

so that for the ultimate steady state:

$$\left.\begin{aligned} T_\infty' &= T_o + \frac{\overline{\Delta T}}{e^{bt_o} - 1} = T_o + \frac{Q}{hAt_o}\left(\frac{bt_o}{e^{+bt_o} - 1}\right) \\ T_\infty'' &= T_o + \frac{\overline{\Delta T}}{1 - e^{-bt_o}} = T_o + \frac{Q}{hAt_o}\left(bt_o + \frac{bt_o}{e^{bt_o} - 1}\right) \end{aligned}\right\} \quad (5)$$

Table 1 – Drum Cooling Tests

Vehicle Speed, mph	Front $b \times 10^3$ sec^{-1}	Front $h \times 10^3$ Btu / ft² × deg F × sec	Rear $b \times 10^3$ sec^{-1}	Rear $h \times 10^3$ Btu / ft² × deg F × sec
0	0.65	0.63	0.83	0.81
30	2.20	2.15	1.33	1.29
50	3.63	3.52	1.33	1.29

When $bt_o < 6/5$, as is the case in practice, a good approximation may be obtained by:[7]

$$0 < \frac{bt_o}{e^{bt_o} - 1} - 1 + \frac{bt_o}{2} < \frac{(bt_o)^2}{12} \quad (6)$$

Equation 5 can be then expanded and rewritten as:

$$\left.\begin{aligned} T_\infty' - T_o &\approx Q\left(\frac{1}{hAt_o} - \frac{1}{2Wc}\right) \\ T_\infty'' - T_o &\approx Q\left(\frac{1}{hAt_o} + \frac{1}{2Wc}\right) \end{aligned}\right\} \quad (7)$$

showing that as drum weight is increased, it has opposite effects on T_∞' and T_∞'', respectively.

In normal cruising the ultimate mean temperature is therefore reached in a manner sketched in Fig. 2, and is given by equations 5 or 7. The first part of the curve – but according to a different scale – represents also a burst of heavy braking. The corresponding temperatures can be obtained from equation 4 by writing $n = 6$. When cooling is good, that is, for $bt_o > \frac{1}{2}$, $e^{-6bt_o} < 0.05$, and the simpler formulas 5 or 7 can be used here as well. If, however, $t_o b < \frac{1}{4}$, equation 4 may be expanded as:

$$\frac{1 - e^{-6bt_o}}{1 - e^{-bt_o}} = \frac{6bt_o - 18(bt_o)^2 + \cdots}{bt_o - \frac{1}{2}(bt_o)^2 + \cdots} \approx 6 - 15bt_o \quad (8)$$

resulting in:

$$T_6' - T_o = \frac{Q}{Wc}(5 - 15bt_o)$$

$$T_6'' - T_o = \frac{Q}{Wc}(6 - 15bt_o) \quad (9)$$

When using the preceding formulas the question may arise – what effective drum weight and drum surface should be substituted? As on motor car drums the thickness of the flange and of the cylindrical part are of the same order, roughly, $\overline{A} \propto W$, so that the total drum weight and surface may be substituted into equation 2. On the other hand, since the heat takes some time to permeate down the flange, the equations for T''' do not strictly apply at the end of the stop, but some short time after.

The aerodynamics being rather involved around wheel hubs, the simplest way to find h is to carry out some cooling tests with the vehicle. With the wheel and drum shown in Fig. 3, fitted to a 12-hp (RAC rating) popular make of car, such cooling curves are plotted to a semilogarithmic scale in Figs. 4 and 5. As the results were reasonably linear, they have been condensed into Table 1.

Assuming now that this 3080-lb car would cruise at 45 mph, with a 60/40 front-rear braking ratio, the heat input is:

$$Q_{\text{front}} = 83 \text{ Btu/brake} \qquad Q_{\text{rear}} = 55.5 \text{ Btu/brake}$$

If the car is stopped every 160 sec, maintaining a steady 45 mph in between, with reference to Table 1 and equation 7:

for the front $T_\infty - T_o = 83\left(\frac{1}{0.661} \pm \frac{1}{2.39}\right) = 160 \text{ F}$ or 91 F

[7] See p. 297 of "The Introduction to the Theory of Infinite Series," by T. J. I'a. Bromwich. Second edition (revised by T. M. Macrobert), MacMillan & Co., Ltd., London, 1949.

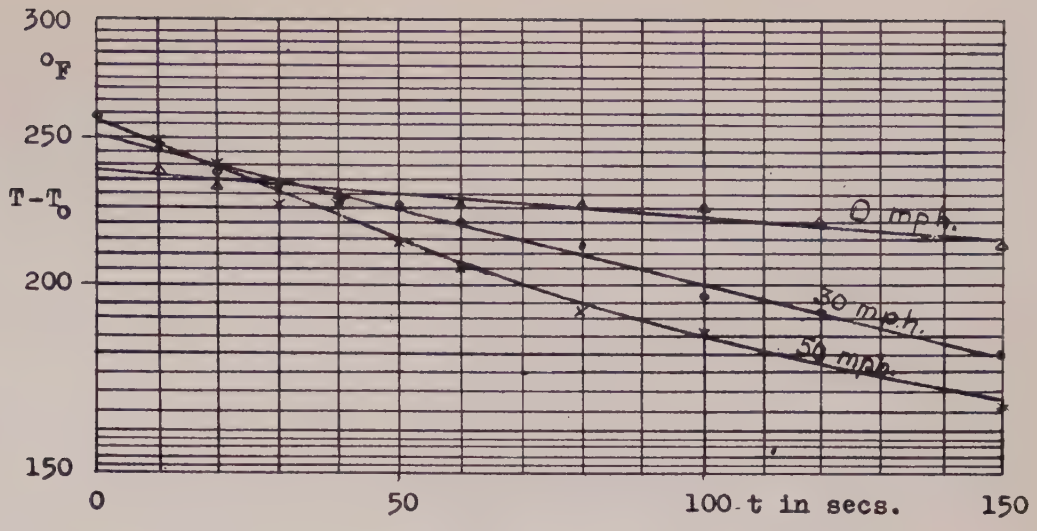


Fig. 4 – Cooling tests results on front drums

or the rear $T_\infty - T_o = 55\frac{1}{2}\left(\frac{1}{0.253} \pm \frac{1}{2.39}\right) = 243$ F $= 197$ F

The figures in parentheses suggest that increasing the heat capacity of the drum would make thus little difference to this brake.

On some modern motor vehicles with full disc wheels h is sometimes less than half of the above value, so that the heat capacity of the drum has a much greater influence. Also with such hot running drums h is in general not constant, mainly because of appreciable radiation effects at high temperatures. An average value of it is, however, good enough to check with equations 4, 5, or 7 whether the drum weight is of right proportion. Such calculations are also necessary to bring dynamometer tests in harmony with field conditions.

To show that during the actual process of braking the heat loss is in practice not significant, let us first consider the previous example. A 40%g stop from 45 mph requires ideally 5.1 sec. The cooling curves in Figs. 4 and 5 show that with a temperature difference of 250 F, the corresponding temperature loss is under 5 deg F, amounting to some 4% on, by modern standards, exceptionally well-cooled brakes – and on the tacit assumption that as the heat is generated it would instantaneously diffuse throughout the drum.

This picture is equally true concerning very heavily rated aircraft drums with a mean temperature rise of approximately 550 C per stop, and an average heat input in the order of 2 Btu/in.^{2} x sec. For if the drum is exposed, for example, to a 120 mph wind, of 20 C temperature, the convective heat loss is in the order of 0.039 Btu/in.^{2} x sec,[8] and the loss due to radiation could not exceed 0.0163 Btu/in.^{2} x sec by the Stefan-Boltzman law. These figures again presuppose instantaneous diffusion of heat, and that the rubbing surface is completely enclosed by a black body maintained at 0 deg K! For the radiation to cause a 10% loss the *whole* surface would have to exceed 1100 C (equal to the melting point of cast iron). Nor can conductive losses be serious for reasons explained in the following section.

I. 2. Temperature Gradients

Temperature Gradients – In a first approximation a brake drum can be envisaged as an infinite slab of uniform thickness (r), with one face heat insulated, and the other receiving a heat flux (q) as in Fig. 1, uniformly distributed over the entire surface. This simplified model does not include any

[8] See p. 221 of "Heat Transmission," by W. H. McAdams. McGraw-Hill Book Co., New York, second edition (revised), 1942.

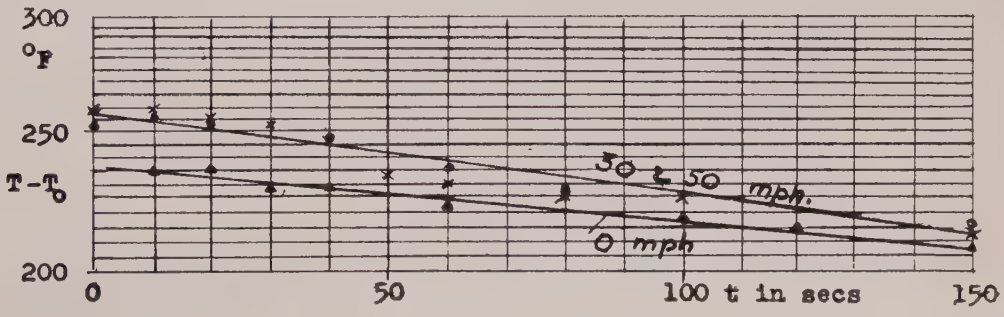


Fig. 5 – Cooling tests results on rear drums

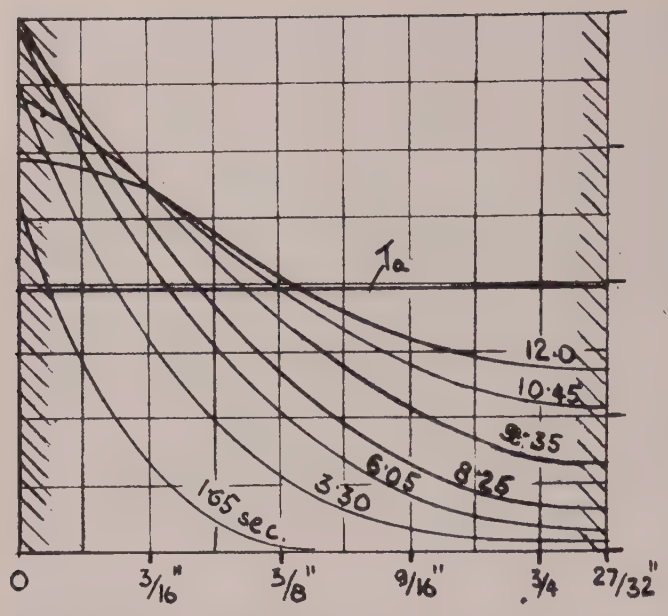


Fig. 6 – Isochronic temperature gradients in 27/32-in. thick cast-iron (aircraft) drum for heat input $H = 10$ Btu/in.^{2} during $t^* = 12$ sec

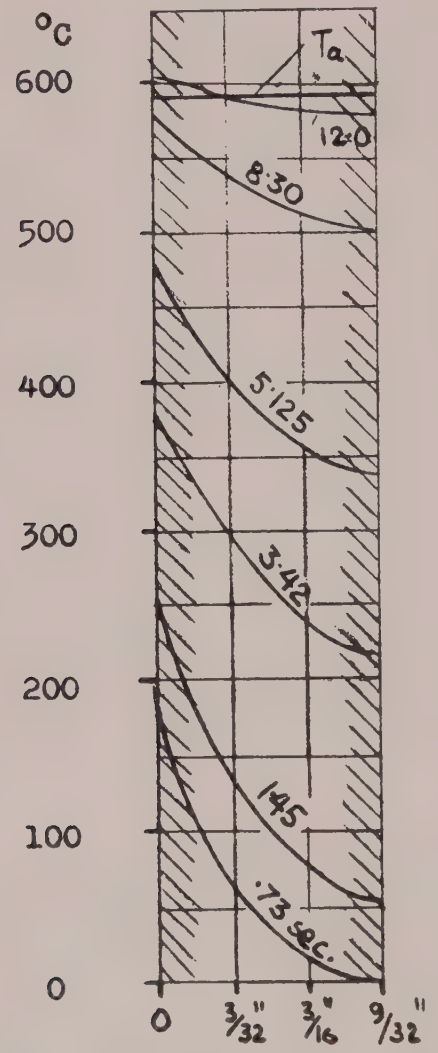


Fig. 7 – Isochronic temperature gradients in 9/32-in. thick cast-iron (aircraft) drum for same conditions as in Fig. 6, to show effect of thickness

flange effects, but it will be shown below that even a heavy flange has a comparatively small direct effect on the temperature field. To simplify matters it is further assumed that the physical properties of the material are not affected by temperature. With heavily loaded drums this is certainly not true, but mean values yield a reasonable approximation.

On the above premises the 1-dimensional form of Fourier's equation:

$$\frac{\partial T}{\partial t} = a \frac{\partial^2 T}{\partial x^2} \tag{10}$$

has to be solved for the boundary conditions:

$$\left(\frac{\partial T}{\partial x} k\right)_{x=0} = q_o (1 - t/t^*) \quad \text{and} \quad \left(\frac{\partial T}{\partial x}\right)_{x=r} = 0 \tag{11}$$

Figs. 6 to 8 show some typical solutions, obtained graphically by Schmidt's Method,[9] for cast iron slabs having the following properties:

$$\text{Density } (w) = 7.30 \text{ g/cc}$$

$$\text{Specific heat } (c) = \frac{0.130 \text{ cal}}{\text{g} \times \text{deg C}}$$

$$\text{Conductivity } (k) = \frac{0.110 \text{ cal}}{\text{cm} \times \text{sec} \times \text{deg C}}$$

$$\text{Diffusivity } (a) = \frac{0.116 \text{ cm}^2}{\text{sec}}$$

Figs. 6 and 7 are taken from aircraft practice, and show the effect of thickness for the same heat input, for a 33%g stop from 90 mph landing speed. Figs. 8 and 9 represent extremes of motor car practice, and show the effect of thickness for the same average temperature rise, for a 75%g stop from 70 mph. Fig. 10 is a crossplot of Fig. 8 to show the surface temperatures (T_s) and their difference (ΔT) across the slab.

Such isolated solutions can be now correlated by dimensional analysis for which the thermal dimensions[10] are:

H = Heat (cal)
L = Length (cm)
t = Time (sec)
T = Temperature (C)

The temperature (T) of a layer (x) distant from the surface is then governed by the factors shown in Table 2.

By the II theorem it follows thus that:

$$\frac{TCr}{H} = 1/\Theta' = f\left(\frac{Cr^2}{kt^*}, \frac{x}{r}, \frac{t}{t^*}\right) \tag{12}$$

where:

$$kt^*/Cr^2 = 1/F = \text{Fourier's number}$$

In equation 12:

$$T_o = H/Cr \tag{13}$$

[9] See p. 197 of "Foeppl Festschrift," by Ernst Schmidt. Springer-Verlag, Berlin, 1924.
[10] See "Dimensional Analysis," by P. W. Bridgman. Yale University Press, New Haven, Conn. (revised edition), 1943.

Table 2 – Main Factors Affecting Temperature

Factors	Symbol	Dimension
Total heat input	H	H/L^2
Volumetric heat capacity	$C = c\rho$	H/TL^3
Heat conductivity	k	H/LTt
Thickness of slab	r	L
Distance from surface	x	L
Stopping time	t^*	t
Time	t	t
Temperature (x, t)	T	T

represents the average temperature rise of the cylindrical part only; $\Delta T \neq T_a$ because the heat does not penetrate beyond the fringe of the flange during (t^*).

Referring further to equation 12, the maximum temperature rise (T_m) and maximum temperature difference across the slab (ΔT_m) are functions of F only. Similarly, their times of incidence, t_m and t_d, are also functions of F only, which served as a basis for plotting these four quantities in Fig. 11. The broken line there shows:

$$1/\Theta = \frac{T_m}{T_a} = 0.855 + 0.353\ F \tag{14}$$

which is a very good approximation for $0.50 < F < 3.50$, covering the practically important range. Another good approximation is:

$$\Delta T_m/T_a = \Delta\tau_m = 0.80\ F^{0.75} \tag{15}$$

As may be seen from its plot, shown in a broken line in Fig. 11, it is valid for:

$$0 < F < 4$$

The foregoing indicates that as r is increased its influence on T_m tends to diminish. When $r = \infty$ the exact solution[11] shows that T_m tends to a saturation limit:

$$\left(T_m\right)_{r=\infty} = T_m^* = H\sqrt{\frac{N}{kcwt^*}} \tag{16}$$

where:

$$N = 32/9\pi \approx 1.130 \tag{17}$$

This affords a convenient measure to compare the intrinsic ability of various materials to absorb a thermal shock. Table 3 shows a few figures for $H = 3$ Btu/in.^{2} and $t^* = 4.25$ sec.

For the present purpose Na has been treated as a solid and its latent heat of fusion ($L_o = 27.5$ cal/g) was neglected. Roughly, the correction is equal to L_o/c=91.5 C which brings it in line with ferrous materials.

Equation 16 supplies also a measure of the useful

[11] See pp. 109, 113, "Modern Operational Mathematics in Engineering," by R. V. Churchill. McGraw-Hill Book Co., New York and London.

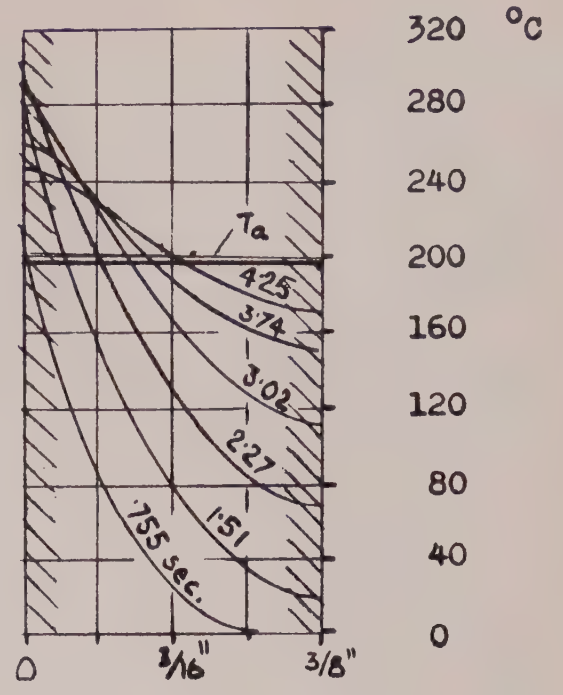


Fig. 8 – Isochronic temperature gradients in 3/8-in. thick cast-iron (passenger car) drum, for heat input $H = 4.50$ Btu/in.^{2} during $t^* = 4.25$ sec

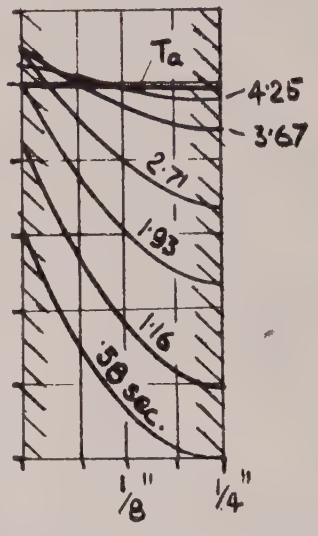


Fig. 9 – Isochronic temperature gradients in 1/4-in. thick cast-iron (passenger car) drum, for same mean temperature as in Fig. 8, but corresponding to $H = 3$ Btu/in.^{2} during $t^* = 4.25$ sec

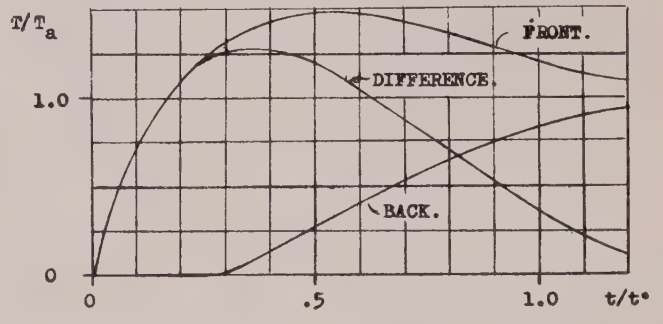


Fig. 10 – Crossplot of surface temperatures from Fig. 8

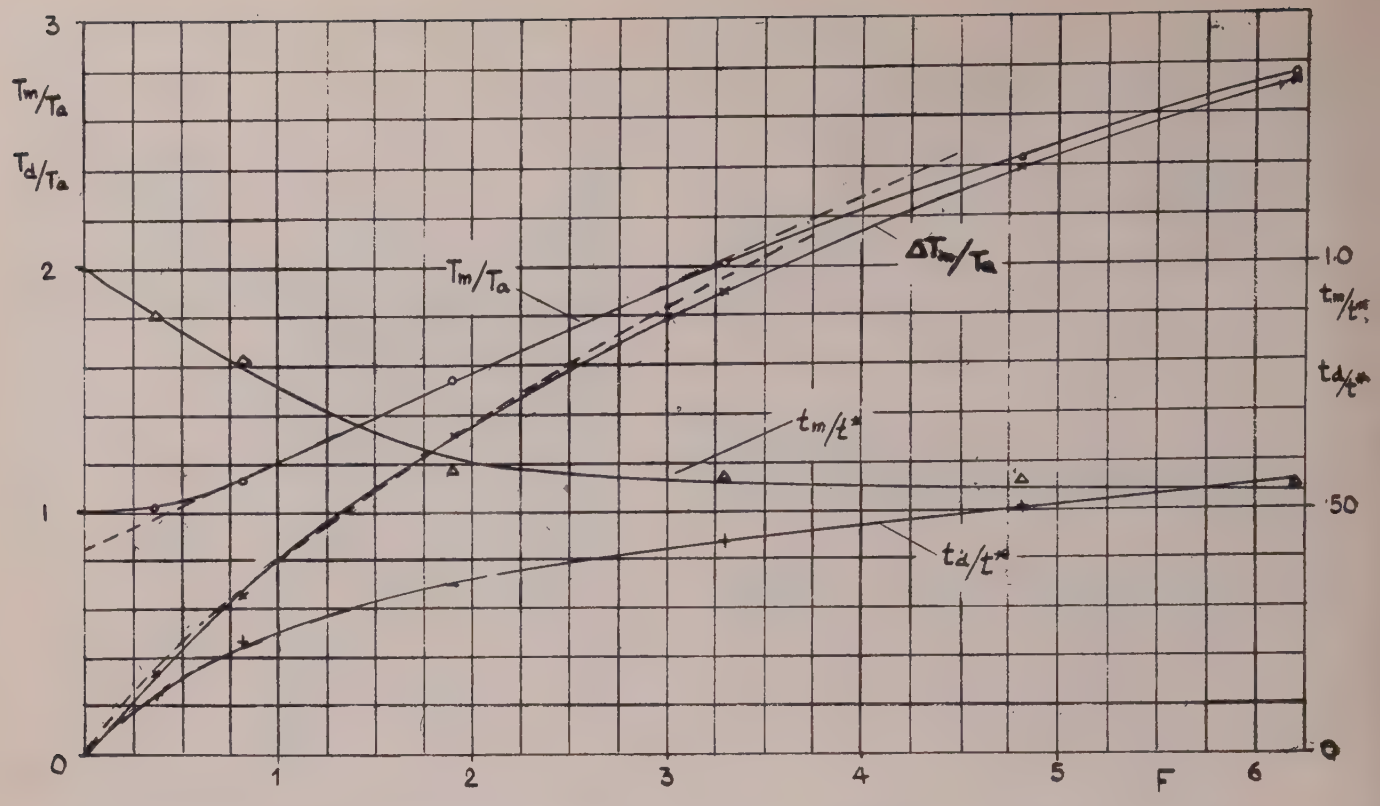


Fig. 11 – Design chart to obtain maximum temperature rise (T_m), maximum temperature difference (ΔT_m), and their times of incidence (t_m) and (t_d), as a function of Fourier's number

amount of r. For with reference to equation 14 the degree of thermal saturation may be defined as:

$$\Phi = T^*_m/T_m = \theta\sqrt{NF} \tag{18}$$

the plot of which in Fig. 12 indicates that beyond a value corresponding to $F = 1.75$, r ceases to have a significant effect on T_m. With $a = k/cw$ as the diffusivity, this value of r is thus:

$$r^* = 1.75\sqrt{at^*} \tag{19}$$

It follows too that the effective heat capacity of the section is given by:

$$C_e = cwr\theta \tag{20}$$

θ is also plotted in Fig. 12, showing that for $r = r^*$, $\theta = 68\%$.

The previous treatment referred to slabs of uniform thickness. When r varies, the 2-dimensional form of equation 10 can be also solved graphically,[12] and leads to two main conclusions:

(1) During the critical phase, $t \leqq \frac{5}{8}t^*$, only that part of the section need be considered for which $F \leqq 1.75$. Heavy flanges, or ribs, have therefore a surprisingly small direct effect on T_m.

The previous formulas remain then valid a little distance away from a heavy flange or the like.

(2) If the thickness varies in a manner shown in Fig. 13, it can be assumed to be made up of infinitesimally narrow strips, each insulated from its neighbor, and for each T_m can be calculated separately. Since in such strips T_m does not occur simultaneously, a distance-integrated average yields a reasonable approximation. Fig. 14 shows an example from aircraft practice – the drum being of cast iron encased in a steel shell. With such drums the heavy sections bear stronger against the shoes, which further assists equalization of temperature in the bulk.

Table 3 – Saturation Temperature of Some Metals

Material	Cast Iron	Steel	Cu	Al	Cr	Na
kcw, $\left(\frac{\text{cal}}{\text{cm}^2 \times \text{deg C}}\right)^2 \frac{1}{\text{sec}}$	0.105	0.108	0.780	0.340	0.091	0.058
T^*_m, deg C	187	185	88.5	104	201	252

I. 3. Temperature Flashes

Temperature Flashes – The foregoing treatment of temperature gradients has been based on the assumption that the heat flux q is uniform over the

[12] See *Zeitschrift des Vereines Deutscher Ingenieure*, Vol. 75, July 25, 1931, p. 969 (abstract): "Lösung von Differentialgleichungen für nicht stationäre Warmeleitung mittels Differenzenrechnung," by Ernest Schmidt.

entire track surface. It will be shown now that this applies only to the bulk of the drum, about 0.005 to 0.030 in. away from the surface. For on the actual surface q is constricted to a fraction of the nominal area, because of the following points:

(a) The nominal brake-lining surface rarely covers more than two-thirds of the track, sometimes as little as one-tenth.

(b) As evidenced by the shape of lining wear, even the macroscopic pressure pattern is rarely uniform. For example, Fig. 15 shows the wear pattern of a 3 leading-shoe racing-car brake, described elsewhere.[13] Although the U-section shoes symmetrically straddled a center-backplate, even in this case there was a noticeable lateral skewing of lining wear, in addition to its well-known arcuate variation.

(c) Between solids, effective contact is generally confined to asperities of the surface. There is also very strong evidence to show[14] that with p^* as the mean yield pressure of the softer material, the load P is actually supported only over an area (A') given by:

$$A' \approx P/p^* \tag{21}$$

With p as the nominal pressure on the friction material, the contact efficiency (Y) can be thus defined as:

$$Y = A'/A = p/p^* \tag{22}$$

Few brake drums remain straight when hot, this is where ribs or flanges exert a great effect, so that in the general case it does not seem possible to estimate T_c without some involved experiments. Its order of magnitude, and the main factors affecting it may be approximated, however – at least qualitatively – by considering a simplified model.

To this end let it be supposed that q passes into the drum through n equidistant circular spots, each having the same diameter (d) which is assumed to be small compared with the dimensions of the track. It therefore follows that:

$$A' = \tfrac{1}{4}\pi n d^2 \tag{23}$$

and that the d spots can be treated as if lying on the boundary of a semi-infinite slab. It can be also safely assumed that for the usual nonmetallic friction materials $k = c = 0$, and p^* is small compared with the yield strength of the drum material. The drum tends to plow thus through the brake lining, which is consistent with the fact that the wear rates of the two usually differ by a factor of about 100. Once initiated, such d spots would grow as they receive heat, and plough through a distance U during a time (t_c). As the latter is small, q remains sensibly constant, and the heat received in unit time by each d spot is qA/n. On further supposing

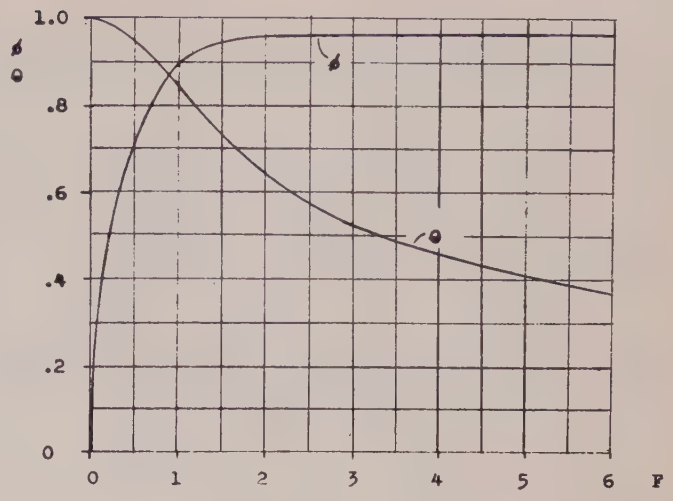


Fig. 12 – Thermal saturation (ø) and dynamic heat storing efficiency θ as a function of Fourier's number

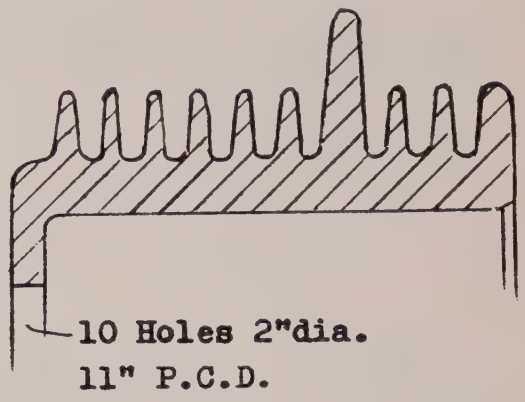


Fig. 13 – Cross-section of 14 x 2 1/4-in. cast-iron (racing car) drum for 3-leading-shoe brakes

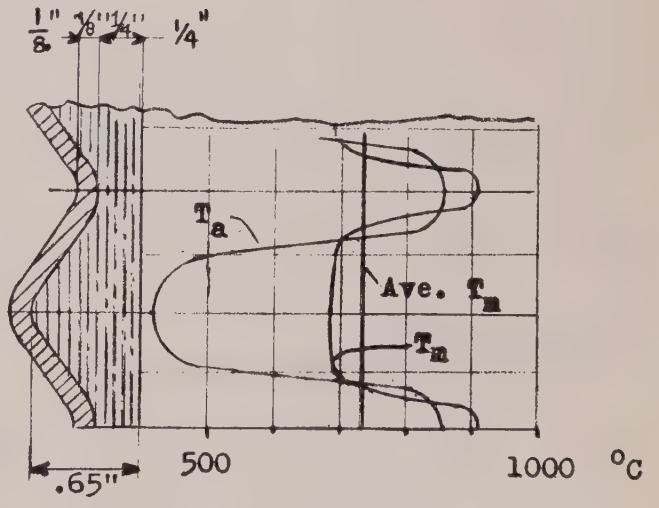


Fig. 14 – Maximum and average temperature across width of 15 1/2 x 5-in. corrugated cast-iron (aircraft) drum, for heat input $H = 20$ Btu/in.^{2} during $t^* = 15$ sec

[13] See *Automobile Engineer*, Vol. 41, February, 1951, pp. 53-60: "Brake Design – A Detailed Investigation of Fade Sensitivity," by G. A. G. Fazekas.

[14] See pp. 20, 51 of "The Friction and Lubrication of Solids," by F. P. Bowden and D. Tabor. Oxford, Clarendon Press, London, 1950.

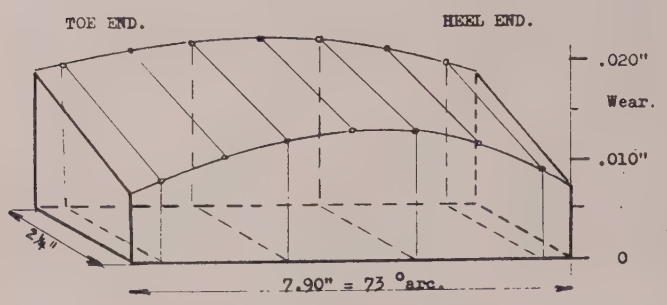


Fig. 15 – Wear pattern of U-section shoe after 196 stops at 227,000 ft-lb energy input, at 400 F mean drum temperature, against drum shown in Fig. 13

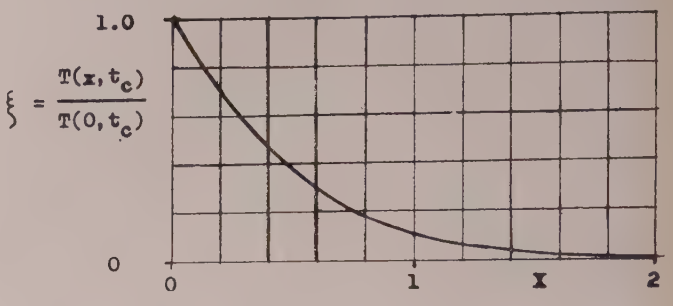


Fig. 16 – Depth of flash temperature penetration

that these d spots are too far from each other to exert a significant interference during t_c, as is often the case with well-designed brakes, at their centers the maximum local flash temperature is[15]:

$$T_s = \frac{2}{\pi} \frac{qA}{nkd} B \tag{24}$$

where:

$$B = \frac{1}{\sqrt{\pi}} \left\{ \frac{1 - e^{-Z^2}}{Z} + \sqrt{\pi}\ \text{erfc}\ Z \right\} \tag{25}$$

and:

$$Z = \frac{d}{4\sqrt{at_s}} \tag{26}$$

When $Z > 7/8$, that is, for $t_c \to 0$, equation 25 can be approximated by:

$$B \approx 1/Z\sqrt{\pi} \tag{27}$$

so that:

$$T_c \approx \frac{2}{\sqrt{\pi}} \frac{q}{Y} \frac{\sqrt{t_c}}{\sqrt{kcw}} \tag{28}$$

The value of t_c can be now approximated by writing:

$$t_c = U/v \tag{29}$$

where v = rubbing velocity, and:

$$U = \beta d \tag{30}$$

β being in the order of 2 to 6.[8]

Ultimately, equation 28 results thus in:

$$T_c \approx \frac{2}{\sqrt{\pi}} \frac{q}{\sqrt{v}} \frac{1}{\sqrt{kcw}} \frac{\sqrt{\beta a}}{Y} \tag{31}$$

[15] See Proceedings of the General Discussion on Lubrication and Lubricants, Institution of Mechanical Engineers, London, October, 1937, Vol. 2, pp. 222-235: "Theoretical Study of Temperature Rise at Surfaces of Actual Contact Under Oiliness Lubricating Conditions," by H. Blok.

Actually, equation 28 coincides with the exact solution for a semi-infinite slab receiving a constant flux at its boundary.[11] For the latter the temperature of a point (x) distant from the boundary is at a time (t):

$$T_{(x,\,t)} = \frac{qx}{k} \left(\frac{e^{-X_2}}{X\sqrt{\pi}} - \text{erfc}\ X \right) \tag{32}$$

where:

$$X = \frac{x}{2\sqrt{at}} \tag{33}$$

The surface temperature is therefore:

$$T\ (0,\ t) = \frac{2q\sqrt{at}}{k\sqrt{\pi}} \tag{34}$$

The depth of penetration during t_c can be expressed thus as:

$$\xi = \frac{T\ (x,\ t_c)}{T\ (0,\ t_c)} = e^{-X^2} - X\sqrt{\pi}\ \text{erfc}\ X \tag{35}$$

a plot of which in Fig. 16 shows that $\xi = 1.50\%$ for $X = 1.50$, so that the thickness of the skin affected by T_c can be taken as:

$$x^* = 3\sqrt{at_c} \tag{36}$$

which is obviously, an upper limit.

In equation 31, the second term is entirely fixed by the duty, and the size of the brake; the third term depends only on drum material; whereas the fourth one is largely determined by the physical properties of the friction material. To show more explicitly the effects of drum diameter (D), co-

efficient of friction (μ), and other physical aspects, the following have to remain constant:

$$\begin{aligned} &\text{Heat input} && = Q = qA \\ &\text{Torque} && = \mu pAD/2 \\ &\text{Angular velocity} && = 2v/D \end{aligned} \qquad (37)$$

With

$$d = \sqrt{\frac{4AY}{\pi n}}$$

It follows then from equation 31 that:

$$T_c \propto \frac{q}{\sqrt{vkcw}} \left(\frac{A}{Y^2 n}\right)^{1/4} \propto \frac{(\mu p^*)^{3/4} (D/n)^{1/4}}{\sqrt{kcw}} \qquad (38)$$

To minimize T_c the fundamentals desired are thus that (1) soft (2) low friction materials should be used, with (3) uniform pressure distribution over the whole track surface – that is, n should be large – and (4) that for a given (A) small diameter wide brakes seem preferable. It is also desirable that (5) the friction material should have a sharp fade, for example, around 700 C, limiting thereby the maximum temperature to about this figure. The reason is because flash temperatures do not exceed the melting point of the material,[14] and a pronounced friction fade has of course a similar effect. Concerning the drum material (6) the value of kcw should be high, but strictly this applies to a shallow surface skin only.

This mechanism of piecemeal wear is further influenced by (a) surface finish and (b) shape of drum deflection, (c) rigidity and (d) tendency of the brake shoe to pick-up, and (e) size of the friction surface, for the bigger it is the more it tends to rest on 'three legs' only.

In spite of the sweeping generalizations underlying the above – which did not take into account the effect of T_s on p^*, or the ploughing of the drum, and so forth – there is strong supporting evidence on each point. Concerning (1) the writer found that about a 3/1 change in p^* can cause over 100 C reduction of T_c. Again (2) and (5) explain why low friction linings prone to fade (as used extensively some years ago) seldom caused thermal failure; and conversely, why the acute trouble with heat checks and crazing dates from the introduction of hard high-friction materials with excellent antifade properties, and which often embody steel chips or the like as a filler. Items (3), (4), (a), and (b) are well-known empirical facts. The effect of (c) is so great that it is reflected even by the macroscopic wear pattern of the shoe.[16] About (d) the writer's evidence (unpublished) is conclusive, and regarding (e) there seems to be strong indication that for each brake there is an optimum size of friction surface.[17]

Finally, for the case considered before, in Figs. 8 and 10, $T_m + T_c$, x^*, and the nominal temperature rise:

$$T_n = 2\,T_s \left(1 - \frac{t}{2t^*}\right) \frac{t}{t^*} \qquad (39)$$

are plotted in Fig. 17, based on the following data:

$H = 4.50$ Btu/in.^{2}
$t^* = 4.25$ sec
$q = 2.12$ Btu/in.^{2} sec
$v_o = 35$ mph
$kcw = 0.105 \left(\frac{\text{cal}}{cm^2 \times C}\right)^2 \frac{1}{\text{sec}}$
$\mu = 0.38$
$p = 133$ psi
$p^* = 1800$ psi
$\beta = 2.5$
$d = 1/4''$

Assuming then that all the above factors remain constant during t^*, we have:

$$T_c = 121 \sqrt{1 - t/t^*} \text{ deg C}$$

and:

$$x^* = \frac{0.0125}{\sqrt{1 - t/t^*}} \text{ in.}$$

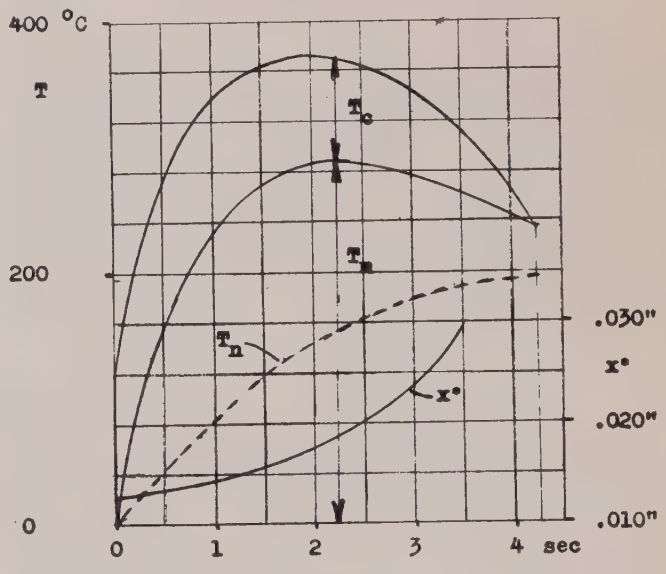


Fig. 17 – Resultant surface temperature for $T' = 0$ of passenger-car drum in Fig. 8, and depth of flash temperature penetration

[16] See Proceedings of the Institution of Automobile Engineers, Vol. 41, 1946, pp. 419-424: Discussion by G. A. G. Fazekas of paper, "Some Problems in the Design of Braking Systems," by F. A. S. Acres.

[17] See SAE *Journal* (Transactions), Vol. 54, May, 1946, pp. 205-214: "Brake Designs and Methods of Rating Brakes," by R. K. Super.

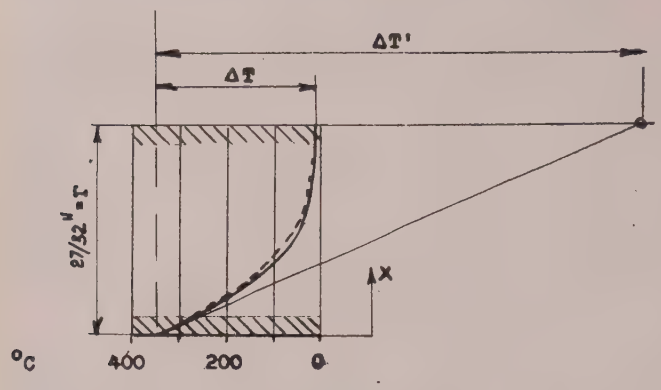


Fig. 18 – Approximation of an isochronic temperature gradient from Fig. 6 by a power curve and geometrical construction to obtain power index

It may be of interest to consider that if a lining with $p^* = 3 \times 1800 = 5400$ psi were fitted to the same brake, under the same duty, by equation 38, T_c would increase $3^{0.75} \approx 2.3$ fold. That is, the maximum surface temperature would increase by approximately 130 C with n remaining the same, which is probable.

Let us suppose now that the above brake would be fitted to a much slower vehicle, $v_o = 20$ mph, but otherwise would have to absorb the same H in the same t^*. With pv_o equal to constant it follows then from equation 31 that:

$$T_c \propto \frac{\sqrt{d}}{Y\sqrt{v}} \propto (p^*/p)^{3/4} v^{-1/2} \propto v^{1/4} \tag{40}$$

and again on the assumption that n remains unaltered. The reduction in v_o would thus lower T_c by $100 \times (20/35)^{1/4} = 13\%$ only.

It will be also evident from Fig. 17 that x^* is quite shallow so long as the d spots are small and do not lie in close clusters, as is the case with well constructed brakes. Surface phenomena, stresses included, can be then treated as separate from, and superposed upon, the bulk.

Part II – Heat Stresses

The previously described temperature gradients in the bulk and in the skin, respectively, give rise to heat stresses there. The bulk stresses themselves are of three kinds: (1) those that would arise in an infinitely long cylinder remote from the ends, (2) those due to end-effects and (3) to flange effects. Because only deformations within the elastic range will be considered, (1), (2), and (3), plus the skin stresses, can be superposed upon those of mechanical origin, and will be treated separately below.

Similarly as before, the drum will be again assumed as a short cylinder of uniform thickness (r), free at both ends. It will be further supposed that the material is homogeneous, isotropic, has a linear heat expansion, and that its elastic modulus (E) is at any given instant constant across r, corresponding to the mean temperature of the section.

Concerning temperature gradients in the bulk, as before it will be assumed that they are symmetrical to, and do not vary along the axis of the cylinder. Let ΔT be denoted here as the temperature difference across r at a time t, and with T the temperature rise above the level at r of points x distant from the surface, the temperature gradients can be well approximated by:

$$T = \Delta T \left(1 - \frac{x}{r}\right)^s \tag{41}$$

where s can be adjusted to have first order osculation at $x = 0$, and also at $x = r$. It follows then that with:

$$\frac{dT}{dx} = -\frac{\Delta T s}{r}\left(1 - \frac{x}{r}\right)^{s-1} \tag{42}$$

we have:

$$\left(\frac{dT}{dx}\right)_{x=o} = -s\Delta T/r = -q/k$$

and:

$$\left(\frac{dT}{dx}\right)_{x=r} = 0 \tag{43}$$

so that:

$$s = qr/k\Delta T = \Delta T'/\Delta T \tag{44}$$

from the geometrical construction in Fig. 18, where the full line shows an isochronic gradient taken from Fig. 6, and the broken one shows its approximation by equation 41. The agreement is seen to be quite satisfactory, but only so long as no points of inflection occur; this condition is always satisfied with thin sections, but for heavy ones the above approximation has to be limited to $0 < t < \frac{3}{4}\, t^*$. This covers the critical phase anyhow.

II. 1. Bulk Stresses Remote from the Ends

Bulk Stresses Remote from the Ends – In considering an infinitely long cylinder of inner radius (a) and outer radius (b), for points at a radius $y = a + x$, the radial, tangential and axial stresses are:[18]

$$S_r = \frac{uE}{1-m}\left(\frac{I_b}{b^2 - a^2}\left(1 - \frac{a^2}{y^2}\right) - \frac{I_y}{y^2}\right) \tag{45}$$

$$S_t = \frac{uE}{1-m}\left(\frac{I_b}{b^2 - a^2}\left(1 + \frac{a^2}{y^2}\right) + \frac{I_y}{y^2} - T\right) \tag{46}$$

[18] See *Zeitschrift des Vereines Deutscher Ingenieure*, Vol. 51, May 11, 1907, pp. 743-747: "Temperaturspannungen in Hohlzylindern," by Rudolf Lorenz.

$$S_z = \frac{uE}{1-m}\left(\frac{2mI_b}{b^2-a^2} - T\right) + Ee_z \tag{47a}$$

where m = Poisson's ratio, u = coefficient of heat expansion, e_z = axial strain, and:

$$I_y = \int_a^y Tydy \tag{48}$$

with I_b denoting the value of I_y for $y = b$. The value of e_z follows now from the condition that there being no axial force:

$$2\pi \int_a^b S_z y dy = 0$$

that is:

$$e_z = 2u\frac{I_b}{b^2-a^2} \tag{49}$$

We have, therefore:

$$S_z = \frac{uE}{1-m}\left(\frac{2I_b}{b^2-a^2} - T\right) \tag{47b}$$

To integrate I_y we substitute equation 41 for T, and on writing in the latter $b - y = r - x$, we obtain:

$$I_y = \frac{T}{r^s}\left(\frac{(b-y)^{s+2} - r^{s+2}}{s+2} - b\frac{(b-y)^{s+1} - r^{s+1}}{s+1}\right) \tag{50}$$

For $y = b$ this reduces to:

$$I_b = \frac{\Delta T r}{(s+1)(s+2)}\left(a + b + sa\right) \tag{51}$$

so that one can write:

$$I_y = -\frac{T(b-y)}{(s+1)(s+2)}(b + y + sy) + I_b = I_b - I_y' \tag{52}$$

On substituting these into equation 45, we have:

$$S_r = \frac{uE}{1-m}\frac{1}{(s+1)(s+2)}$$
$$\left(\Delta T(1 + \frac{sa}{a+b})(1 - \frac{b^2}{y^2}) + T(\frac{b}{y} - 1)(\frac{b}{y} + 1 + s)\right) \tag{53a}$$

Considering that for most brake drums $r/a < 0.07$, little error will be incurred by simplifying equation 53a to:

$$S_r \approx -\frac{uE}{1-m}\frac{T}{s+1}\left(\frac{b}{y} - 1\right)\left(1 - \frac{(b-y)^s}{r^s}\right) \tag{53b}$$

Quite analogously, we find also that:

$$S_t = \frac{uE}{1-m}\frac{1}{(s+1)(s+2)}\left(\Delta T(1 + \frac{sa}{a+b})(1 + \frac{b^2}{y^2})\right.$$
$$\left. - T(s+1)(s+2) - T(\frac{b}{y} - 1)(\frac{b}{y} + s + 1)\right) \tag{54a}$$

which can be again well approximated by:

$$S_t \approx \frac{uE}{1-m}\frac{\Delta T}{s+1}\left(1 - (1+s)(1 - \frac{x}{r})^s\right) \tag{54b}$$

On comparing now equations 45, 46, and 47b, it follows that:

$$S_t + S_r = S_z \tag{55}$$

and since S_t and S_r differ by a factor of 100, for practical purposes $S_t \approx S_z$.

To obtain the diametral heat expansion $\Delta D/D$), we have to consider that the tangential strain e_t is:

$$Ee_t = S_t - m(S_r + S_z) + uET \tag{56}$$

For $x = 0$ this reduces to:

$$\frac{\Delta D}{D} = (e_t)_{x=0} \approx u\Delta T/(s+1) \tag{57}$$

Considering that the mean temperature of the section is, if the curvature be neglected:

$$\frac{1}{r}\int_a^b Tdy = \Delta T/(s+1) \tag{58}$$

it follows that at least within the elastic range, and to the first order of small quantities, the diametral heat expansion is not influenced by any gradients.

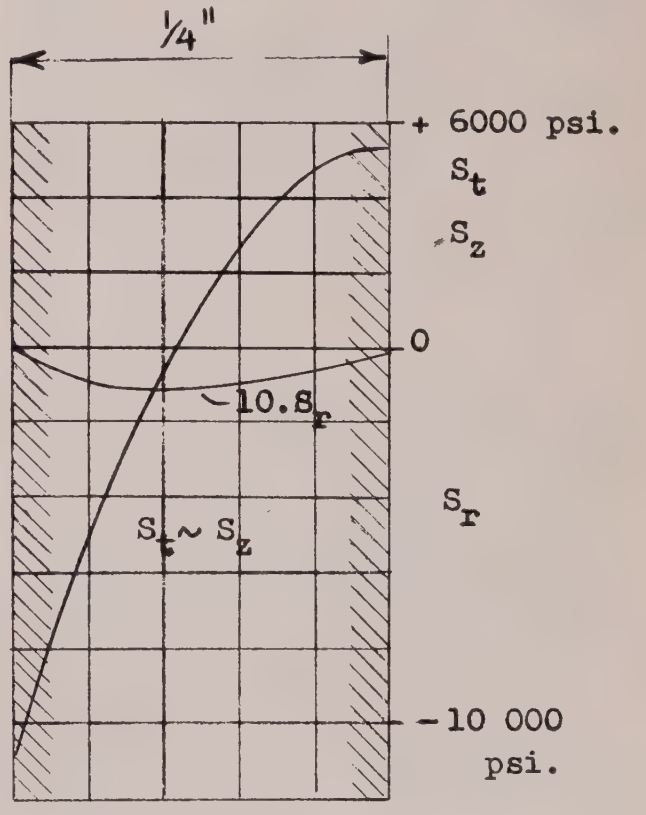


Fig. 19 – Tangential, radial, and axial stress distribution in a 1/4-in. thick cast-iron drum, subject to a temperature difference $\Delta T = 100$ C, with a square law profile

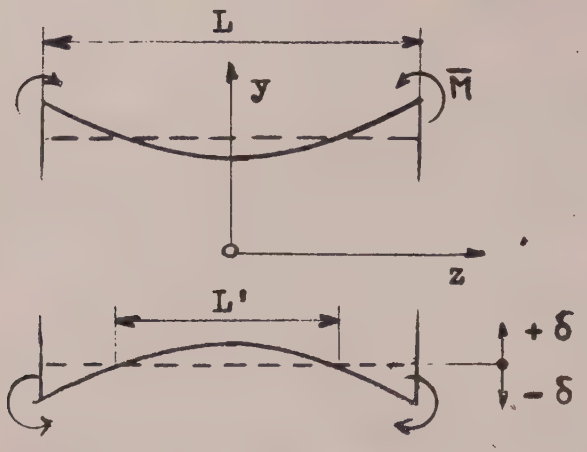


Fig. 20 – Shape of drum deflection due to end-effects

which will be later shown as about correct for the critical phase.

We obtain thus:

$$S_r = -5400\left(\frac{4.25}{4+x} - 1\right)\left(1 - \left[1 - \frac{x}{\frac{1}{4}}\right]^2\right)$$

and:

$$S_t = S_z = +5400\left(1 - 3\left[1 - \frac{x}{\frac{1}{4}}\right]^2\right)$$

which are plotted in Fig. 19. As will be seen the maximum values of S_z and S_r differ by a factor of about 100, so that in what follows S_r will be neglected.

It will also be seen that at $x = 0$ the compressive stress is twice as great as the tensile stress at $x = r$, and that the neutral layer is at approximately $x = 0.423\,r$.

In practice, however, $\Delta D/D$ will be less than given above, even within the elastic range, owing to the variation of E across r.

To show the stress distribution across r, let it be supposed that:

$$E = 10\text{x}10^6 \text{ psi}$$
$$a = \tfrac{1}{2}D = 4\text{-in.}$$
$$m = 0.26$$
$$r = \tfrac{1}{4} \text{ in.}$$
$$u = 0.000012/\text{deg C}$$
$$\Delta T = 100 \text{ C}$$

and: $s = 2$

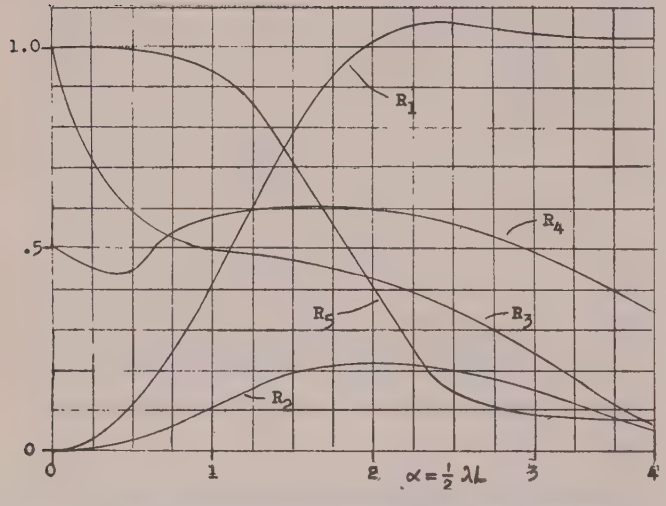


Fig. 21 – Factors characterizing the warping and stresses generated in a short cylinder, subject to a radial temperature gradient to a power curve

II. 2. Bulk Stresses Due to End-Effects

Bulk Stresses Due to End-Effects – For calculating the stresses in a cylindrical shell with free ends, the previous expressions apply only to parts about one diameter distant from the ends. For at the ends S_z gives rise to a couple $\overline{M}$ about uniformly distributed around the edges.[19] On neglecting the curvature, this couple is, per unit length of periphery:

$$\overline{M} = \int_0^r S_z x \cdot dx = \frac{uE}{1-m}\Delta T r^2 \frac{S}{2(s+1)(s+2)} \qquad (59)$$

and deforms the drum in a manner indicated in Fig. 20. To find this deformation, an infinitesimally narrow axial section of the shell may be considered as a short beam of length (L) on an elastic foundation.[20] Taking the middle as the center of coordinates, as in Fig. 20, it can be shown that the radial deflection (δ) in planes $(\pm z)$ is:

$$\delta = -\frac{\overline{M}a}{Er^2}\,2\sqrt{3(1-m^2)}$$
$$\left(\frac{c\cdot sh + s\cdot ch}{s\cdot c + sh\cdot ch}\,\overline{s\cdot sh} + \frac{c\cdot sh - s\cdot ch}{s\cdot c + sh\cdot ch}\,\overline{c\cdot ch}\right) \qquad (60)$$

where:

$$s,\ sh,\ \cdots = \sin\alpha,\ \sinh\alpha,\ \cdots$$
$$\overline{s},\ \overline{sh},\ \cdots = \sin\lambda z,\ \sinh\lambda z,\ \cdots \qquad (61)$$
$$\lambda^4 = 3(1-m^2)/(ar)^2$$
$$\alpha = L\lambda/2$$

[19] See p. 374 of "Theory of Elasticity," by S. P. Timoshenko. McGraw-Hill Book Co., New York, first edition, 1934.

[20] See p. 399 of "Theory of Plates and Shells," by S. P. Timoshenko. McGraw-Hill Book Co., New York, first edition, 1940.

On denoting the second bracketed expression of equation 60 as R, and combining it with equation 59, we obtain ultimately:

$$\delta = -\frac{s\sqrt{3}}{(s+1)(s+2)}\left(\frac{1+m}{1-m}\right)^{1/2} au\Delta TR \qquad (62)$$

At the ends the deflection is therefore, for $z = \frac{1}{2} L$.

$$\delta^e = -\frac{s\sqrt{3}}{(s+1)(s+2)}\left(\frac{1+m}{1-m}\right)^{1/2} au\Delta TR_1$$

where:

$$R_1 = \frac{sh\cdot ch - s\cdot c}{sh\cdot ch + s\cdot c} \qquad (63)$$

Similarly, in the center, for $z = 0$, we have:

$$\delta_o = \frac{s\sqrt{3}}{(s+1)(s+2)}\left(\frac{1+m}{1-m}\right)^{1/2} au\Delta TR_2$$

where:

$$R_2 = \frac{c\cdot sh - s\cdot ch}{sh\cdot ch + s\cdot c} \qquad (64)$$

Alternatively, δ_o may be expressed as:

$$\frac{\delta_o}{\delta^e} = R_3 = \frac{s\cdot ch - c\cdot sh}{sh\cdot ch - s\cdot c} \qquad (65)$$

Between $z = 0$ and $z = L/2$, δ vanishes for $z = L'/2$, as sketched in Fig. 20. With $\alpha' = L'\lambda/2$, the corresponding value of L' follows from:

$$\tan\alpha' \tan h\,\alpha' = \frac{s\cdot ch - c\cdot sh}{s\cdot ch + c\cdot sh} \qquad (66a)$$

a graphical solution of which yields explicitly:

$$L'/L = R_4(\alpha) \qquad (66b)$$

which is plotted together with R_1, R_2, and R_3 in Fig. 21. This shows that (1) as the width of a drum is increased, δ_e increases very rapidly within the practically important range, that (2) roughly, $\delta_o \approx \frac{1}{2}\delta_e$, and (3) $L' \approx 0.6L$, so that (4) any ribs or fins situated at L' may well constrain diametral expansion but would not keep the surface straight. (5) Reinforcements, or local chilling, should be therefore close to the ends, and in the center, respectively.

To appreciate the relative significance of δ, it can be conveniently compared with the diametral heat expansion (ΔD). With reference to equations 57 and 63, one obtains then:

$$\frac{\delta^e}{\Delta D} = \frac{s\sqrt{3}}{s+2}\left(\frac{1+m}{1-m}\right)^{1/2} R_1 \qquad (67)$$

With $s = 2$, and $m = 0.26$, as before, $\delta_e = 1.13\,\Delta DR_1$, that is, within the practically important range, $\frac{1}{2} < \alpha < 1\frac{1}{4}$, δ_e, and ΔD are about of the same order, being about one-third to one-fifth of the total heat expansion corresponding to $T' + T_e$.

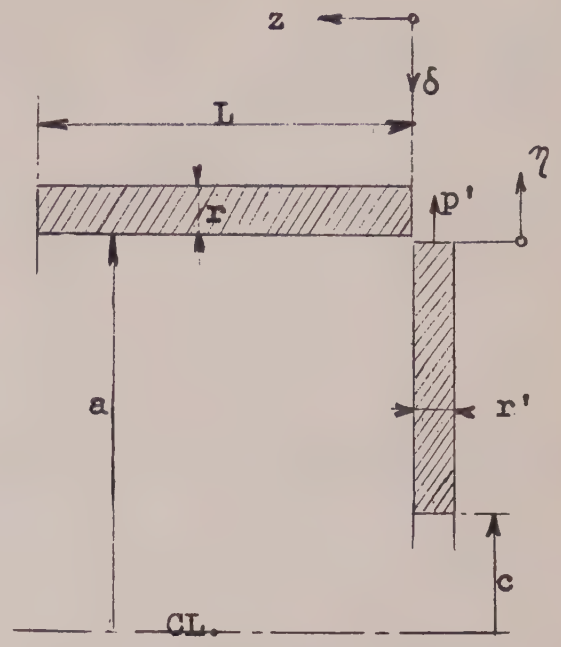


Fig. 22 – Idealized constraint of a flange

In general, δ gives rise to some hoop S_h, and bending stresses S_b. In the practically important range they tend to increase $|S_t|$ at ($z = L/2$, $x = r$) and at ($z = 0$, $x = 0$), elsewhere L, s_z, and S_t are diminished. The additional stresses at these two points can be obtained by superposing:

$$S_h \approx \frac{E}{1-m^2}\left(-\frac{\delta}{a} + m\frac{r}{2}\frac{d^2\delta}{dz^2}\right) \qquad (68)$$

upon S_t. This overestimates matters slightly, be-

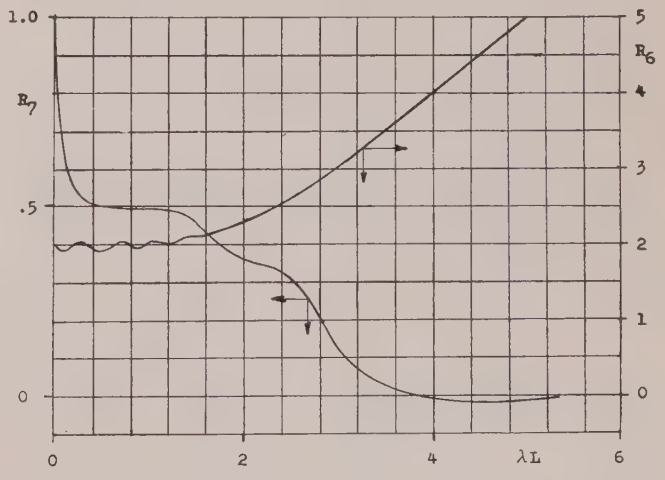


Fig. 23 – Factors characterizing deflection of a short cylinder circumferentially compressed at one end, with other end free

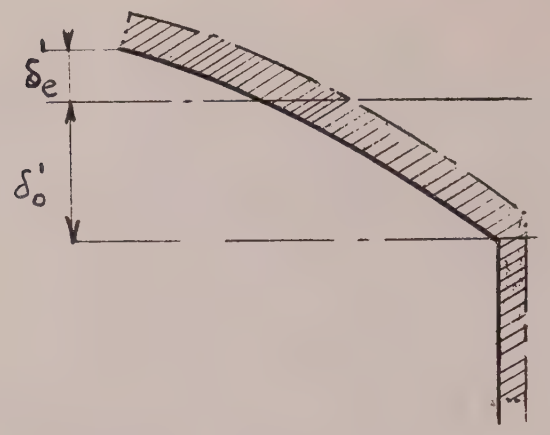


Fig. 24 – Exaggerated shape of bellmouthing caused by a rigid flange alone

cause δ should have been divided by y, and not by a. On differentiating now equation 60 twice, one obtains:

$$\frac{d^3\delta}{dz^2} = -M\frac{12\,(1-m^2)}{Er^3}\left(\frac{c\cdot sh + s\cdot ch}{s\cdot c + sh\cdot ch}\,\bar{c}\cdot\bar{ch} - \frac{c\cdot sh - s\cdot ch}{s\cdot c + sh\cdot ch}\,\bar{s}\cdot\bar{sh}\right) \quad (69)$$

so that on substituting for δ from equation 60 we have:

$$S_h = \frac{\overline{M}}{r^2}\left(2\sqrt{3/(1-m^2)}\,[\ ] \pm 6m\,[\ ]\right) \quad (70)$$

where the brackets denote the expressions in the parentheses of equations 60 and 69, the (+) and (−) signs of the second term referring to $x = 0$ and $x = r$ respectively.

$z = L/2$, $x = r$ the value of S_h is therefore:

$$S_h'' = \frac{\overline{M}}{r^2}\left(2\sqrt{3/(1-m^2)}\,R_1 - 6m\right) \quad (71)$$

Also with reference to equation 54b, at the same points:

$$\left(S_t\right)_{\frac{1}{2}L,\,r} = S_t'' = \frac{Eu\Delta T}{1-m}\,\frac{1}{s+1} \quad (72)$$

On substituting for $\overline{M}$ from equation 59, the resultant stress is therefore:

$$S_t'' + S_h'' = \frac{Eu\Delta T}{1-m}\,\frac{s}{(s+1)\,(s+2)}\cdot \left(R_1\sqrt{3/(1-m)} - 3m + 1 + 2/s\right) \quad (73)$$

Taking again $s = 2$, $m = 0.26$, and with $R_1 \sim 1$, the maximum increase of stress is in the order of 50%, diminishing rapidly as the drum width is narrowed. In fact, with narrow drums S_h vanishes when:

$$R_1 = m\sqrt{3\,(1-m^2)} \rightarrow \alpha = 0.83 \quad (74)$$

At $z = 0$, $x = 0$ the value of S_h is:

$$S_h' = -\frac{\overline{M}}{r^2}\left(2\sqrt{3/(1-m^2)}\,R_2 - 6m\,R_5\right)$$

and: $$R_5 = \frac{c\cdot sh + s\cdot ch}{s\cdot c + sh\cdot ch} \quad (75)$$

The corresponding value of S_t being:

$$\left(S_t\right)_{o,\,o} = S_t' = \frac{Eu\Delta T}{1-m}\,\frac{s}{s+1} \quad (76)$$

the resultant stress is:

$$S_t' + S_h' = \frac{Eu\Delta T}{1-m}\,\frac{s}{(s+1)\,(s+2)}\left(3m\,R_5 - 2 - s - \sqrt{3/(1-m^2)}\,R_2\right) \quad (77)$$

R_5 is also plotted in Fig. 21, and shows that only for wide drums ($\alpha > 1.45$) does a small stress increase occur; otherwise the stress relief due to curling penetrates right to the center.

It would thus seem that curling of the edges causes a serious stress increase only on wide drums with unsupported ends, tending to increase the tensile hoop stresses. Elsewhere the stresses are generally relieved, particularly on the track surface. On the other hand, the deflections, or rather their effect, is appreciable even with comparatively narrow drums. For as the track surface curls up, usually the shoes are far too rigid to follow its contour, and the d spots tend to develop thus in a narrow band. The consequent concentration of heat tends to expand the particular band, brings a fresh one into contact, and so forth, so that the drum surface is burnt up piecemeal.

Except in extreme cases, increasing r is of little help: for whereas α is admittedly somewhat diminished thereby, both s and ΔT increase rapidly with r. This explains why usually, heavy sections are much higher stressed and do not stand as high a temperature as light ones.

Finally, it may be added that the bending moment M at z being:

$$M = \overline{M}\left(\frac{c\cdot sh + s\cdot ch}{s\cdot c + sh\cdot ch}\,\bar{c}\cdot\bar{ch} - \frac{c\cdot sh - s\cdot ch}{s\cdot c + sh\cdot ch}\,\bar{s}\cdot\bar{sh}\right) \quad (78)$$

R_5 represents the value of M for $z = 0$ and $\overline{M} = 1$.

II. 3. Stresses Due to Flange-Effects

Stresses Due to Flange-Effects – The drum shown in Fig. 3 is a type almost universally employed on European motor vehicles. It is cheap, easy to manufacture, and rather rigid, which saves pedal travel. For the purpose of a rough stress analysis, in a first approximation it can be imagined as consisting of a cylindrical shell, hinged to a disc of the same elasticity, as indicated in Fig. 22. For reasons explained in the section on temperature gradients, during t^* the heat does not permeate down the flange appreciably, so that the latter constrains the expansion of the cylindrical part.

If p' is the tension uniformly distributed around the periphery of the disc (we neglect bending), it follows from the Lamé equations that the radial extension η of the periphery is:

$$\eta = a \frac{p'}{E} \left(\frac{a^2 + c^2}{a^2 - c^2} - m \right) \tag{79}$$

Considering the cylindrical part again as a short beam of unit width on an elastic foundation, loaded at one end by a force $p'\,r'$, with the coordinates as in Fig. 22, the radial deflection δ' of a point z is:[21]

$$\delta' = 2 \frac{p'r'}{Er} \lambda a^2 \left(\frac{sh}{sh^2 - s^2} \cos \lambda z \cosh \lambda (L - z) - \frac{s}{sh^2 - s^2} \cosh \lambda z \cos \lambda (L - z) \right) \tag{80}$$

$(s, sh, \ldots)$ denoting here $(\sin \lambda L, \sinh \lambda L, \ldots)$.

At $z = 0$ the deflection is therefore:

$$\delta_o' = 2 \frac{p'r'a}{ErL} aR_6$$

where:

$$R_6 = \lambda L \frac{sh \cdot ch - s \cdot c}{sh^2 - s^2} \tag{81}$$

If end-effects be disregarded for the present purpose, the mean expansion of the cylindrical part would be auT_n, and p' follows then from:

$$auT_n = 2 \frac{p'r'a}{ErL} aR_6 + a \frac{p'}{E} \left(\frac{a^2 + c^2}{a^2 - c^2} - m \right) \tag{82}$$

so that ultimately, equation 79 may be written as:

$$\eta = auT_n \left\{ 1 + \frac{\frac{r'a}{rL} 2R_6}{\frac{a^2 + c^2}{a^2 - c^2} - m} \right\}^{-1} \tag{83}$$

From Fig. 23 it will be seen that $R > 2$, and since usually $r' \approx r$, $a \approx 3\frac{1}{2}L$, and $c \approx \frac{1}{2}a$, in most practical cases η differs from auT_n by a factor of 10. In fact this is an underestimate, as the constraint due to bending has been disregarded, and the difference in E between the cold flange and the hot cylindrical part, usually is in favor of the flange, so that:

$$\delta_o' \approx auT_n$$

At $z = L$ the deflection is, if end-effects be disregarded for the present purpose:

$$\delta_e' = \delta_o' \frac{sh \cdot c - s \cdot ch}{sh \cdot ch + s \cdot c} = \delta_o' R_7 \tag{84}$$

a plot of which in Fig. 23 shows that with the conventional narrow drums $\delta_e' \approx \frac{1}{2}\delta_o'$, so that ultimately, the drum tends to deflect as sketched in Fig. 24, otherwise known as bellmouthing. (The shape of the R_7 curve is due to a quasi-resonance effect.)

Similarly, as explained in the section on bulk stresses due to end-effects, the above deflections give rise to some hoop S'_h, and bending stresses S'_b, at $x = 0$ both being additional to S_t and S_z respectively. The worst case arises for $z = 0$ where according to our simplified treatment $S_b' = 0$ so that:

$$S_h' = -E\delta_e'/a\,(1 - m^2) \approx -EuT_n/(1 - m^2) \tag{85}$$

With the data quoted in equations 53b to 58, we have then $-S_h' \approx 12{,}000$ psi/100 deg C which is greater than the corresponding maximum value of S_t. It would also follow that since with thin drums $T_a > \Delta T$, S_h' becomes there the dominant stress. This can be readily demonstrated by comparing equation 85 and 54b, and on substituting for ΔT_m from equation 15:

$$\frac{S_h'}{S_{to}} = (1 + m)\left(1 + \frac{1}{s}\right) \frac{1}{0.80\,F^{0.75}} \tag{86}$$

where S_{to} refers to $x = 0$. The value of this expression does not differ greatly from unity for the extreme values of T_n and ΔT respectively. When F is large, that is, the drum is thick, the relative stress increase is less, owing to the lower mean temperature rise, and consequently smaller heat expansion. But since T_a and ΔT_m occur almost simultaneously, the resultant stress is about double that of S_{to}. If F is small, that is, the drum is thin, the two stresses do not occur simultaneously: when one is small the other is big, and vice versa. Generally, S'_h is greater, but whether this is or is not so depends very much upon the effect of the mean temperature level on E.

With the type of drum discussed before, it is customary to provide a fin near the mouth. Usually, however, its strength and effect are quite negligible compared with those of a deep flange; also the fin is rarely in the correct position even to prevent curling of the ends. For it will be evident that the

[21] See p. 24 of "Strength of Materials," by S. P. Timoshenko. D. Van Nostrand Co., Inc., New York, second edition, 1943.

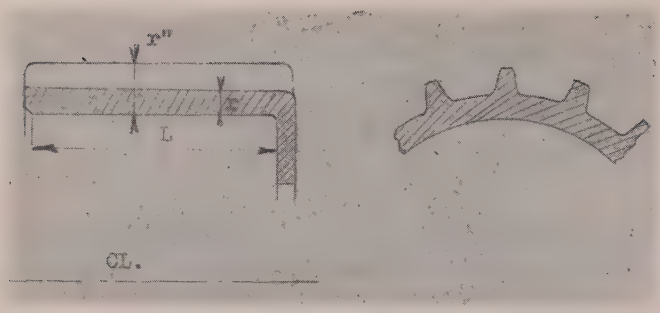


Fig. 25 – Drum reinforced with lateral fins

effect of $\overline{M}$ has to be countered where it acts, a relatively weak flange or fin at both free ends being sufficient to keep the surface straight, and to reinstate the regime of stress corresponding to S_z and S_t which is the unavoidable minimum.

However desirable it may be to use drums that remain straight whether hot or cold, with certain brakes it is necessary to lend the drum great rigidity, for otherwise the shoes may get out of control. For this reason the drum is sometimes stiffened by lateral ribs so that the section becomes somewhat star-shaped, as sketched in Fig. 25.

It is sometimes claimed that this method provides great lateral rigidity without serious additional hoop stresses. Since, however, the ribs restrict lateral expansion they give rise to local stress-raisers. The order of such stress concentrations may be roughly assessed on the premise that (1) the heat flux remains uniform over the whole drum surface, and that (2) in an axial plane of symmetry through the middle of a lateral rib the temperature field is the same as in a full cylinder of the same thickness (r''). On considering then the center part of the rib as a beam of unit width, length (L), and subject to the temperature profile given by equation 41:

$$T'' = \Delta T''\,(1 - x/r'')^s \tag{41}$$

the bending stress (S''_z) follows, from:

$$S''_z = E(e''_z - uT'') \tag{87}$$

There being no forces at the free ends, the axial strain (e''_z) follows from:

$$\int_o^{r''} S''_z\,dx = 0 = Er''\left(e''_z - u\Delta T''/(s+1)\right)$$

that is:

$$e''_z = \frac{u\Delta T''}{s+1} \tag{88}$$

so that:

$$S''_z = Eu\Delta T''\left(\frac{1}{s+1} - (1 - x/r)^s\right) \tag{89}$$

The consequent bending moment M'' is then:

$$M'' = \int_o^{r''} S''_z \cdot x \cdot dx = Eu\Delta T'' r''^2 \frac{s}{(s+1)2(s+2)} \tag{90}$$

warping the beam into a circular shape. The deflection δ''_e is at $z = L$ to the first order of small quantities:

$$\delta''_e \approx \frac{3/4\ s}{(s+1)(s+2)} \frac{L^2}{r''} u\Delta T'' \tag{91}$$

It would thus follow that lateral ribs may well impart rigidity against external loads, but do not keep the surface straight. In fact on comparing equations 63 with 91, and on substituting for ΔT_m from equation 15, we find that:

$$\frac{\delta''_e}{\delta_e} = \frac{\sqrt{3}}{4}\left(\frac{1-m}{1+m}\right)^{1/2} \frac{L^2}{ar''} \frac{\Delta T''}{\Delta T} \propto \frac{L^2}{a\sqrt{rr''}} \tag{92}$$

the value of which is in the order of 1, unless the ribs be made grotesquely deep. The price to be paid for such stiffeners follows from equation 88, showing that the compressive stress for $x = 0$ is:

$$S''_{zo} = Eu\Delta T'' \frac{s}{s+1} \tag{93}$$

and on comparing it with equation 54b, we have:

$$\frac{S''_{zo}}{S_{zo}} = (1-m)\frac{\Delta T''}{\Delta T} = (1-m)(r''/r)^{1/2} \tag{94}$$

Therefore under lateral ribs the stress can be easily three times as high as elsewhere; with highly rated brakes the track surface is liable to resemble thus an internal spur gear, and to promote severe shudder.

When the section of the ribs is trapezoidal, the warping is somewhat less, but only at the expense of still higher stresses. It would thus seem that (1) if the track surface is to remain straight, only a relatively weak annular reinforcement is called for at the ends; (2) this being a unique solution of the problem, other reinforcements, whatever their shape may be, can increase rigidity against external loads only at the cost of serious warping, and (3) stress concentrations. *Rigidity and utmost energy dissipating capacity are therefore incompatible.*

II. 4. Skin Stresses

Skin Stresses – So long as the d spots are small, well-dispersed, and the penetration of T_c is shallow, each d spot can be considered as lying on the boundary of a semi-infinite solid. The maximum compressive skin stress S_c is then:[22]

$$S_c = \frac{EuT_c}{1-m} \tag{95}$$

[22] See ASME Transactions (*Journal of Applied Mechanics*), Vol. 4, March, 1937, pp. A33-36: "Thermal Stress," by J. N. Goodier.

With:

$E = 10 \times 10^6$ psi
$m = 0.26$
$u = 0.000012$/deg C
$T_c = 100$ deg C
$S_e = 16{,}200$ psi

As T_c can easily reach several hundred deg C, temperature flashes alone can bring the material into a state of local plastic yield when abrasive resistance tends to disappear. Also since the d spots scintillate rather rapidly, it is the fatigue, and not static, strength properties of the surface skin that are of import.

II. 5. Resultant Stresses

Resultant Stresses – The importance and extent of various deformations having been explained before, two problems remain to be discussed here: (1) the amount of total skin stress on the track surface, and (2) the tensile stresses on the outer periphery. As mentioned, (1) accounts for excessive scoring or crazing, and (2) for outright fracture.

In calculating the stresses, the effect of temperature on the strength properties of the metal cannot be neglected, except with very lightly loaded brakes. This will be evident from Fig. 26 which refers to a proprietary brand of cast iron often used for brake drums. Fig. 27 shows that for this cast iron, the strength/elastic modulus ratio remains almost unchanged as the temperature is increased. This important property (which will be discussed again later) is also substantially true in regard to other metals. These data were used for the numerical work below, and on the assumption that:

(a) For calculating $S_t \approx S_z$, at any instant E was taken as constant across r, and corresponding to the momentary $T_n + T'$.

(b) For calculating S_e a value of E corresponding to the instantaneous T_s was used.

(c) m was considered constant = 0.26.

(d) The various stress components were superposed, which is strictly valid for the truly elastic state only. Nevertheless even in the plastic state such stress figures are reasonably informative concerning the extent of yield.

(e) As before, the heat of recalescence has been neglected.

Figs. 28 and 29 show the stresses in some aircraft drums with temperature fields as in Figs. 6 and 7. In both cases H and t^* are the same so as to show the effect of a threefold variation of r. As such drums have usually weak flanges, they have little inherent tendency to bellmouth; for this reason only the unavoidable stresses – namely, $S_z \approx S_t$ and S_e – were calculated. For the latter it was assumed that at $t = 0$ the statistical average is $T_e = 140$ C, corresponding to 84 C/(Btu/in.^{2} x sec), again the same in both cases. The broken line shows s, plotted up to the commencement of an inflection in the

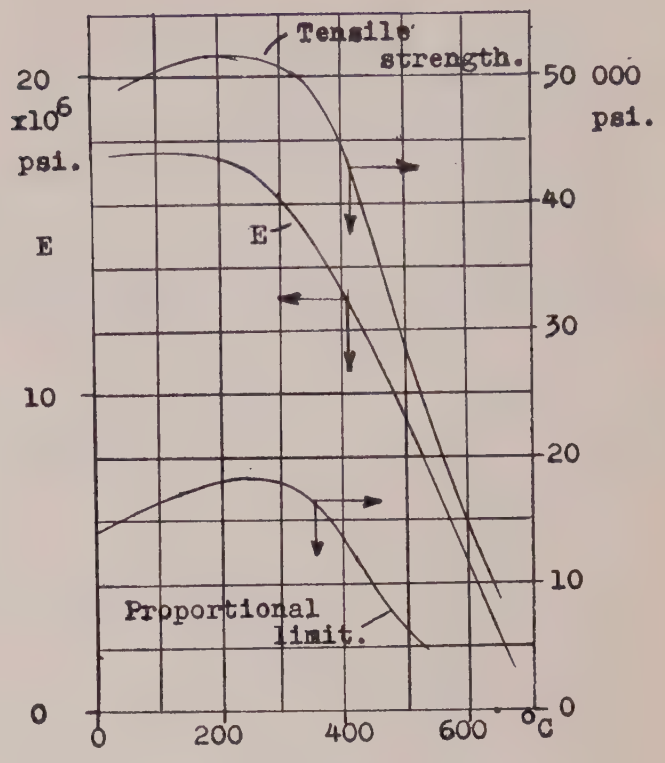


Fig. 26 – Strength properties of a typical cast iron as a function of temperature

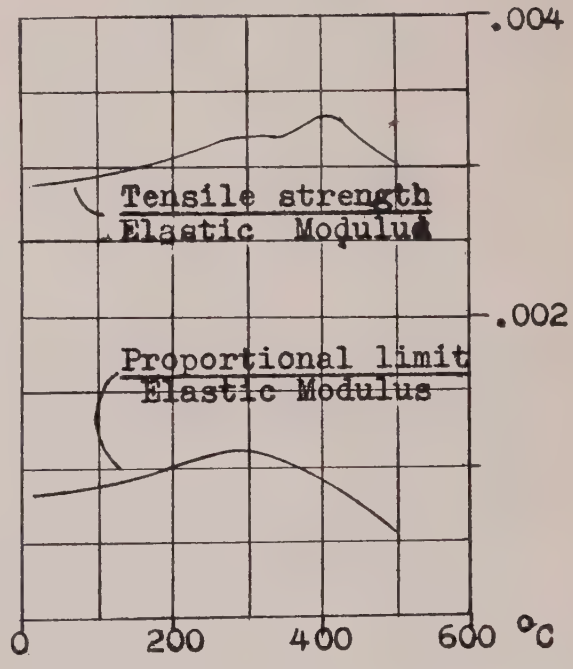


Fig. 27 – Effect of temperature on relative strength properties of cast iron in Fig. 26

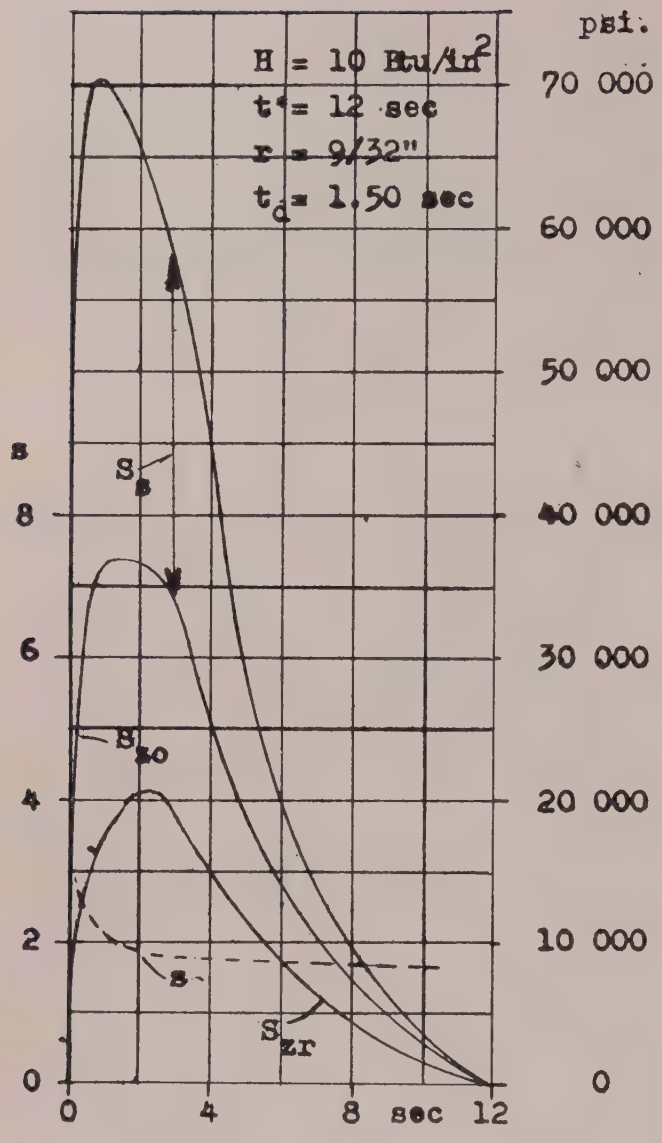


Fig. 28 – Skin stresses and variation of power index (s) during a 12-sec braking in drum shown in Fig. 7

temperature curve; for the few unimportant points remaining an extrapolation has been used. S_c has been plotted, however, only for the case in Fig. 28, showing that the resultant stress is outside the elastic region.

Figs. 30 and 31 refer to the passenger-car brake drums previously treated in Figs. 7 and 8, respectively. In both cases T_c and t^* are the same, showing the effect of a 50% increase in r and H. The ¼-in. drum was assumed to have a heavy flange, similarly as in Fig. 3, whereas the ⅛-in. drum was supposed to represent the ideal case when the flange keeps the drum perfectly straight. S_c was calculated on the same basis as before, namely, that the statistical average of T_c at $t = 0$ is again 84 C/(Btu/in.² x sec). It may be of interest to note that the case shown in Figs. 8 and 30 can correspond, for example, to some 9 x 1¼-in. front brakes of a 2230-lb vehicle that is braked to standstill from 60 mph at 65%g deceleration, with a 60/40% front/rear braking ratio.

These examples show that with quite moderately loaded drums the skin stress on the ID is well beyond the elastic range. The tensile stresses on the OD can be severe but considerably less, and the constraint of a heavy flange can suppress them. A heavy flange may help therefore in overcoming fracture if it originates from the OD but only at the expense of increasing the compressive stresses on the track surface. As the latter are usually sufficiently severe to cause a yield and set on the ID, when the drum cools the stresses are reversed. This explains why heavily loaded drums shrink. Also, when the permanent set on the ID is greater than the maximum elongation in tension, the skin will be torn in numerous hairline cracks, known as crazing.

It has been also shown that, other things being equal, thick drums are far more heavily stressed than thin ones, and that the maximum stress occurs at $t \approx t_d$, when $s \approx 2$. The plots of s further show that during $t > t_d$ it does not fall much below 2. Since its effect on S_z is anyhow small, it follows that roughly $S_z/E \propto \Delta T$, and that the highest stresses should arise for $t \sim t_d$, as they indeed do. To derive for this case explicit formulas, first it will be shown that in all practical cases $s \sim 2$. For it follows from equation 44 that here:

$$s = \frac{q_o(1 - t_d/t^*)}{k\Delta\tau_m . H/cwr} = 2F \frac{1 - t_d/t^*}{\Delta\tau_m} \qquad (96)$$

which is plotted in Fig. 32 from the data in Fig. 11. As is seen, in the practically important range $s \sim 2$ within about 5%. For $F > 6$, however, when $\Delta T_m \to T_m$, and $t_d \to \frac{1}{2}t^*$ it follows with equation 16 that:

$$= \tfrac{1}{2} q_o \frac{\sqrt{kcwt^*}}{Hk\sqrt{N}} = \sqrt{F/N} \qquad (97$$

The unavoidable bulk stresses on the inner and outer surface are therefore from equation 15 and 54b:

$$S_{zo} = \frac{2}{3} \Delta T_m \frac{Eu}{1 - m} = 0.267\, q_o \frac{Eut^{*1/4} \cdot \sqrt{r}}{(1 - m)(cwk^3)^{1/4}} \qquad (98)$$

and:

$$S_{zr} = \tfrac{1}{2} S_{zo} \qquad (99)$$

both expressions being valid for about $\frac{3}{8} < F < 4$ where the approximations for ΔT_m and s hold good. It further simplifies matters that for E a value corresponding to room temperature can be safely used with ferrous drums: because with thin drums ΔT_m occurs too early for the drum to exceed about 300 C where the metal commences to soften, and with thick drums T_n is low anyhow.

For $F > 6$, when

$$\Delta T_m \to T_m'' \quad \text{and} \quad s \to \sqrt{F/N},$$

the above expressions tend to the limit:

$$\left(S_{z\circ}\right)_\infty = \frac{Eu}{1-m}\,\tfrac{1}{2}\,q_\circ\left(\frac{Nt^*}{kcw}\right)^{\frac{1}{2}}\frac{\sqrt{F/N}}{1+\sqrt{F/N}} \to \tfrac{1}{2}\,q_\circ\frac{Eu}{1-m}\left(\frac{Nt^*}{kcw}\right)^{\frac{1}{2}} \quad (100)$$

$$\left(S_{zr}\right)_\infty = \frac{Eu}{1-m}\,\tfrac{1}{2}\,q_\circ\left(\frac{Nt^*}{kcw}\right)^{\frac{1}{2}}\frac{1}{1+\sqrt{F/N}} \to 0 \quad (101)$$

It is interesting to note too that within $\frac{3}{8} < F < 4$:

$$\frac{\left(S_{z\circ}\right)_\infty}{S_{z\circ}} = \frac{3\sqrt{N}}{1.6}\,F^{-\frac{1}{4}} \approx \tfrac{1}{2}\,F^{-\frac{1}{4}} \propto r^{-\frac{1}{2}} \quad (102)$$

so that no stress relief can be obtained on the ID by increasing r. The doubtful benefits of very thick drums are thus manifest only in lowering mechanical stresses, and S_{zr}. Actually, however, S_{zr} tends to increase up to about $F = 16$, so that apart from freak cases r should be diminished for stress relief!

Another interesting point is that $S_z \propto q_o t^{*0.25} z \sim q_o$ showing that the customary hp/in.^{2} rating of brakes has a sound physical basis so far as heat stresses are concerned. The same applies to the skin stresses S_c as well. For with reference to equations 31 and 95:

$$S_c = \frac{2}{\pi}\,\frac{Eu}{1-m}\,\frac{1}{\sqrt{kcw}}\,\frac{q_\circ}{\sqrt{v_\circ}}\,\frac{\sqrt{\beta d}}{Y}\,\sqrt{1-t/t^*} \quad (103)$$

that is, again $S_c \propto q_o$. But the hp/in.^{2} rating certainly does not apply to S'_h which is often the dominant stress with thin drums, its maximum value being at $t \doteq t^*$

$$S'_h = (EuT_a)/(1-m^2) \quad (104)$$

when the flange is rather rigid.

As intimated in the section on resultant stresses, the skin stresses on the ID reach well into the plastic state so that the previous formulas should be interpreted as indicating merely the extent of plastic yield. In this interpretation too the worst case roughly coincides with the maximum stress at $t \approx t_d$, because as shown in Fig. 27 the ratio of proportional limit/elastic modulus varies little with temperature. For this reason, in the absence of other strength data, those pertaining to room temperature may be used without incurring a great error.

II. 6. Heat Check Formation

Heat Check Formation – The foregoing has shown that heat stresses alone can bring about severe yield on the track surface when the latter's abrasive resistance tends to vanish. This explains fully the smeared appearance of heavily loaded tracks, heavy scoring even if the mechanical loads are comparatively small, and the origin of crazing. But it does not account for the onset of symmetrically-spaced heat checks that can often appear long before the rest of the surface is seriously damaged. For a tentative explanation let us consider now the heat stresses in a thin drum with a stiff flange, as shown, for example, in Fig. 3.

Referring to Fig. 31, evidently, in a thin drum S'_h can be much greater than S_{tr}. It follows then that adjacent to the flange, up to about $z = \frac{1}{2}L$,

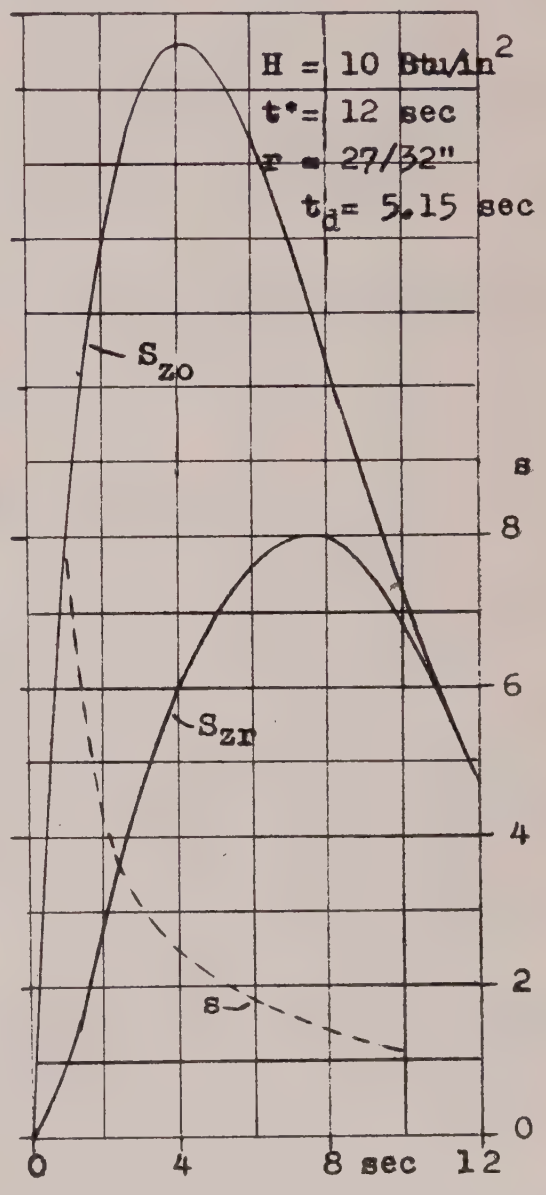


Fig. 29 – Variation of bulk stresses on inner and outer periphery of drum shown in Fig. 6 and appropriate power index (s) during a 12-sec braking

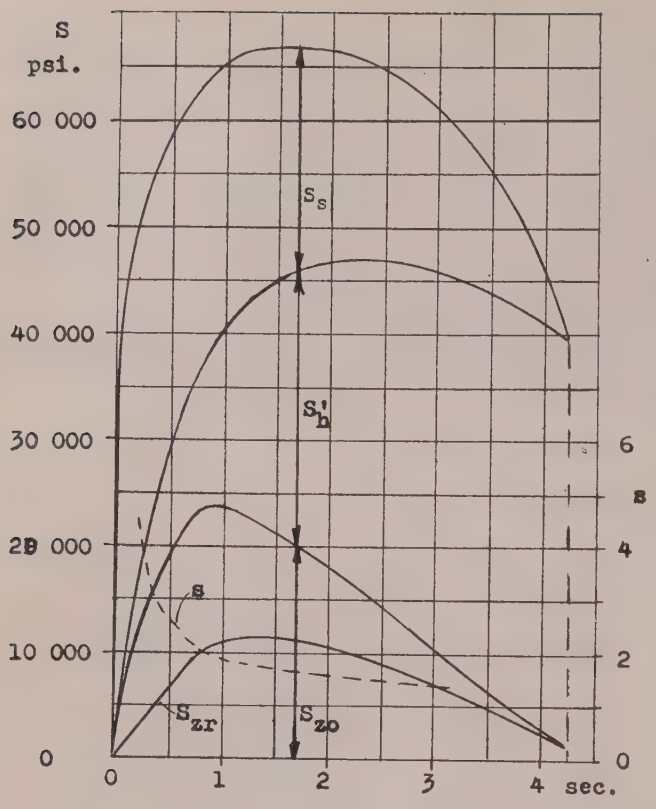


Fig. 30 – Surface stresses – due to temperature gradients, constraint of rigid flange and flash temperature – and variation of power index (s) during a 4.25-sec braking for drum in Fig. 9

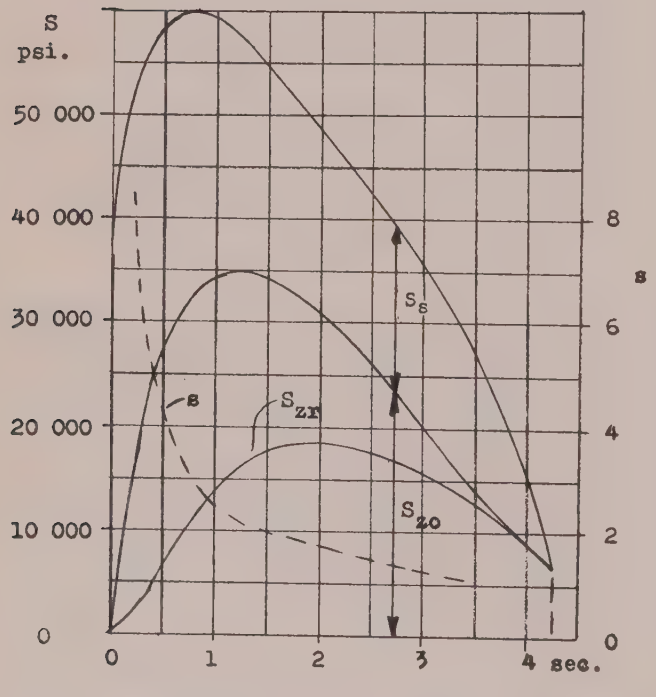


Fig. 31 – Surface stresses due to temperature gradients and flash temperature with tensile stresses on outer periphery and variation of power index (s) during a 4.25-sec braking, for the drum in Fig. 8

the drum is subject to severe compressive hoop stresses, as if they had been generated by external pressure. Again it is well known that such external pressure can easily bring about buckling of a cylindrical shell. Once the surface has buckled, even if slightly, at local high spots the pressure and heat input can be very great. Assuming for example that the height of a small high spot is 0.001 in., with a ¼-in. thick lining that has an elastic modulus equal to 50,000 psi, roughly, the local pressure increase is 0.001 x 50,000/¼ = 200 psi! This is about twice as high as the maximum nominal pressure conventionally permitted on shoe brakes. Hence the local heat input will be also about twice as great.

To explain matters in more detail, we have to refer to Fig. 24 which shows that only about two-thirds of the width of the drum, the part adjacent to the flange, is throughout in compression. Very roughly, we have thus a short cylindrical shell of width equal to $\frac{5}{8}L$, with both ends held circular, and subject to an external pressure $\approx \frac{1}{2}S'_h r/a$. Following the treatment of von Mises[23] it would then appear that for such very short cylinders the effect of L is negligible and the cylinder tends to buckle in a number of lobes, given by:

$$\sqrt{1 + 2(1 - m^2)a/r} \qquad (105)$$

These lobes, equally spaced around the periphery, would arise beyond a certain critical pressure $\sim E_r r/a$, E_r denoting the reduced modulus of elasticity. Owing to the complicated state of stress the value of E_r is rather difficult to determine here. Qualitatively, however, we may conclude that since E_r is lowered by increasing stresses and temperature, the effect of r is bound to be less than one would infer from the above.

Whereas there is no direct evidence available yet to support the above hypothesis, existing experience is not inconsistent with it. In particular the number of lobes given by the above expression is of the right order; perhaps because theoretically at least, it is independent of E_r. It has to be emphasized, however, that although a rigid flange seems to be a potent bias, whether heat spots proper actually develop or not depends also on (1) the tendency of the brake shoe to pick-up, (2) on its pressure pattern, (3) on the fade properties, and (4) hardness of the brake lining, plus a few more secondary factors.

Part III – Mechanical Stresses

The inference one could draw from the previous stress analysis might be misleading without taking the mechanical stresses into account. For this rea-

[23] See *Zeitschrift des Vereines Deutscher Ingenieure*, Vol. 58, May 9, 1914, pp. 750-755: "Der kritische Aussendruck zylindrischer Röhre," by R. von Mises.

son a brief treatment of some simpler cases will be developed below.

With many aircraft drums the full track surface is covered by a number of small pads. The customary radial pressure is in the order of 150 psi, and if the pads follow reasonably the drum's contour, both hoop and shear stresses are very small if compared with heat stresses. Similarly, the effect of the centrifugal force does not induce more than about 1000 psi hoop stress.

The same applies to motor car shoe brakes as well, but as the shoes usually cover only about 220 deg of the track, in addition to and superposed upon other stresses, there is a considerable amount of bending. In what follows only the simple case will be considered when the shoes are the same, are symmetrically disposed, and equally loaded, as is the case with motor car brakes of the 2-leading-shoe or fixed cam type. It will be further supposed that the cross-section of the drum is sufficiently uniform and symmetrical to be considered as a ring for the present purpose. This is equivalent to assuming the absence of bellmouthing and end-effects, as it should be for well-designed drums.

Concerning the load on the drum it will be supposed that the shoes exert uniform radial pressure. This would again correspond to an ideal though not impossible case. Stresses, deflection, and the effect of the number of shoes will be discussed on these premises.

III. 1. Mechanical Stresses

Mechanical Stresses – Let it be supposed that a ring of radius a width L, with its cross-section having a moment of inertia (I) is loaded by two small forces Δq, as shown in Fig. 33. If the thickness of the ring be small compared with its radius, the bending moment $\Delta M'$ in the quadrants 012, and 014 is according to standard texts:

$$\Delta M' = \tfrac{1}{2}\Delta q a \left(\frac{2}{\pi} - \cos\gamma\right) \qquad (106a)$$

But if 01 is to be used as the axis of reference for γ, in the quadrants 023, and 034 the bending moment is:

$$\Delta M'' = \tfrac{1}{2}\Delta q a \left(\frac{2}{\pi} + \cos\gamma\right) \qquad (106b)$$

Apart from the bending moment one has to consider too the effect of a tensile hoop stress $(\Delta S_t')$ given by:[24]

$$\Delta S_t' = \Delta q/\pi F' \qquad (107)$$

where F' = area of cross-section. The effect of shear stresses is negligible.

If now the brake has a pair of diametrically opposed shoes, of equal arcuate length (U), and

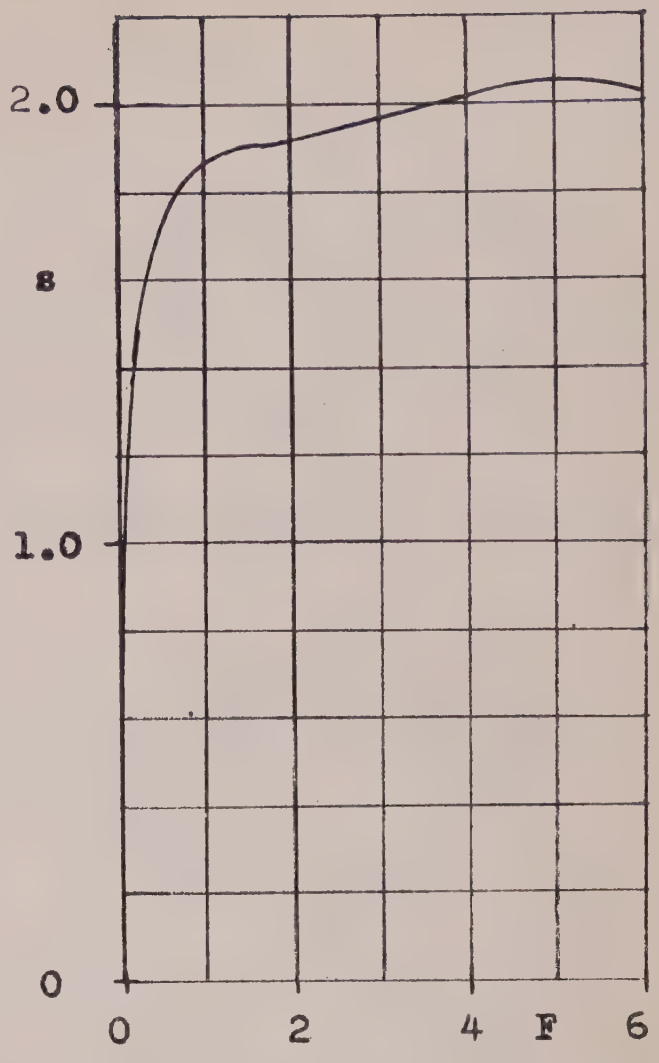


Fig. 32 – Effect of Fourier's number on power index (s) for the incidence of maximum temperature difference, $t = t_d$

having the same macroscopically uniform pressure pattern, as in Fig. 34, we can write:

$$\Delta q = pLad\phi \qquad (108)$$

The bending moment and the hoop stress can be thus obtained by integrating the previous equations with respect to φ. It follows then that within the region: $|\gamma| < \tfrac{1}{2}(\pi - U)$

$$M' = \tfrac{1}{2}pLa^2 \int_{-\frac{1}{2}U}^{+\frac{1}{2}U} (\tfrac{1}{2}\pi - \cos\overline{\gamma - \varphi})\, d\varphi$$

$$= pLa^2\left(\frac{U}{\pi} - \cos\gamma \sin\frac{U}{2}\right) \qquad (109)$$

For the region $|\nu| > \tfrac{1}{2}(\pi - U)$ we have similarly:

$$M'' = \int_{-\frac{\pi}{2}+\gamma}^{+\frac{1}{2}U} \Delta M' + \int_{-\frac{1}{2}U}^{\frac{\pi}{2}+\gamma} M''$$

$$= pLa^2\left(\frac{U}{\pi} - 1 + \sin\gamma\cos\frac{U}{2}\right) \qquad (110)$$

[24] See p. 44 of "Berechnung der Maschinenelemente," by M. ten Bosch. Springer-Verlag, Berlin, first edition, 1929.

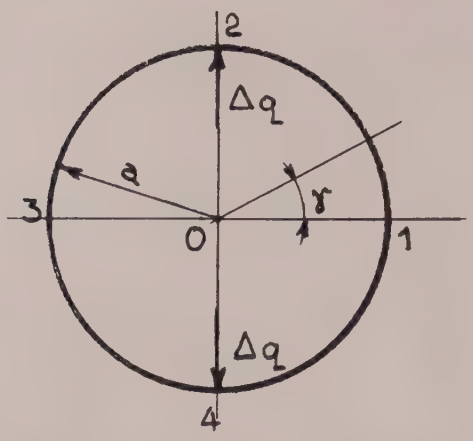


Fig. 33 – Thin ring subject to bending by diametrically opposite forces

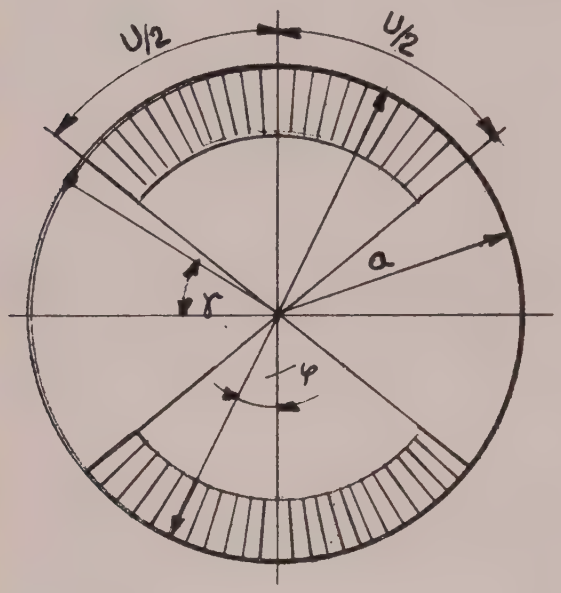


Fig. 34 – Assumed pressure distribution in a brake drum

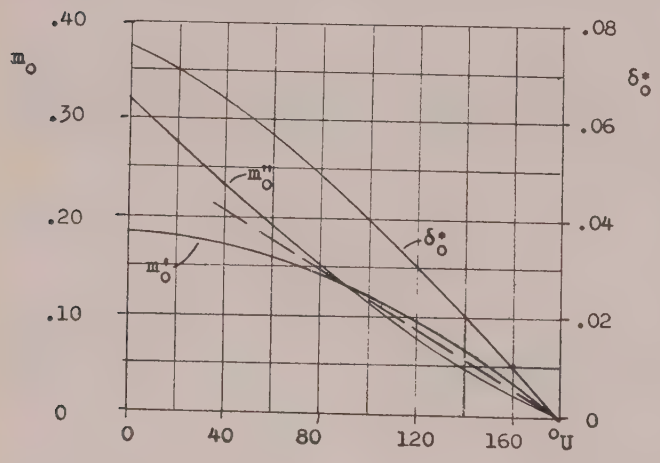


Fig. 35 – Effect of length of lining arc on drum deflection and bending stresses

To obtain the effect of U when a certain torque T_b has to be developed, the above can be transformed with:

$$T_b = 2\mu pLa^2U \tag{111}$$

by eliminating p, so that:

for $\gamma = 0$

$$M'_o = \frac{T_b}{2u}\left(\frac{1}{\pi} - \frac{\sin \frac{1}{2}U}{U}\right) \tag{112}$$

and for: $\gamma = \frac{1}{2}\pi$

$$M''_o = \frac{T_b}{2u}\left(\frac{1}{\pi} - \frac{1 - \cos \frac{1}{2}U}{U}\right) \tag{113}$$

The expressions in the brackets, denoted by m'_o and m''_o, are plotted in Fig. 35, showing that for $U > \frac{1}{2}\pi$ numerically the two are about the same, and may be well approximated by:

$$m'_o \approx m''_o \approx 0.265 - 0.0842\,U \tag{114}$$

To obtain the resultant stresses, we have to superpose the hoop stresses:

$$S'_t = \int_{-\frac{1}{2}U}^{+\frac{1}{2}U} \frac{\Delta}{} S'_t = \frac{pLaU}{\pi F'} = \frac{T_b}{2\mu auF'} \tag{115}$$

upon the bending stresses.

To show the order of magnitude, let it be supposed that a 2230-lb vehicle with a 60/40% front/rear braking ratio, is to be braked with 65% g deceleration. If drum diameter is approximately one-half wheel diameter on the front brakes, drum drag = 2230 x 0.30 x 2 x 0.65 = 870 lb. That is, for example, with 9 x 1¼-in. brakes:

$$T_b = 4.5 \times 870 = 3920 \text{ lb-in.}$$

With $r = \frac{1}{4}$ in., $L = 1\frac{1}{2}$ in., we have then $I = 0.00196$ in.^{4}, and for $U = 110$ deg, $m'_o = 0.10$, so that if $\mu = 0.35$, the maximum bending stress is:

$$\frac{3920 \times 0.10 \times 1/8}{0.70 \times 0.00196} = 35{,}700 \text{ psi}$$

whereas the corresponding hoop stress is only:

$$\frac{3920}{0.70 \times \pi \times 4.5 \times 1.5 \times \frac{1}{4}} = 3330 \text{ psi}$$

and may be thus neglected. The bending stresses are, however, very serious, and unfortunately, they reinforce the heat stresses at the center of the shoe.

The deflection can be now obtained by Castigliano's method from the strain energy (V). If we neglect shear and hoop stresses, the strain energy stored in the four quadrants is:

$$V = \frac{2a}{EI}\int_o^{\pi/2} M^2 d\gamma \tag{116}$$

If a fictitious force (P^*) is assumed to act at $\varphi = 0$, the deflection there due to bending alone is:

$$2\delta^* = \frac{2a}{EI}\left(\int_o^{\pi/2} 2M\frac{\partial M}{\partial P^*}\,d\gamma\right)_{P^* = 0} \tag{117}$$

Referring now to equations 106, 109, and 110, the

bending moments in the region $0 < \gamma < \frac{1}{2}(\pi - U)$ are:

$$M' + \tfrac{1}{2}P^*a\left(\frac{2}{\pi} - \cos\gamma\right) \tag{118}$$

and in the region $\frac{1}{2}(\pi - U) < \gamma < \frac{1}{2}\pi$

$$M'' + \tfrac{1}{2}P^*\left(\frac{2}{\pi} - \cos\gamma\right) \tag{119}$$

In both cases:

$$\frac{\partial M}{\partial P^*} = \tfrac{1}{2}a\left(\frac{2}{\pi} - \cos\gamma\right) \tag{120}$$

and since $P^* = 0$, we have for the radial deflection at the center of the shoes:

$$\delta^* = \frac{pa^4L}{EI}\left(\frac{\pi - U}{\pi} + \frac{\pi - U}{4}\sin\tfrac{1}{2}U - \cos\tfrac{1}{2}U\right) \tag{121}$$

To obtain directly the effect of U with other factors remaining the same, p may be again eliminated with the aid of equation 111, resulting in:

$$\delta^* = \frac{T_b a^2}{2\mu EI}\zeta\left(\frac{1}{U} - \frac{1}{\pi} + \frac{\pi - U}{4U}\sin\tfrac{1}{2}U - \frac{\cos\frac{1}{2}U}{U}\right) \tag{122}$$

The expression in the brackets, denoted by $\delta^*_\bullet$, is also plotted in Fig. 35, and shows an almost linear dependence on U. To obtain the total shoe-center-lift one has to add, of course, the expansion due to the hoop stresses, which is:

$$\delta^*_t = \frac{T_b}{2\pi\mu F'E} \tag{123}$$

and is independent of drum radius and lining arc.

Strictly, these expressions are valid for narrow rings only, and to allow for L we have to use $E/(1 - m^2)$ instead of E. The difference is, however, only in the order of 5%, for cast iron.

To show the order of magnitude, with the example given previously, we have with $E/(1 - m^2) = 17 \times 10^6$ psi:

$$\delta^* = 0.120 \text{ in.}$$

and:

$$\delta^*_t = 0.00028 \text{ in.}$$

that is, the effect of hoop stresses on deflection can be safely ignored. On the other hand these figures show that it is absolutely essential to reinforce thin drums with some ribs or fins if conventional shoe brakes are used.

It would thus seem that if the thickness (r), and the track surface $A = \pi aL$ are to remain constant:

$$\text{bending stress} \propto a,$$

and:

$$\text{deflection} \propto a^3$$

Small diameter wide drums are thus preferable on this score as well.

III. 2. Effect of Number of Shoes

Effect of Number of Shoes – If a ring be loaded by n equal, and equiangularly spaced radial loads P, it can be shown[25] that the bending moment M_n under each load is:

$$M_n = \tfrac{1}{2}Pa\left(\frac{n}{\pi} - \cot\frac{\pi}{n}\right) \tag{124}$$

the corresponding deflection being:

$$\delta_n = \frac{Pa}{4EI}\left(\frac{\pi}{n}\operatorname{cosec}^2\frac{\pi}{n} + \cot\frac{\pi}{n} - \frac{n}{\pi}\right) \tag{125}$$

For a certain torque the load $nP = P_o$ has to be constant so that on eliminating P from the above expressions, and on taking $n = 2$ (that is, a 2-shoe brake) as our yardstick, we have for $n > 2$ as figures of merit:

$$M_m \propto 1 - \frac{\pi}{n}\cot\pi/n \tag{126}$$

$$\delta_n \propto \left[\frac{\pi^2}{n^2} - 2 + \left(\frac{\pi}{n}\cot\pi/n\right)^2 + \frac{\pi}{n}\cot\pi/n\right]\left[\tfrac{1}{4}\pi^2 - 2\right]^{-1} \tag{127}$$

These expressions are plotted in Fig. 36 showing that with a 4-shoe brake the bending stresses and ovality become insignificant compared with a 2-shoe brake. Whereas such stress reduction is obviously very desirable *per se,* an excessively rigid drum may well save pedal travel, but in other directions it may not be always advantageous.

To be more explicit, let us first consider the hypothetical case of an infinitely rigid drum being matched against equally rigid shoes, fitted with very hard linings. Assuming that the shoes fit perfectly when *cold,* as the drum temperature increases and the drum expands there will be a widening gap between the radii, as sketched in Fig. 37. (For brake linings are excellent heat insulators so that the shoe temperature, and curvature, tend to lag behind those of the drum.) Usually, this would cause (1) a serious drop in the shoe-factor, (2) intense local heat concentration with all its adverse effects, and (3) a hollow wear on the shoe.

Let it be supposed now that the drum and shoes cool down. On account of the hollow wear generated previously, on applying the brakes again, initially the shoe would contact the drum at its heel and toe only, as sketched in Fig. 38, and this is liable to cause fierce action, known as grab. A serious fluctuation of torque may thus ensue, even if the lining friction remained absolutely constant! It may also partly explain the notoriously erratic behavior of brakes on rapid cycle tests.

Again, if the drum is rigid but the shoe is flexible, or vice versa, the length of lining arc in contact increases with the load on the shoe, resulting in nonprogressive action. Badly matched shoes and drums account, therefore, for many of the well-known inconsistencies in the behavior of brakes.

It would thus follow that a certain amount of drum flexibility is desirable, so that for very heavy-duty shoe brakes the 3-shoe-type[13] is sometimes about the best compromise. The drum can have a

[25] See p. 154 of "Formulas for Stress and Strain," by R. J. Roark. McGraw-Hill Book Co., Inc., second edition, 1943.

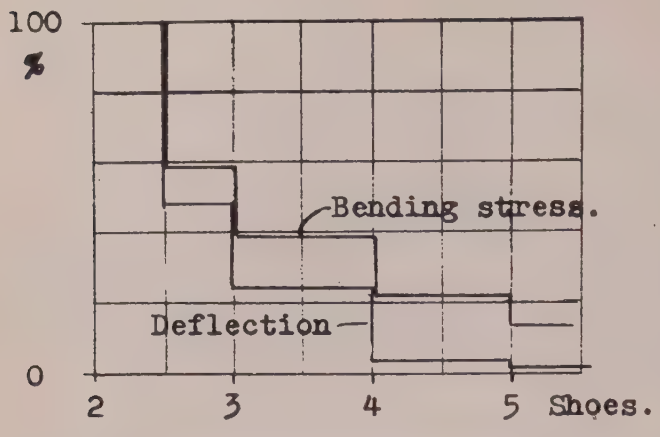


Fig. 36 – Effect of number of shoes on bending moment and deflection for short lining arcs

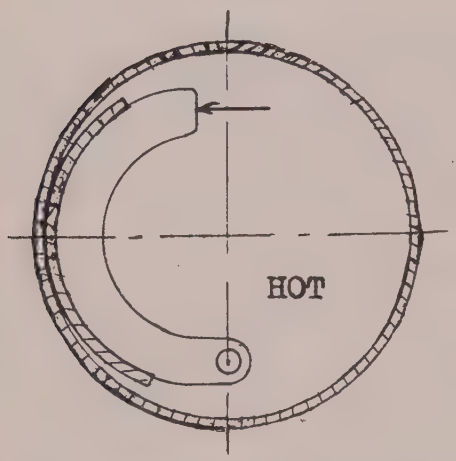


Fig. 37 – Effect of temperature difference between drum and shoe, giving rise to hollow wear and change of shoe factor

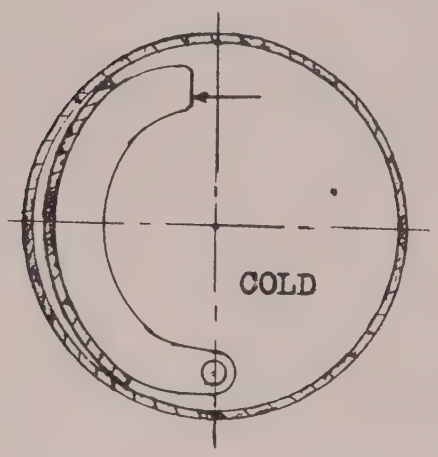


Fig. 38 – Initial contact between cold drum and cold shoe with hollow wear

flexible flange so that it is not too prone to heat checking; the section can be reasonably thin and uniform so that the danger of crazing or heavy scoring is not too great; and the controlled amount of flexibility obviates the need for excessive machining limits.

Part IV – General Design Aspects

In the first instance a brake drum has to have an adequate track surface area (A) and thickness (r) to keep the surface temperature below a level that would impair the drum material, and cause excessive lining wear or fade. With motor vehicle brakes there is also the additional necessity to provide adequate cooling surface ($\bar{A}$) which may call for a number of fins.

The second requirement is that the stresses should be safe, and the third one is that the track surface should remain straight whether hot or cold. The correct amount of flexibility depends also on shoe rigidity, and is best determined experimentally.

These points will be discussed below under the main headings of temperature rating, stress rating, and mechanical design, respectively.

IV. 1. Temperature Rating

Temperature Rating – The physical concept behind temperature rating is that excessive surface temperature alone can bring about (1) deleterious chemical changes in the drum material, (2) impair its abrasive resistance, (3) and/or cause excessive rate of lining wear and fade.

In fixing a limit to the maximum surface temperature, it must be remembered that even under a light load a certain proportion of the d spots is liable to reach the melting point. This is quite unavoidable since very hard foreign matter such as dust, metal particles, and so forth, cannot be excluded altogether. What matters is, however, that the number of such very hot spots should be negligible compared with the rest, and that the average should be well on the safe side. With cast iron the critical temperature for (1) above is about 700 C so that, if the drum surface is protected, for example, by heavy chromium plating, as is customary on aircraft drums, the safe limit is $T_o + T' + T_m < 700$ C.

If the surface is unprotected, as on motor car drums, the safe limit is about $T_o + T' + T_m < 500$ C, which leaves about 200 C margin for T_c.

From the thermodynamic point of view the least amount of material would be required with very thin drums, when $\theta \approx 1$. But that would lead to very large, and consequently too heavy brakes. For a minimum aggregate weight, cost, and size, a compromise is therefore necessary. In determining the appropriate amount of r two cases arise, depending on whether T' is significant or negligible, as typified by motor vehicle or aircraft brakes respectively.

Case 1: $T' \to 0$ – To find the economical thickness of aircraft drums or the like, the effective heat capacity $C_e = rcw\theta$ is plotted in Fig. 39 against weight (wr) per unit track surface area, for $t^* = 15$ sec. The broken lines represent the heat capacity of an infinitely thick section, corresponding to $C_e = (cwkt^*/N)^{1/2}$.

For the present purpose Na has been treated again as a solid, without taking into account its latent heat of fusion (L_o). If $L_o/c = 91.5$ C be deducted from H by way of a correction, as suggested in connection with Table 3, it is equivalent to assuming that of the total thickness r, only $r\theta$ would actually melt. For then:

$$(H - rwL_o\theta)/rcw\theta = T_m - L_o/c \qquad (128)$$

Since in all cases the C_e curves are geometrically similar, it may be concluded that quite generally:

(a) The C_e curves being almost linear up to about $F = 0.80$ – corresponding to $\theta = 0.89$, and $\phi = 0.85$ – the economical thickness seems to be:

$$r_e = \sqrt{0.80\, at^*} \qquad (129)$$

(b) Assuming the same T_m, the most suitable material for weight economy depends on H, since $C_e = T_m/H$. This shows that a moderately loaded Cu drum runs hotter than a cast iron one of the same weight!

(c) In interpreting Fig. 39 the difference in the maximum permissible T_m as between various materials must be taken into account. For this reason the weight advantage of aluminum against cast iron is often small.

Case 2: $T' \neq 0$ – When T' is significant – as is the case with motor vehicle brake drums – it was shown in the section on effect of cooling on mean drum temperature that the heat capacity of the drum Wc has an adverse influence on T'_∞. On the other hand, as r is increased T_m tends to diminish, so that for a certain drum thickness $T'_\infty + T_m$ tends towards a minimum.

To investigate this condition let us take the simple case when the drum has a uniform thickness throughout, and the ratio of track surface A to the total exposed surface $\bar{A}$ is defined by $\sigma = A/\bar{A}$. It follows then from equations 7 and 14 that:

$$T_m + T'_\infty - T_e = H\left(\frac{\sigma}{ht_e} - \frac{\sigma}{2rcw} + \frac{0.855}{rcw} + 0.353\,\frac{r}{kt^*}\right) \qquad (130)$$

and this has a minimum when:

$$F = 2.425 - \sigma/0.71 \qquad (132)$$

independently of the extent of cooling.

To show the order of magnitude, let it be supposed that a 2230 lb vehicle is to be stopped in $t^* = 4$ sec, at $t_o = 2$ min intervals from 60 mph (65%g braking). If the front/rear ratio is 60/40%, with 9 x 1¼-in. brakes, the specific heat input is at the front $H = 3$ Btu/in.² With moderately cooled, modern brakes $h \sim 0.003$ Btu/ft² x deg F x sec, so

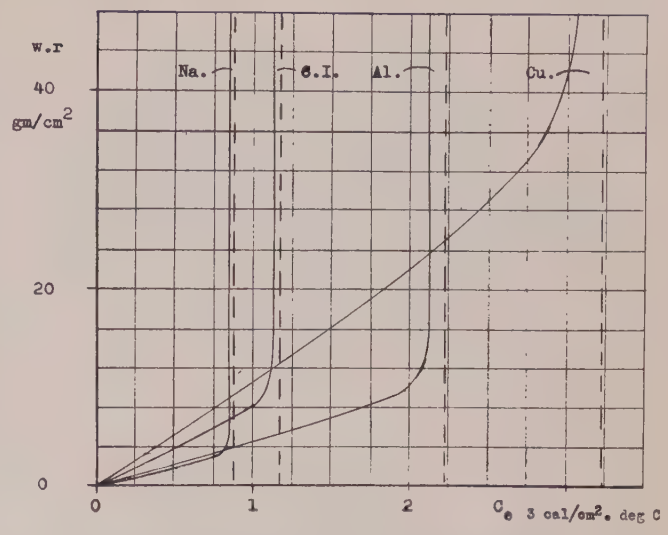


Fig. 39 – Dynamic heat storing capacity versus weight of different metals for $t^* = 15$ sec

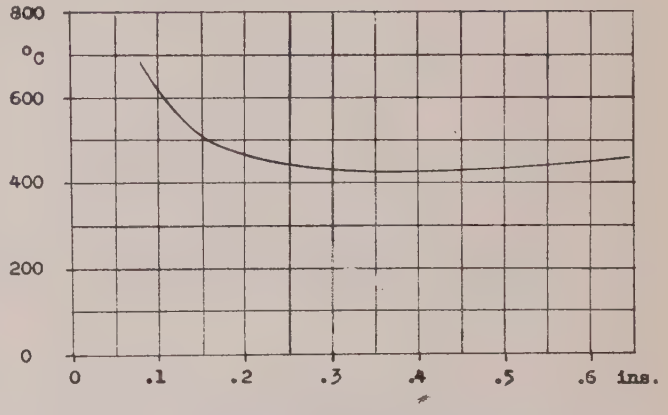


Fig. 40 – Effect of drum thickness on maximum bulk temperature, for t^* = sec, $H = 3$ Btu/in.², $t_o = 2$ min, $h = 0.003$ Btu/ft² deg F sec, and $6 = 0.372$

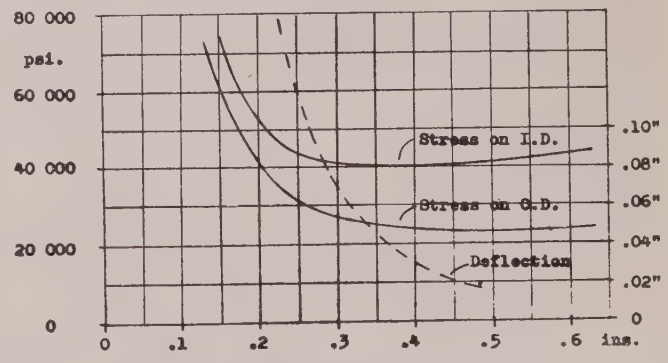


Fig. 41 – Effect of drum thickness on maximum normal stresses and on radial deflection for case treated in Fig. 40

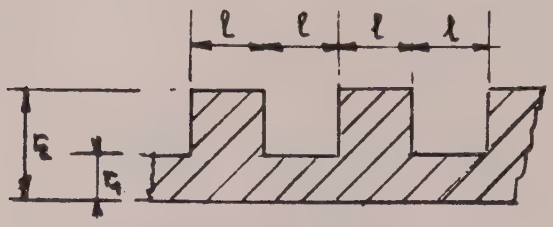


Fig. 42 – Drum with square reinforcing fins

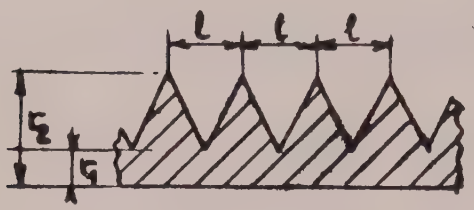


Fig. 43 – Drum with triangular reinforcing fins

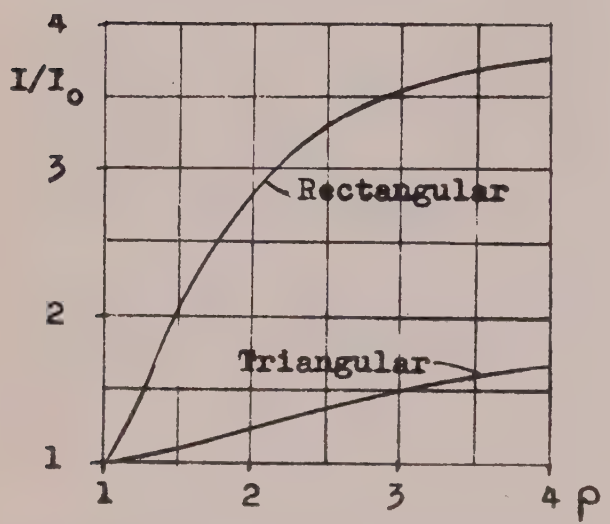


Fig. 44 – Effect of fin depth on moment of inertia for constant cross-sectional area

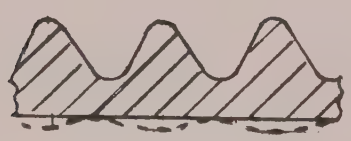


Fig. 45 – Macroscopic surface deformation of finned drum, due to heat

that for a cast iron drum with geometric proportions to $\sigma = 0.372$ we have T in deg C, and r in in.:

$$T_m + T'_\infty - T_e = 247.5 + \frac{33.7}{r} + 240\,r$$

The plot of this typical case in Fig. 40 indicates a reasonably wide favorable range of r, its choice being thus dictated by the stresses generated.

Three more important conclusions follow as well:

1. The widespread, and somewhat indiscriminate use of the hp/in.^{2} rating does not apply here; in fact it does not seem possible to use any such simple rule of thumb.

2. The heat capacity of the flange has a far from negligible effect; this is understandable on considering that with cast iron drums the flange can easily account for 50% of the total weight.

3. The manner of mounting the drum on the wheel is of great importance. For if the contact between the two is intimate, and extends over a considerable surface, the heat capacity and cooling surface of the wheel can be drawn upon to suppress the residual temperature T'_∞, provided that no similar service is imposed upon the tires or upon the grease in the hub bearing.

IV. 2. Stress Rating

Stress Rating – As mentioned before, heat stresses are usually much more severe than those of mechanical origin, so that the drum material should be selected accordingly. On denoting the ultimate strength by S_u, the desired data are then from equations 98 and 103. For the bulk the value of the following should be large:

$$\frac{S_u(1-m)}{Eu}\left(cwk^3\right)^{1/4}$$

For the plated surface the value of the following should be large:

$$\frac{S_u(1-m)}{Eu}\sqrt{kcw}$$

Both criteria are consistent with the observation[5] that for good wear resistance the

$$\text{modell} = \frac{\text{Brinell number}}{\text{elastic modulus}}$$

should be high. This concept is extended and modified here by including the thermal properties, and Poisson's number of the material. As will be seen, the modification cannot be regarded as universally applicable, but depends upon the duty the material is to perform. For the surface (plating) k is less important than for the bulk, and so on. A further point worth noting is that for most metals the ratio Su/E does not vary a great deal with temperature, as mentioned before with reference to Fig. 27, and approximately the same is true about kcw. It follows that one may judge various materials by their properties at room temperature.

It has been also shown in the section on mechanical stresses that bulk stresses increase with $\sqrt{r}$,

whereas stresses due to flange effects $\propto 1/r$, and those caused by bending $\propto 1/r$.[2] In any given case, therefore, a decided optimum condition exists. To show its order of magnitude let us consider the examples treated following equations 115 and 132. If it be assumed that the flange is sufficiently flexible to keep the drum straight, for a uniform thickness the stresses may be computed from equations 98, 99, and 113, and are plotted in Fig. 41 both for the ID and for the OD, the broken line showing the shoe-center-lift.

As may be seen from Figs. 40 and 41, the effect of r on temperature and stresses is rather similar in character: in both cases there is an optimum, it is not pronounced, and the two cases practically coincide. The choice of r is thus dictated by the shoe-center-lift. Expressing r in in. and the stresses in psi the curves in Fig. 41 were plotted from:

for ID $\quad S_{z\theta} + S_b = 52{,}000\sqrt{r} + 1140/r^2$

for OD $\quad S_{zr} + S_b = 26{,}000\sqrt{r} + 1140/r^2$

so that around the optimum condition:

$$\frac{\text{heat}}{\text{bending}} \text{ stresses } = \begin{cases} 2.5 \text{ on ID} \\ 1.3 \text{ on OD} \end{cases}$$

This shows that on motor vehicle brakes mechanical stresses are not negligible, and with large diameter heavy-duty brakes they may assume a dominant role. The worst condition may then arise at $t = t^*$, when the brakes are hottest. This shows another limitation of the hp/in.^{2} method of rating; without many reservations it seems to be applicable to aircraft drums only, when $U = 360$ deg.

In assessing the maximum permissible stress level, it will be evident that some permanent set is unavoidable and drum life is limited if the brake size is not to be excessive. It follows, therefore, that on cooling the above stresses reverse. In the absence of reliable and relevant fatigue data one may thus merely postulate an extreme criterion as:

$S_{zr} + S_b <$ ultimate tensile strength (for fracture)

$S_c + S_{z\theta} + S_b <$ ultimate compressive strength (for heavy scoring)

$S_c + S_{z\theta} + S_b <$ yield + ultimate tensile strength (for heavy crazing)

IV. 3. Mechanical Design

Mechanical Design – Apart from various features, there are three fundamental points in drum design. They are: (1) the geometrical shape as characterized by L/D; (2) the shape of the drum's section; and (3) that of the flange.

Geometrical proportion: When the track surface area ($A = \pi DL$) and the thickness (r) are fixed, it follows from the foregoing that, other things being the same:

1. Flash temperature $\propto D^{3/4}$
2. Bending stress $\propto D$
3. Deflection through bending $\propto D^3$
4. Flange weight, approximately $\propto D^2$

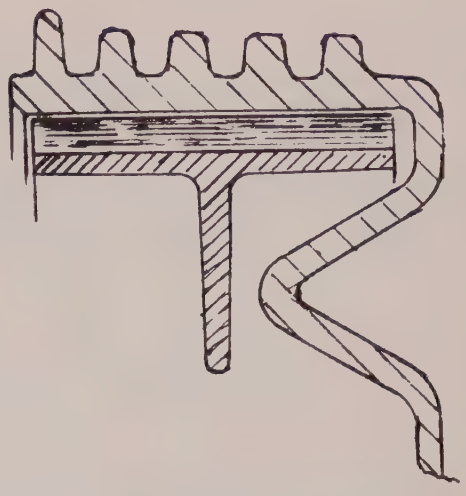


Fig. 46 – 10 x 2-in. cast-iron (passenger car) drum with flexible flange

5. Radial heat expansion $\propto D$
6. Total brake weight, approximately $\propto D^{3/4}$

As a result small-diameter wide brakes are favored. Against these militate:

1. Pedal load $\propto 1/D$.
2. Abrasive pressure $\propto 1/D$.
3. Difficulty to ensure uniform pressure across a wide shoe.
4. Consequent deterioration of contact efficiency.
5. Contortions of the drum when hot.
6. Possible restrictions of available space.

Fig. 47 – Weakening of flange of drum shown in Fig. 13

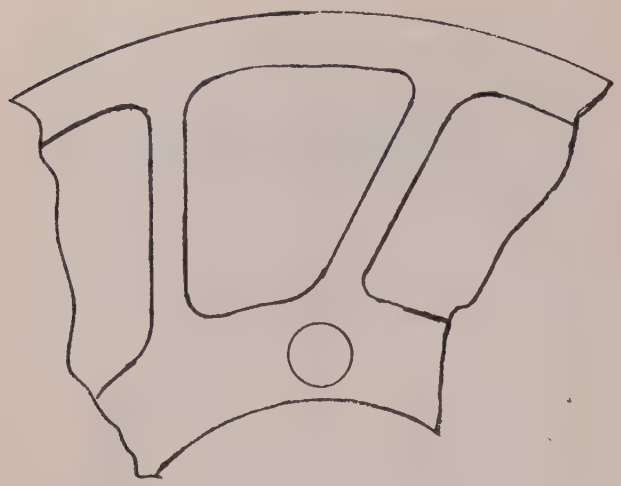


Fig. 48 – Weakening of flange to reduce bellmouthing

The loss of pedal load is usually more than made up through saving of lost motions, so that ultimately, available room and shoe rigidity seem the decisive factors limiting L. Depending on the type of brake, we have thus:

$$0.175 < L/D < 0.80$$

Shape of drum section: Referring to Fig. 24, it has been shown in the section on stresses due to flange-effects that the constraint of a stiff annular flange against drum deflection is of local character only. Particularly with large diameter heavy-duty shoe brakes it is necessary to stiffen the drum with some peripheral ribs.

In a first approximation the section can be then either as shown in Fig. 42 or in Fig. 43, in both cases the geometrical proportions being characterized by $(\rho = r_2/r_1)$. Let us suppose now that the cross-sectional area $\frac{1}{2}Lr_1(1 + \rho)$ is to remain constant, and that the moment of inertia I_o of a uniform section $(\rho = 1)$ is to serve as a basis of comparison. It can be then shown that for other values of ρ the moment of inertia (I) is:

with rectangular ribs:

$$\frac{I}{I_o} = 4\,\frac{1 + \rho^3}{(1 + \rho^3)} + 12\,\frac{\rho(\rho - 1)^2}{(1 + \rho)^4} \qquad (133)$$

with triangular ribs:

$$\frac{I}{I_o} = \frac{8}{3}\,\frac{1 + \rho^2}{(1 + \rho)^2} - \frac{16}{3}\,\frac{\rho^2}{(1 + \rho)^4} \qquad (134)$$

Both equations are plotted in Fig. 44, showing how ineffective triangular ribs are, and that with rectangular ones it does not pay to go beyond $\rho \approx 2$. The additional stiffening is small, and heat stresses are liable to increase rapidly, beyond the $\sqrt{r}$ law. The reason is because a ribbed section tends to deflect as sketched in Fig. 45, resulting in a local increase of pressure and therefore of heat input under the ribs. Whereas this tends to iron out axial variations of the mean temperature, it can also easily overshoot the target. With many drums heat spots are concentrated under the ribs when they are too stiff and/or too widely spaced.

Thermodynamically, a uniform section may thus appear ideal. It has to be considered though that ribbing increases the cooling surface, breaks up the bellmouthing tendency, and with some brakes the shoes would get out of control unless they can lean against a stiff drum.

Flange design: The fundamental requirements for the flange are: (1) great torsional rigidity to transmit the torque smoothly, (2) great radial flexibility to avoid bellmouthing, (3) small heat capacity for the same reason, and (4) great lateral stiffness to shift the critical speed well outside service conditions. These requirements are difficult to reconcile; perhaps the simplest compromise is a corrugated flange along the lines shown in Fig. 46 (patent applied for) which the writer designed for passenger vehicles.

It may be of interest to note that when the flange of the drum shown in Fig. 13 was modified as shown in Fig. 47, lining wear was reduced by some 15%, owing to the more uniform spread of pressure. The penalty was a critical speed around 800 rpm. The peculiar shape of the arms developed as the outcome of some prior tests when the flange had been weakened as sketched in Fig. 48. Apparently this gave rise to a kind of toggle action and resulted in a violent squeak.

Apart from these mechanical aspects, the thermodynamic effect of the flange is of considerable importance with passenger vehicle brakes, where the flange weight can be as high as 50% of the total. The flange acts then as a heat reservoir and impedes cooling in normal usage, as may be seen from equations 7 and 130. On the other hand, with poorly cooled brakes a heavy flange is a useful thermal cushion to absorb a short burst of heavy braking, and without increasing unduly the bulk stresses through excessive r. The demand for heat capacity and flexibility can be then reconciled by backing a corrugated flange with aluminum or sodium.

A flexible flange is, however, no universal panacea, as it can be used *only* if there is no appreciable offsided load on the drum. It is thus suitable for most, but not all, currently employed brakes.

Acknowledgment

In concluding, the writer wishes to acknowledge his indebtedness to the directors of Girling Ltd., Tyseley, Birmingham, England, for facilities to carry out some of the experimental work embodied in this paper.

Engineering Possibilities of Synthetic Lubricants

W. A. Zisman Naval Research Laboratory

This paper was presented at the SAE National Fuels and Lubricants Meeting, Tulsa, Nov. 7, 1952.

THIS paper is concerned with a special class of synthetic oils which for lack of a better name are often identified as tailor-made oils. These are tailor-made in order to obtain certain optimum properties, in the way that synthetic fibers are synthesized to obtain specific qualities. Just as synthetic fibers are not intended to entirely replace cotton or wool, so tailor-made oils are not intended to replace petroleum oils; instead, they are designed for special applications where petroleum products do not function well enough.

The synthetic liquids which have received the greatest attention in the field of lubrication in the past 10 years are the aliphatic esters, the polyalkylene oxides, the silicones, the esters of phosphoric and silicic acids, and the highly fluorine-substituted hydrocarbons and ethers. It is interesting, and of much theoretical importance, that the most promising of these new liquids are free from molecular association and are essentially made up of flexible linear chains of atoms. Much progress has been made in relating the molecular structure of such liquids to the viscosity, viscosity-temperature coefficient, volatility, freezing and boiling points, and boundary lubricating properties. A good basis now exists for predicting the properties

TAILOR-MADE oils, together with their engineering possibilities, are discussed in this survey of synthetic liquids developed during the past 10 years. Within that time, tailor-made oils have been especially designed to cover a wide range of applications where petroleum products do not function well enough.

Since these oils are homogeneous in the chemical sense, all of their properties are readily controlled and predicted. The author points out that chemists are using these inherent advantages to synthesize oils which provide necessary antiwear, antiseize, and corrosion-inhibiting properties. Various chemical additives take care of nearly all extreme conditions of lubrication.

The Author

W. A. ZISMAN is head of the Surface Chemistry Branch at the Naval Research Laboratory, Washington, D. C. He obtained his Bachelors and Masters degrees in physics in 1927 and 1928, respectively. His Ph.D. was obtained from Harvard in 1932. Since 1939 he has specialized in research on surface chemistry, corrosion prevention, and lubrication.

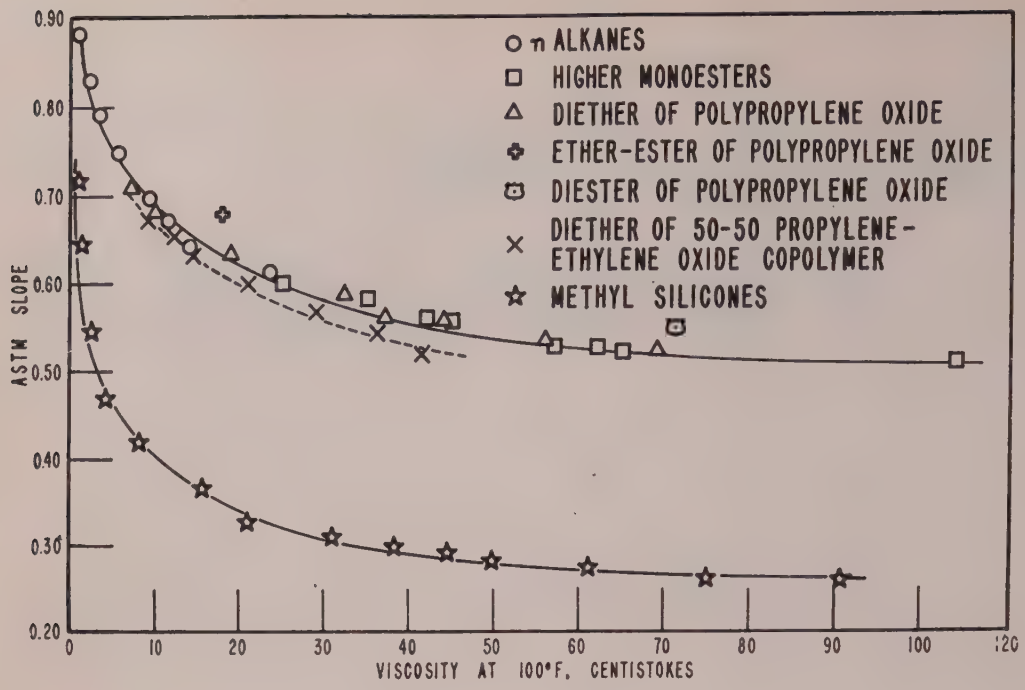


Fig. 1 – Comparison of ASTM slope versus viscosity for analogous liquids

of the many interesting chemical variations of these and analogous materials.

These tailor-made oils have certain inherent advantages over the mineral oils produced by modern methods. As they are homogeneous in the chemical sense, all of their properties are readily predicted and controlled. Familiar types of chemical addition agents can be used to improve them; and the development of compounded oils is generally simpler, since there are no undesired chemicals present to cause variations from one production batch to another in the response to antioxidants, V. I. improvers, detergents, oiliness additives, and wear preventives.

When thick film or hydrodynamic conditions of lubrication prevail, the rheological and heat-transfer properties of the oil are dominant. When boundary conditions prevail, the significant properties are now known to be the physical properties of the surfaces, and also the nature of the surface reaction products formed with chemically reactive materials in the oil. There definitely is a contradiction in requiring that a synthetic oil with a low temperature coefficient of viscosity and a low freezing point should not need antiwear additives.[1, 2] Chemists are beginning to recognize that it is much more effective to synthesize oils with the desired bulk properties, and to use surface-active additives to provide the necessary antiwear, antiseize, and corrosion-inhibiting properties.[2] Therefore, in developing either synthetic or petroleum oils, various kinds of chemical additives are nearly always needed to care for the more extreme conditions of lubrication.

Comparative Properties

Graphs of viscosity against temperature on ASTM chart paper (D341-43) for the higher boiling liquids of linear molecular structure are usually straight lines over the temperature range 210 F to 0 F and often over even larger ranges.[3] The slope of each line (or the ASTM slope) is greater the larger the temperature coefficient of viscosity. The ASTM slope and the viscosity at 100 F when taken together give a useful description of the viscosity-temperature characteristics of each oil. In Fig. 1 is a graph of the ASTM slope against the viscosity at 100 F of a group of synthetic oils. The upper solid curve is for the n-alkanes (or paraffins). It has been found[2] that the ASTM slope for any pure hydrocarbon of a given 100 F viscosity is not less than that given here for the n-alkane of the same 100 F viscosity. This curve exemplifies the empirical rule that the lower the viscosity required in a homologous series of liquids, the greater is the temperature coefficient of viscosity. Open-chain dimethyl silicone liquids plot in Fig. 1 as the lowest curve, which emphasizes their uniquely low temperature coefficients. In striking contrast are the viscometric properties of the highly fluorinated hydrocarbons which have ASTM slopes of 0.9 or more, and plot in Fig. 1 too much above the other liquids discussed here to be shown on the chart. This is the most serious limitation to their use in

---

[1] See Annals of the New York Academy of Sciences, Vol. 53, June 27, 1951, pp. 836-861: "Fundamental Aspects of Lubrication," by J. E. Brophy and W. A. Zisman.

[2] See *Industrial and Engineering Chemistry*, Vol. 42, December, 1950, pp. 2415-2468: "Functional Selection of Synthetic Lubricants," by C. M. Murphy and W. A. Zisman.

[3] See ASME Transactions, Vol. 71, July, 1949, pp. 561-574: "Viscosities and Densities of Lubricating Fluids from −40 to 700 F," by C. M. Murphy, J. B. Romans, and W. A. Zisman.

lubrication technology.

High-boiling aliphatic monoesters containing up to 60 carbon atoms have been studied[2] because of their many physical resemblances to the paraffins. These plot in Fig. 1 on practically the same curve as the n-alkanes. The aliphatic diesters having the shortest side chains plot close to but above the n-alkane curve; the larger the side chain, the larger the distance above the n-alkane curve. In general, the higher boiling monoesters, diesters, triesters, and so forth, whether of carboxylic, phosphoric or silicic acids plot on or above the n-alkane curve. Esters containing rings always plot even further above the n-alkane curve. This explains why the aliphatic esters have been preferred over all other esters, for they have the smallest ASTM slopes for a given 100 F viscosity. They also offer a more versatile and cheaper chemical route for obtaining liquids having very low freezing points and nearly the lowest ASTM slopes obtainable in hydrocarbons.

Polyalkylene oxides, if they are not chain-stoppered, usually plot above the n-alkane curve in Fig. 1. When chain-stoppered at one end of the molecule by either an alkyl ester group or an alkyl ether group, the ASTM slope is lowered because of the decreased degree of association obtained by blocking one of the terminal hydroxyl groups. When these compounds are chain-stoppered at both ends to eliminate molecular association, the ASTM slope decreases further.[4] In Fig. 1 are plotted the properties of various types of polyisopropylene oxides and of the 50/50 copolymers of ethylene oxide and isopropylene oxide. Terminating both chain ends with alkyl ether groups lowers the ASTM slope more than with alkyl ester groups. The 50/50 copolymers have even lower ASTM slopes than the propylene oxide polymers of the same 100 F, and this is shown in Fig. 1 by the dashed curve. No doubt the polyethylene oxides, similarly terminated at both ends, will have the lowest slopes of all of the polyalkylene oxides, and hence they will plot even further below the n-alkane curve. However, these polymers are quite soluble in water and have high freezing points. It is obvious from Fig. 1 that none of the polyalkylene oxides have such low ASTM slopes as the open-chain dimethyl silicones of the same 100 F viscosity.

Since the n-alkanes have the smallest temperature coefficients of viscosity of the hydrocarbons of the same 100 F viscosity, it is logical to turn to similar aliphatic hydrocarbons as a source of improved tailor-made lubricants; but chemical difficulties intrude. The synthesis of hydrocarbons having the proper molecular structure to have freezing points below —40 F and such low ASTM slopes is much more expensive than the synthesis of either aliphatic esters with short branches or of double chain-stoppered polyalkylene oxides.

Lubricating oil volatility has become an important property in the past 15 years. This is due to the increasing use of oils of lower viscosity and the relation of the volatility to the rate of oil consumption in some uses and to the flammability in others. Thus in turbojet engines the rate of oil consumption rises with the oil volatility as do the flammability, the rate of lacquering, and the tendency for the oil to boil off the bearing after engine shutdown. Volatility in instrument oils decreases the maximum allowable time before relubrication; in aircraft control bearings it causes the greases to dry up; and in oils and greases for electric motors it decreases the upper temperature limit of operability and the service lifetime of the lubricant.[5]

Especially useful are oils having the combined properties of low volatility at 210 F, low viscosity at 100 F, and low pour points (or freezing points). In Tables 1, 2, and 3 will be found data for synthetic oils of interest. Included are the viscometric properties from 210 to —65 F, the ASTM evaporation rate at 210 F, the Cleveland open cup flash point, and the pour point (or the freezing point). In Table 1 are the liquids having viscosities at 210 F of around 10 cs; in Table 2 are those having viscosities at 100 F of around 10 cs; and in Table 3 are those having viscosities at —65 F of 1500 cs or less. These three groupings are useful because the liquids in Table 1 are in the viscometric and boiling-point ranges of oils that might be useful at the higher temperatures; those in Table 2 are more suited, because of viscometric and low pour-point properties, for use in equipment exposed to a wide range of temperatures; while those in Table 3 are especially suited to applications in fine instruments or other mechanisms having low power input for operation at ordinary and extremely low temperatures.

Although nearly all of the liquids in Table 1 have flash points well above 400 F and evaporation rates of under 0.1%, many synthetics but few mineral oils have pour points much below 0 F. The fluorocarbons are noteworthy because they have pour points above 0 F and evaporation rates which are much greater than those of any other oils in this group. Polyethylene oil has one of the lowest flash points of all and a pour point of only —20 F. But the silicones, the polyalkylene oxides, and the high molecular-weight aliphatic esters are each becoming increasingly more useful as high-temperature oils because of their low volatilities, high flash points, and nonsludging characteristics.

A wide spread in the evaporation rates is exhibited by the liquids listed in Table 2, especially when the low pour-point petroleum oils are compared with most of the synthetics. Many esters, doubly chain-stoppered polyalkylene oxides, and methyl silicones have evaporation rates of from 0.1% to 0.5% as compared to from 8% to 20% or more for the petroleum products. It will be noted that many

---

[4] "Present Problems and Future Trends in Lubrication," by W. A. Zisman. Presented at Annual Meeting of the American Chemical Society, Atlantic City, Sept. 16, 1952.

[5] See "High-Temperature Lubrication of Electric Motor Ball Bearings," by J. E. Brophy, J. Larson, C. R. Singleterry, and W. A. Zisman. Naval Research Laboratory Report No. 3971, May, 1952, Washington, D. C.

Table 1 – Some Synthetic and Reference Oils Having Viscosities around 10 Cs at 210 F

	Viscosity, cs					ASTM Evaporation Rate at 210 F, % in 22 hr	Flash Point (C.O.C.), F	Pour Point, F
	At −65 F	At −40 F	At 0 F	At 100 F	At 210 F			
Grade 1120 Petroleum Oil (MIL-0-6082)	.....	.....	.....	369.0	24.9	<0.1	545	<20
Methylphenyl Silicone (DC-550)	.....	>80,000	2960	95.4	20.7	<0.1	>600	−62[a]
Polyethylene Oil (SS-903)	.....	.....	25,000	219.0	20.1	.....	385	−20
Grade 1060 Petroleum Oil (MIL-0-6082)	.....	.....	.....	121.0	12.4	<0.1	455	<0
Polyisopropylene Oxide Diester (Cal. Research Corp.)	.....	.....	.....	70.8	12.6	.....	465	−55
Polyisopropylene Oxide (LB-300-X)	.....	.....	4000	65.0	11.0	0.4	490	−40
Petroleum Oil (AN-0-6a)	.....	4000 (max)	.....	.....	10.0 (min)	22.0 (max)	265 (min)	<−70
Petroleum Hydraulic Oil, Army (2-79B)	.....	7500 (max)	.....	38.0 (min)	10.0 (min)	.....	225 (min)	<−50
Methyl Silicone (DC-200)	300	170	78	22.0	9.2	<0.1	520	−70
Polyisopropylene Oxide (DLB-130 BX)	200,000	13,000	750	28.4	6.5	.....	485	<−65
Fluorinated Oil (Hooker-Fluorolube Standard)	.....	.....	.....	86.9	5.6	.....	None	> 0
Perfluorocarbon (du Pont FCD-332)	.....	.....	.....	164.0	4.6	40.0	None	> 0
Pentaerythritol Tetracaproate	.....	4690	452	20.0	4.4	<0.1	477	−40[a]

[a] Freezing point.

Table 2 – Some Synthetic and Reference Oils Having Viscosities around 10 Cs at 100 F

	Viscosity, cs					ASTM Evaporation Rate at 210 F, % in 22 hr	Flash Point (C.O.C.), F	Pour Point, F
	At −65 F	At −40 F	At 0 F	At 100 F	At 210 F			
Petroleum Hydraulic Oil (MIL-0-5606)	.....	500 (max)	.....	14.0[a]	.....	Not required	200 (min)	<−75
Synthetic Oil (MIL-0-6085)	.....	2000 (max)	.....	13.0[b]	.....	1.0 (max)	365 (min)	−70
Bis (2-Ethylhexyl) Sebacate	8600	1410	187	12.6	3.3	0.1	450	−67[c]
Bis (3, 5, 5-Trimethylhexyl) Glutarate	17,000	1890	190	11.9	3.2	0.5	.....	<−75
Polyisopropylene Oxide (DLB-65-B)	17,000	1950	190	11.8	3.2	.....	380	<−65
Bis (2-Ethylhexyl) Azelate	8000	1190	156	11.4	3.1	0.2	445	<−75
Synthetic Gas Turbine Oil (MIL-L-7808)	13,000 (max)	.....	.....	11.0 (min)	3.0 (min)	Not required	385 (min)	<−75
Triamyl Tricarballylate	.....	2000	207	10.9	2.6	0.4	.....	<−65
Dihexyl Pinate	6480	1135	149	10.3	2.8	0.3	385	<−75
2-Methyl-1, 3 Pentanediol Dipelargonate	4680	885	123	10.0	2.7	0.5	.....	0[c]
Grade 1010 Petroleum Oil (MIL-0-6081)	(40,000)	3000 (max)	.....	10.0 (min)	.....	Not required	265 (min)	−70
Dipropyleneglycol Dipelargonate	.....	.....	125	9.5	2.7	1.1	.....	−30[c]
Polyisopropylene Oxide Diether (Cal. Research Corp.)	7700	1000	120	9.4	2.9	.....	370	<−65
Bis (2-Ethylhexyl) Pimelate	4970	878	122	9.3	2.7	0.3	.....	<−65
Bis (2-Ethylhexyl) Adipate	4500	807	107	8.2	2.4	0.6	380	<−70
Polyisopropylene Oxide (DLB-50 BX)	9500	905	109	8.0	2.5	16.0	350	<−70
Methylsilicone (DC-200)	.....	60	28	7.9	3.5	.....	325	−80[c]
1, 5-Pentanediol Bis (2-Ethylhexanoate)	5600	920	117	7.8	2.2	0.8	.....	−50
Perfluorocarbon (du Pont FCD-330)	.....	14,500	297	6.3	1.3	100.0	None	−45

[a] Approximate viscosity at 100 F; specification requires a min of 8 Cs at 130 F.
[b] Approximate viscosity at 100 F; specification requires a min of 10 Cs at 130 F.
[c] Freezing point.

Table 3 – Some Synthetic and Reference Oils Having Viscosities of 1500 Cs or Less at −65 F

	Viscosity, cs					ASTM Evaporated Rate at 210 F, % in 22 hr	Flash Point (C.O.C.), F	Pour Point, F
	At −65 F	At −40 F	At 0 F	At 100 F	At 210 F			
Special Low-Temperature Lubricating Oil (MIL-L-17353 BuOrd)	1500 (max)	.....	.....	5.0 (min)	.....	6.0 (max)	300 (min)	<−75
Tetraisooctyl Silicate (Cal. Research Corp.)	1500	365	73.0	7.6	2.5	2.2	.....	.....
Bis (2-Methylpentyl) Glutarate	1390	279	48.1	5.2	1.7	4.1	385	<−65
1, 5-Pentanediol (2-Ethylbutyrate)	1360	280	48.0	5.2	1.7	4.9	.....	<−65
Dibutyl Trimethyladipate	1220	220	43.0	4.9	1.6	9.1	350	<−65
Bis (2-Methylbutyl) Adipate	932	204	39.2	4.8	1.6	5.6	340	<−65
3-Methyl-1, 5 Pentanediol Bis (2-Methyl Pentanoate)	924	203	40.0	4.8	1.7	3.9	355	<−65
1, 5-Pentanediol Bis (2-Methyl Pentanoate)	719	170	35.0	4.5	1.6	4.8	340	<−65
Bis (Amyl[a]) Pimelate	671	180	41.0	5.4	1.9	2.6	375	<−75
Dibutyl (2-Ethyl Glutarate)	627	140	28.0	3.7	1.4	15.1	310	<−75

[a] Oxo process amyl alcohol.

Table 4 – Ranges in Physical Properties of Synthetic Oils

	Range I (Approximately 10 Cs at 210 F)	Range II (Approximately 10 Cs at 100 F)	Range III (Less Than 1500 Cs at −65 F)
Viscosity, cs at 400 F	1.0 to 6.0	...	...
210 F	5 to 20	2.2 to 3.3	1.5 to 2.0
100 F	30 to 100	7 to 13	4.5 to 5.5
0 F	400 to 4000	100 to 200	35 to 48
−65 F	Mostly > 100,000	5000 to 20,000	700 to 1500
Pour Point (ASTM), F	−40 to −65	<−65	<−75
Flash Point (C.O.C.) F	400 to 600+	350 to 450	325 to 385
ASTM Evaporation Rate at 210 F, % in 22 hr	under 0.1	0.1 to 0.5	2.0 to 6.0
Usual Temperature Range of Application, F	−25 to 400	−65 to 300	−80 to 250

of the esters have lower evaporation rates and higher flash points than the other liquids. Viscosities at —65 F range from 5000 cs in some of the synthetics to 20,000 cs in others, while the petroleum products have values of 40,000 cs or more. The extremely high volatility of fluorinated oil FCD-330 is characteristic of the perfluorocarbons having low pour points.

When liquids are required in the viscosity range of Table 3, there are obvious advantages in employing synthetic materials. Thus grade 1005 petroleum oil, which has a viscosity at 100 F comparable to most of the liquids in Table 3, has a viscosity at —65 F of around 3000 cs, a flash point of 225 F or less, and an evaporation rate of 20%. In marked contrast, most of the synthetics in this group have viscosities at —65 F of 700 to 1500 cs, flash points of 340 to 385 F, and evaporation rates of 3 to 6%.

In Table 4 is a useful summary of the spread of the physical properties and temperature ranges of application of the most promising of the tailor-made oils of Tables 1, 2, and 3. Ranges I, II, and III correspond to those exhibited by the most promising liquids of Tables 1, 2, and 3, respectively. Many of the other properties of these synthetic oils are now fully described in the literature and so will only be mentioned briefly. These include the limited oiliness properties of silicones[6, 7, 8] and their excellent oxidation resistance;[9, 10] the stability and nonflammability of the fluorocarbons,[11, 12] as well as the limited solubility for oils and additives; the appreciable detergency of ester oils,[4] their excellent response to inhibitors and oiliness additives,[13] their freedom from engine deposits and sludging, their compatability with petroleum oils and polyisopropylene oxides,[14] and their deleterious effects on certain types of plastics, paints, and rubber; and finally, the detergency of polyalkylene oxides, their freedom from sludging and lacquering, their peculiar water solubility properties, the problem of obtaining a high degree of rust inhibition in some applications,[15] and their attack on some types of paints.[16, 17, 18]

In comparing or classifying the synthetic oils, it should be recognized that, as with synthetic fibers and plastics, no one oil is best for all uses. Much research has been directed to learning the proper fields of application of each of these new liquids, how to blend them, and how to modify their properties with chemical addition agents. Naturally, engineers would like to know which synthetic oil is best for each type of application, or in which oil there exists each important property to the highest degree. However, the frequent existence of physical, chemical, and economic requirements peculiar to each lubrication problem can make it difficult to use either of these two general types of comparison. Wherever possible each potential user or equipment designer should seek guidance from specialists about the applicability of the new synthetic lubricants to his own problems.

Greases

New greases made from synthetic oils were introduced early in World War II commencing with the aliphatic esters[19] and followed by the polyalkylene oxides,[20] the silicones,[21] the silicone-diester blends,[21] and more recently, the fluorinated hydrocarbons. In the postwar years there has been increasing interest in nonsoap gelling agents for greases, and this trend has been accelerated by a threatened shortage in lithium soaps due to an increased demand by the Armed Services for lithium-soap greases. New gelling agents for greases that have been particularly interesting have been colloidal silica, coated bentonite,[22] copper phthalocyanine,[23] various special forms of carbon black, and coated silica; and the most recent arrival in this field is treated Attapulgite.

The advent of these new gelling agents has brought new problems. Both silica and bentonite when uncoated were too sensitive to the presence of water to be useful in many applications of greases. That difficulty was cured by coating the particle with a water-repellent film; however, the resulting greases were deficient in rust-preventive properties. When polar-type rust inhibitors were added to these greases, there resulted a decrease

[6] See ASME Transactions, Vol. 68, May, 1946, pp. 355-360: "Dimethyl-Silicone-Polymer Fluids and Their Performance Characteristics in Unilaterally Loaded Journal Bearings," by J. E. Brophy, R. O. Militz, and W. A. Zisman.

[7] See ASME Transactions, Vol. 68, May, 1946, pp. 361-369: "Dimethyl-Silicone-Polymer Fluids and Their Performance Characteristics in Hydraulic Systems," by V. G. Fitzsimmons, D. L. Pickett, R. O. Militz, and W. A. Zisman.

[8] See *Industrial and Engineering Chemistry*, Vol. 42, December, 1950, pp. 2452-2468: "Boundary Lubricating Characteristics of Organopolysiloxanes," by C. C. Currie and M. C. Hommel.

[9] See *Industrial and Engineering Chemistry*, Vol. 39, November, 1947, pp. 1395-1401: "Polymethylsiloxanes – Thermal and Oxidation Stabilities," by D. C. Atkins, C. M. Murphy, and C. E. Saunders.

[10] See *Industrial and Engineering Chemistry*, Vol. 42, December, 1950, pp. 2462-2468: "Thermal and Oxidation Stability of Polymethylphenylsiloxanes," by C. M. Murphy, C. E. Saunders, and D. C. Smith.

[11] See *Lubrication Engineering*, Vol. 5, October, 1949, pp. 231-235, and December, 1949, pp. 264-269: "Non-Flammable Hydraulic Fluids," by C. M. Murphy and W. A. Zisman.

[12] See *Industrial and Engineering Chemistry*, Vol. 39, March, 1947, pp. 367-374: "Properties of Fluorocarbons," by A. V. Grosse and G. H. Cady.

[13] See *Industrial and Engineering Chemistry*, Vol. 39, April, 1947, pp. 491-497: "Development of Additives and Lubricating Oil Compositions," by D. C. Atkins, H. R. Baker, C. M. Murphy, and W. A. Zisman.

[14] See "Low-Temperature Crankcase Lubricants," by Carl Lamb and C. M. Murphy. Naval Research Laboratory Report No. 3273, March, 1948, Washington, D. C.

[15] See *Industrial and Engineering Chemistry*, Vol. 40, December, 1948, pp. 2338-2347: "Polar-Type Rust Inhibitors," by H. R. Baker and W. A. Zisman.

[16] See *SAE Journal* (Transactions), Vol. 54, May, 1946, pp. 228-237: "New Synthetic Lubricants," by J. C. Kratzer, D. H. Green, and D. B. Williams.

[17] See *Lubrication Engineering*, Vol. 2, December, 1946, pp. 151-157: "Properties and Uses of Some New Synthetic Lubricants," by J. M. Russ, Jr.

[18] See *Industrial and Engineering Chemistry*, Vol. 42, December, 1950, pp. 2436-2441: "Polyalkylene Glycol Synthetic Lubricants," by W. H. Millett.

[19] See *Industrial and Engineering Chemistry*, Vol. 39, April, 1947, pp. 500-507: "Synthetic Low-Temperature Greases from Aliphatic Diesters," by G. H. Hain, D. T. Jones, R. L. Merker, and W. A. Zisman.

[20] See *Mechanical Engineering*, Vol. 73 June, 1951, pp. 469-476: "Some Industrial Experiences with Synthetic Lubricants," by C. H. Sweatt and T. W. Langer.

[21] See *Industrial and Engineering Chemistry*, Vol. 41, November, 1949, pp. 2546-2551: "Extreme Temperature Lubricating Grease from Silicones and Silicone-Diester Mixtures," by R. L. Merker and W. A. Zisman.

[22] See Institute Spokesman (National Lubricating Grease Institute, Kansas City), Vol. 14, No. 2, 1950, p. 13: "Bentone Greases," by C. M. Finlayson and P. R. McCarthy.

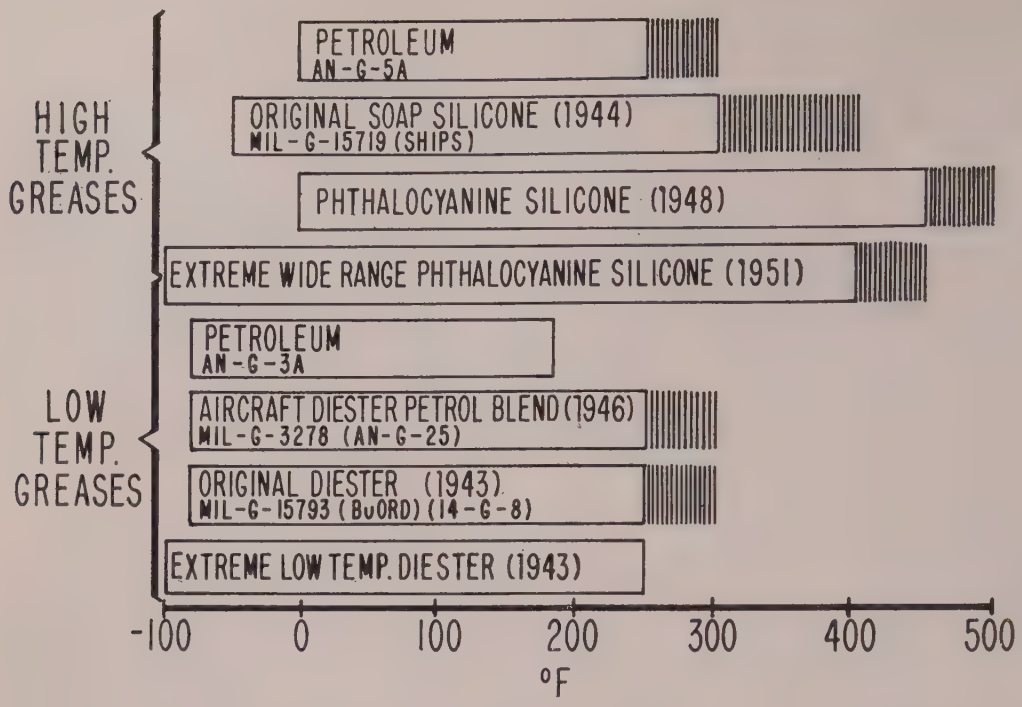


Fig. 2 – Extreme - temperature synthetic greases. Crosshatch indicates temperature in which lubricant functions, but with shortened service life

in the effectiveness of the gelling agent and in the heat stability. Also, the water-repellent films on the particles of the gelling agents appear to oxidize gradually at 250 F or higher, and this together with increased syneresis at such temperatures causes these greases to gradually become hard and caked. Such difficulties still limit the applications of these greases at high temperatures.

Fig. 2 represents in bar chart form the temperature ranges of operability of the more useful of the greases developed in the past 10 years from synthetic oils and the newer gelling agents. Some of the gains obtained by diesters or diester-petroleum blends and lithium or other soaps, are shown in the lower half of Fig. 2. Petroleum-base products, which were useful around 250 F and also at —20 to —40, were of very limited value at —65 F. Others useful at —65 F were of little value above 175 F. In contrast, greases made from diesters, or from diesters containing a minor proportion of petroleum, have excellent, long-life performance characteristics over the temperature range from 250 F to —75 F in some instances and from 250 F to —100 F in others. Furthermore, the former group of ester-base greases have a more limited (but useful) service life at 300 F as indicated by the darkened area at the right end of the bar chart.

In the upper half of Fig. 2 are given the temperature ranges for the new, synthetic, high-temperature greases. The large gains at both extremes of temperature resulting from the development of the lithium soap-silicone greases is evident in the second horizontal bar from the top. Whereas the earlier AN-G-5A petroleum greases were useful from —20 F to 250 F with a much more limited life at 300 F, the soap-gelled, silicone greases were useful from —50 F to 300 F with a more limited life up to 400 F. The phthalocyanine gelling agents have greatly widened the temperature range of greases in general whether they are made from silicones, diesters, fluorocarbons, or petroleum. In Fig. 2 are shown the gains made possible by gelling silicones with copper phthalocyanine. When the silicone liquid used is a methylphenyl silicone like DC-550, the temperature range possible is from —20 F to 450 F, and a more limited life is obtainable at 500 F. Long operation in ball bearings at temperatures much above 250 F requires the use of specially made bearings.[5, 23] When the methyl silicone liquids are used, the resulting phthalocyanine greases are useful from —100 F to 400 F, and they have more limited usefulness at 450 F.

Continued progress can be predicted in the improvement of properties of greases such as their work and storage stability, their resistance to oxidation, and their temperature range of operation. Wider temperature ranges, and much longer storage and operating lives in grease lubricated systems, can be expected with the adoption of properly manufactured greases made from synthetic oils. Indicated improvements in greases, antifriction bearings, and in the end-bell design of

[23] See *Industrial and Engineering Chemistry*, Vol. 44, March, 1952, pp. 556-563: "Phthalocyonene Lubricating Greases," by V. G. Fitzsimmons, R. L. Merker, and C. R. Singleterry.

electrical motors[5] will make it possible to approach much closer to the engineering goal of lifetime lubrication for equipment operating in the temperature range —65 to 250 F. At operating temperatures of 300 F the relubrication intervals will eventually be lengthened to 5000 hr, while at 450 F it will be at least 1000 hr. The increased cost of such new greases, and the necessary bearings to be used with them, should be more than justified by the resulting improved performance and the decreased cost of maintenance and repairs.

Recent Applications

With the advent of gas-turbine-powered aircraft during World War II, better lubricants were soon needed to make the most of the performance capabilities of these new engines. Since the military value of turbojet engines was in their rapid rate of climb and their high speed, a lubricant was needed which would be sufficiently fluid to permit starting at subzero temperatures without dilution or the use of heaters. It was also necessary to retain enough viscosity at engine operating temperatures to support large bearing loads. In combat the extremes of temperature frequently range from —65 F (or lower) to main bearing temperatures in flight, of from 285 F to as high as 400 to 500 F soon after engine shutdown. The upper temperature is caused by the flow to the bearing of heat which had been largely dissipated in flight. Hence, the oil had to be unusually stable to heat and oxidation. Low volatility at elevated temperatures was essential, since evaporation of the oil was increasing oil consumption and was leaving the bearings unlubricated after shutdown. Furthermore, evaporation of the less viscous and more volatile fractions in some oils increased the viscosity of the residual oil and increased the power required for low-temperature starts.

The first lubricants used in gas-turbine-driven aircraft were petroleum oils. Much lower viscosity grades were adopted a few years after World War II, and there resulted viscosity grades 1010 and 1015 which were listed in 1947 in specification AN-0-9 and again in 1950 in MIL-0-6081. Since grade 1015 oil had a pour point of only —50 F and a viscosity of 20,000 cs at —40 F, it was unsuitable for most military aircraft. The physical properties of a typical grade 1010 oil and of three suitable diesters are compared in Table 5. These diesters are only one-fifth as viscous at —65 F, have flash points from 100 to 175 F higher, and are only from one-eighth to one-eightieth as volatile at 210 F as the grade 1010 oils. It was evident to us soon after World War II ended that such lubricants as the diesters, and possibly the doubly chain-stoppered polypropylene oxides, when used either alone or in mixtures, could extend greatly the temperature and altitude range of turbojet-powered aircraft. The properties of several other diester oils of promise are compared in Table 6 with the specification requirements for grade 1005 petroleum oils. Not only had these diester oils larger viscosity indices but also much smaller viscosities and volatilities at —65 F than grade 1005 oils. In fact, some diester oils have the viscosity at 210 F of grade 1010

Table 5 – Properties of Promising Synthetic Oils for Comparison with Grade 1010 Oil and MIL-L-7808 Oil

	MIL-0-6081 Grade 1010	Bis (isooctyl) Adipate[a]	Bis (2-Ethylhexyl) Azelate	Bis (2-Ethylhexyl) Sebacate	MIL-L-7808 Observed	MIL-L-7808 Required
Viscosity, cs, at 210 F	2.53	2.8	3.1	3.3	3.28	(3.0 min)
100 F	10.3 (10 min)	10.0	11.4	12.6	12.68	(11.0 min)
0 F	220	135	156	187	200	....
—20 F	685	330	380	445	490	....
—40 F	2600 (3000 max)	1040	1190	1410	1600	....
—65 F	40,000 (approximate)	7000	8000	8600[b]	10,500	(13,000 max)
Viscosity Index	70	141	146	154	147	....
Pour Point, F	<—70 (—70 max)	<—75	<—75	—67	<—75	—75
Low-Temperature Stability	.....	<—75	<—75	—67	<—65	—65
Flash Point, F	300 (265 min)	370	445	450	450	385
Fire Point, F	.....	430	470	490	490	....
Evaporation Loss 210 F, weight % in 22 hr	8 (approximate)	1	0.2	0.1	0.1	....

[a] Isooctyl alcohol is a mixture of isomers, predominately dimethyl hexanols made by oxo process.
[b] Interpolated or extrapolated.

Table 6 – Properties of Promising Synthetic Oils for Comparison with Grade 1005 Oil

	Air Force Specification 3619 Grade 1005	50-50 Blend Bis (3-Methylbutyl) + Bis (2-Ethylbutyl) Azelate	Bis (1-Ethylpropyl) Sebacate	50-50 Blend Bis (2-Ethylhexyl) + Bis (3-Methylbutyl) Adipate
Viscosity, cs at 210 F	.....	2.0	2.28	2.4
100 F	(5 min)	6.2	7.38	7.6
0 F	.....	59	80	71
—20 F	.....	142	182	148
—40 F	(400 max)	354	504	393
—65 F	(3000 max)	1800	2800	2000
Viscosity Index	80	124	125	150
Pour Point, F	—75	<—75	—90	<—75
Low-Temperature Stability	(—75 max)	<—75	<—75	<—75
Flash Point, F	(225 max)	>350	380	350
Fire Point, F	.....	>390	410	395
Evaporation Loss 210 F, weight % in 22 hr	20 (approximate)	3	3	2

oil, the viscosity of —65 F of grade 1005 oil, and yet they are able to satisfy the specification requirements of grade 1010 oil relating to flammability and volatility.

In 1947 several of the groups of diester oils of Tables 5 and 6, which were inhibited with either of the high-temperature antioxidants, phenothiazine[24] or one of the dialkyl selenides,[25] were recommended by us to the Bureau of Aeronautics for full-scale tests in turbojet engines. The first tests in 1947 at the Naval Air Material Center, Philadelphia, were so encouraging that an extensive testing program on these and very similar diester oils was inaugurated by the Bureau of Aeronautics and the Air Force. Participating agencies included Wright Field, N.A.M.C., the NACA, the Petroleum Refining Laboratory of the Pennsylvania State College, Pratt and Whitney Co., the Allison Division of GMC, and the Gas Turbine Division of General Electric Co. The resulting laboratory tests, full-scale engine tests, and later flight tests in the United States and in Britain, have fully confirmed our predictions that the use of aliphatic diester oils stabilized with the recommended antioxidants would result in much better low- and high-temperature performance, less hazard to fire, very much lower oil consumption, and the ability to start at very low temperatures without auxiliary aids.

One requirement of the Armed Services was that a single specification should care for a lubricant for turbojet engines as well as for the reduction gear systems of turboprop engines. Gear tests proved it was necessary to incorporate an additive in the diester oil in order to support the high unit loads developed. Many of the conventional types of e-p additives could not be used because the same gear oil had to lubricate the antifriction bearings of the turbine and the supporting bearings of the gears. Tricresyl phosphate was the first acceptable wear preventive found which would increase the load-carrying capacity of diester oils without corroding bearings or causing deposits. However, its adoption considerably raised the viscosity at —65 F.

This led to the adoption of specification MIL-L-7808 for a synthetic lubricating oil for gas-turbine-powered aircraft engines. The requirements of this specification are given in the last column of Table 5. It is interesting to compare the requirements of grade 1010 petroleum oils with those of MIL-L-7808. It is seen that the ester oil is slightly more viscous at 210 F and also at 100 F; yet it is only one-third as viscous at —65 F. The flash point of the ester oil is 150 F higher than that of grade 1010 oil, and yet it is only one-eightieth as volatile. In preparing this specification, there was some doubt about the completeness of present knowledge of how to specify such radically different aircraft engine oils. Hence, the device was used of citing in the new specification the chemical composition of a reference oil which would be used to establish any changes in the specifications that service experience should make necessary. Unfortunately, many people have misunderstood this point and have assumed that the Armed Services were departing from their traditional use of performance specifications for materials. As more efficient antiwear agents are found, and as the availability of the newer esters increases, it may become possible to lower the maximum permissible viscosity at —65 F considerably below the present specified value at 13,000 cs. The requirements of MIL-L-7808 are by no means as severe as laboratory data indicate they could be, and many synthetic esters exist which are capable of surpassing the specified viscometric characteristics. However, the present specification was based on the immediate availability of a few of the aliphatic diesters.

Another important application of synthetic oils and greases is in the lubrication of aviation instruments.[13, 26] In general, the introduction of the synthetic oils or greases has greatly decreased the lubrication difficulties encountered in aviation and ordnance gyros. During World War II there was much difficulty with the gyros used for aviation and fire control; and in some extreme instances it was necessary to overhaul them after 50 hr of flight. Although some of these difficulties were due to faults in maintenance or in design, many engineers believed they were often caused by the petroleum lubricants then in use. In postwar years investigations have shown[27] that there are three sources of difficulty. The first source arises from ball-bearing specification and maintenance practices. Nowhere is there more need for careful control of manufacturing tolerances; and many service difficulties have been caused by the use of bearings that were not the best grade commercially available. The second source of trouble is caused by pits and other surface defects in ball bearings which have been traced to faulty or inadequate packaging and storage techniques. As an example, in one military repair establishment it was found that nearly 50% of the troubles encountered with gyro lubricating systems were caused by the use of ball bearings which had not been protected adequately from rusting during shipment or storage. The third source of trouble in certain new gyros has been caused by the high operating temperatures because of which much more high-temperature-resistant oils and greases were needed. We have rectified some of these troubles by the use of either phenothiazine or dilauryl selenide to inhibit the diesters.

---

[24] See *Industrial and Engineering Chemistry*, Vol. 42, December, 1950, pp. 2479-2489: "Mode of Action of Phenothiazine-Type Antioxidants," by C. M. Murphy, H. Ravner, and N. L. Smith.

[25] See *Industrial and Engineering Chemistry*, Vol. 41, May, 1949, pp. 944-948: "Oxidation of Lubricating Oils – Dialkyl Selenides as Inhibitors," by G. H. Denison and P. C. Condit.

[26] See *Industrial and Engineering Chemistry*, Vol. 39, April, 1947, pp. 484-491: "Synthetic Lubricant Fluids from Branched-Chain Diesters," by E. Bried, H. Kidder, C. M. Murphy, and W. A. Zisman.

[27] See "The Lubrication of Gyroscopes," by J. E. Brophy and J. B. Romans (ASME Preprint 52-A-51). Paper presented at ASME Annual Meeting, New York City, Dec. 5, 1952.

In extreme cases it has been necessary to use copper phthalocyanine greases made from esters or from silicone-diester blends.

One of the most interesting of the postwar applications of synthetic tailor-made lubricants has been in the lubrication of automatic weapons such as aircraft cannons and machine guns. The synthetic ester oils have been found so effective for lubricating these essential mechanisms over the temperature range —70 to 150 F that they are now specified by the Navy as indicated in Table 7. The diester greases have also been found ideal for lubricating the feeder mechanisms. Some of the esters listed in Table 3 have been used widely in the past year, others are of promise, and many more equivalent materials have been synthesized.[28] Although these light-grade oils were developed specifically for various military applications, they should be valuable in lubricating a much wider variety of mechanisms which must be operated at subzero temperatures or with very limited power at ordinary temperatures.

Undoubtedly, the use of the new synthetic lubricants can be expected to broaden and become increasingly valuable to the engineer. Probably some of these materials will one day become commonplace in the shop and home for the lubrication of familiar mechanisms varying from electric motors, sewing machines, fans, and typewriters, to locks, sportsman's rifles, and fishing reels.

Chemical and Supply Problems

Although some of the tailor-made lubricants have already found important applications, increasing concern has been expressed in the postwar years about the peacetime and wartime availability of these materials. In Table 8 are shown the generic relations between the various types of acids and alcohols used in the synthesis of the aliphatic esters. As one of the results of our research program of the past 10 years, and also because of the postwar activities of cooperating chemical and petroleum organizations, the early dependency upon sebacic and adipic acids for commercially produced esters has been greatly decreased. A broader and more versatile domestic commercial supply base exists now, and the volume is growing.

There is good interchangeability in the choice of dibasic acids in going from sebacic and azelaic acid to suberic, pimelic, adipic, and glutaric.[13, 26, 28] If the lower molecular-weight acids are used, it is necessary to employ with them branched alcohols of proportionately higher molecular weight. An industrial report in 1947[29] that glutaric acid esters

Table 7 – Specification MIL-L-17353 (BuOrd)
Lubricating Oil, Low Temperature, Special

Viscosity, Kinematic	
At 100 F, cs	5.0 (min)
At —65 F, cs	1500 (max)
Flash Point (C.O.C.)	300 F (min)
Evaporation Rate ASTM (210 F, 22 hr), %	6.0 (max)
Pour Point	—75 F (max)
Neutralization Number	1.0 (max)
Color	No. 5 Union (max)
Static Drop Rust Test at 140 F, hr	168
Fog Cabinet Rust Test at 120 F, hr	96

were not sufficiently responsive to antioxidants has been found incorrect;[28] and this is a good example of the dangers of using insufficiently purified materials in measuring the response of a synthetic oil to antioxidants. Recently, our joint research program with the Southern Regional Laboratory of the Department of Agriculture has pointed the way to the development of a large volume of synthetic oils from turpentine chemicals,[30] and in time this should help to broaden and cheapen our domestic sources of supply of suitable esters. Also, large supplies of glutaric, suberic, and azelaic acids are developing commercially.

In making esters from the glycols or polyglycols, a great many more chemical variations are possible today than were found during World War II.[20, 26] Many new and useful esters have been made[28] by reacting various branched-chain monobasic acids with any one of a variety of glycols, some of which are shown in the right-hand lower half of Table 8. Other new diesters have been made by employing branched-chain glycols reacted with straight-chain acids such as pelargonic, caprylic, and caproic acids,[28] as indicated in the lower left-hand half of Table 8. Some useful aliphatic monoesters having sufficiently low freezing points and low volatilities have been made recently, using branched-chain alcohols reacted with branched-chain monobasic acids – such as pelargonic or caprylic acid – or with straight-chain alcohols reacted with branched-chain monobasic acids. Some of these products have an immediate economic interest; for they will help to find outlets for pelargonic acid, which is a

Table 8 – Sources of Esters of Interest

Prime Source	Acids, Diabasic		Alcohols, Monohydric	Prime Source
Petroleum Coal Oats	Glutaric Adipic		Pentanols	Fermentation Petroleum Natural gas
Stockyard fats Castor beans Turpentine	Azelaic Sebacic Pinic	ESTERIFIED	Hexanols Heptanols Octanols Nonanols	Natural gas Petroleum
	Acids, monobasic		Alcohols, Polyhydric	
Natural gas	2 Ethylhexanoic 3, 5, 5 Trimethylhexanoic		Triethylene glycol Polyethylene glycol Dipropylene glycol Tripropylene glycol Polypropylene glycol Branched-Chain glycol	Natural gas Petroleum
Stockyard fats Castor beans	Pelargonic Caprylic			

[28] See "Further Research on Aliphatic Esters," by G. Cohen, C. M. Murphy, J. G. O'Rear, H. Ravner, and W. A. Zisman. Naval Research Laboratory Report No. 4066, September, 1952, Washington, D. C.

[29] See pp. 31-33, discussion by M. A. Dewey and R. L. Willis in "Symposium on Synthetic Lubricants," ASTM Technical Publication No. 77, June, 1947.

[30] See *Industrial and Engineering Chemistry*, Vol. 45, January, 1953, pp. 119-125: "Pinic Acid Diesters; Preparation, Properties, and Lubricant Applications," by C. M. Murphy, J. G. O'Rear, and W. A. Zisman.

byproduct in the manufacture of azelaic acid from stockyard fats and oils. This should help lower the cost of azelaic acid and the synthetic oils and plasticizers made from it. Considering the already large variety of commercial organic acids, alcohols, and glycols from which suitable esters can be made and also the varied domestic sources from which these materials can be derived, it is evident that there should not be serious concern about the future supply of these lubricants or about the monopolization by any single producer of such a large class of materials.

The necessary raw materials are abundant for the manufacture of the polymers of ethylene oxide and propylene oxide, and relatively low-cost reactions are involved. There are ample supplies of the lime and chlorine used, while the ethylene and propylene needed are readily obtained from natural gas and cracked petroleum fractions.

Although the silicones are still expensive, it is not because of any shortage of the raw materials required, but because of the many chemical reactions involved, the large amount of power required, and the amount of skilled labor needed. The silicones are based upon sand, coal, chlorine, magnesium, and natural gas, of which there are ample supplies for any predictable silicone production of the future. If the demand for these products becomes much greater, doubtlessly considerable economies resulting from increased mass production would permit some reduction in cost. Also, future chemical advances may lower the cost even more.

As for the fluorocarbons, they are made from natural gas or petroleum and either fluorine or hydrogen fluoride, both of which come from calcium fluoride (fluorite). Here, the main reasons for their high cost are the number of chemical reactions and the expensive equipment needed, the high power consumption, and the amount of skilled labor required. Supplies of the raw materials are adequate in terms of any expected increase in the demands for some time to come; however, the supply of calcium fluoride may become limiting. Therefore, it is going to be difficult to produce the higher-boiling perfluorocarbons for less than $10 per lb – even when much larger volumes are produced.

It is evident that the aliphatic esters and the polyalkylene oxides are by far the cheapest and most available of the exciting tailor-made oils now known. Fortunately, both classes of liquids can be made (in part or entirely) from petrochemicals; and probably the proportion of petrochemicals used will grow rapidly in the next decade. For any specialized applications now known or expected, there need be no shortage of any of these synthetic lubricants provided that a lead time of several years is given to the chemical producers concerned in order to increase the rate of manufacture of the desired products.

DISCUSSION

Quality Control Problem Due To Performance Variations

– Elias Margolin
Wright Air Development Center

THE author states that the development of compounded oils is generally simpler than petroleum since there are no undesired chemicals present to cause variations in response to additives from one production batch to another. This may be true in the laboratory using pure compounds, but large performance variations have been found in synthetic oils of different batches that supposedly were of the same composition, and which had been controlled on the basis of physical properties. This variation in performance has greatly increased the quality control problem of synthetic oils.

The use of the reference oil in Specification MIL-L-7808 was not, as stated by the author, to establish any changes in the specification that service experience made necessary. The reference oil is only applicable to the "gear test" of the specification. This reference oil has been deleted from MIL-L-7808A (USAF) but a reference oil for gear tests may again be included by revision of the specification. The British Specification for synthetic oil (D. Eng. R. D. 2487) also uses a reference oil for gear tests. The oil used is a petroleum oil comparable to MIL-L-6082 Grade 1100 oil.

Additional Research on Synthetic Oils Urged

– G. H. Denison
California Research Corp.

DR. ZISMAN has presented an excellent summary and analysis of the types of synthetic oils, their properties, and applications. The presentation adequately bears out this point that synthetic oils are finding a definite application in a variety of mechanisms where their outstanding properties are a necessity. His presentation further makes it evident that, because of the variety of synthetics and the change in the balance of their properties, it is necessary, at least at present, that a user discuss his applications with one well-versed in the field of synthetics. Although, as pointed out in the paper, the time has arrived to increase applications research on synthetic oils, this should not be misconstrued to indicate that the time has come to decrease research on new synthetics.

One example can be cited to show that exploration in this field is still alive. Very recently, materials have been obtained which disprove one of the rather generally accepted rules which Dr. Zisman cites in his paper. This particular rule is one which states that, for a given viscosity at 100 F, no hydrocarbon possesses a better viscosity slope than a normal paraffin or, in more frequently used lan-

guage, no hydrocarbon has a greater viscosity index than that of the normal paraffin which has the same viscosity at 100 F.

API Research Project No. 42, which is under the direction of R. W. Schiessler at the Pennsylvania State College, and which has done outstanding work in synthesizing and determining the properties of pure hydrocarbons in the lubricating oil viscosity range, has recently shown that materials containing cyclopentane rings break the above rule. For example, one compound which contains three cyclopentane rings in a row shows a viscosity slope 1/10-unit less than does the normal paraffin with a comparable viscosity at 100 F. Other examples of this effect of the cyclopentane ring have also been established by API Research Project No. 42.

Some of the simple concepts considered useful in the early stages of synthetic oil development are now known to be in error. For example, it used to be a popular concept that long, slim molecules have high viscosity indexes, and that roly-poly molecules have low viscosity indexes. Such a generalization is obviously untrue since it is broken by the cyclopentane compounds mentioned above, and by a variety of other materials such as the silicate esters. Tetra-2-ethylhexyl silicate, for example, which is certainly not a long, slim molecule, shows the outstandingly good viscosity-temperature slope of 0.69, corresponding to a viscosity index of 170.

It is assuring to find that Dr. Zisman has carefully refrained from using the simplified presentations which some authors have used to interrelate volatility and viscosity. This relation is obviously an important one and one which should be a controlling factor in oil consumption, yet many of the simplified presentations of this property given by other authors lead to erroneous conclusions. Such misuse of simplified volatility-viscosity plots emphasizes the importance of Dr. Zisman's statement that when new applications for synthetics develop, it is advisable to consult with one well-versed in the synthetic lubricant field before choosing the product for that application.

Author's Closure To Discussion

THE difficulties of controlling the properties of tailor-made synthetic oils like esters are certainly much more minor than is the case with mineral oils. However, lack of chemical control of either the base fluid or the additives can result in a significant loss in oxidation stability. Control of purity is particularly important when the lubricant is to be used under conditions of prolonged storage or for exposures to temperatures above 150 C; it is of major importance in producing synthetic lubricants to be used in the engines of gas-turbine-driven planes. I presume that the experiences Mr. Margolin has in mind are concerned with oils for just such uses. I am confident that there are no major chemical problems to be solved. Inadequate chemical control of the oil composition may be causing the troubles he refers to. Chemical engineering problems may have to be solved by industry in order to lower the cost of mass producing synthetic lubricants without loss of any valuable properties. While commercial suppliers develop the necessary know-how, it is essential that the specifications set up by the consumers of the new lubricants be based upon adequate chemical and physical tests. High-grade synthetic lubricants have been made commercially during the past 10 years when suitable specifications were laid down.

There is little to be gained in discussing all the reasoning and history involved in the evolution of the original specification MIL-L-7808. Many people at Wright Air Development Center, the Navy Bureau of Aeronautics, and my own laboratory were concerned. I am confident that as far as some were concerned, the point of view given in my paper was the only reason for justifying the inclusion in that specification of the precise chemical composition for the reference oil.

There is no statement anywhere in my paper that "the time has arrived to increase application research on synthetic oils." Such a statement is manifestly unnecessary because much application research has been going on in this field for 15 years. My greatest dissatisfaction with present trends is that there is too little fundamental research on the subject. Therefore, I cannot understand why Dr. Denison is so concerned lest anyone construe from my paper that "the time has come to decrease research on new synthetics." On the contrary, I referred to an earlier ACS paper[4] in which I indicated specifically many of the research problems yet to be solved in the field of synthetic "tailor-made" lubricants.

As regards the exception, indicated by Dr. Denison, to the rule which had been proposed by Murphy and me, relative to the relation of ASTM slope and 100 F viscosity in hydrocarbons, several points need to be remembered. In checking this rule with any new compound, it is essential to demonstrate that the purity of the compound is adequate; and this requires the elimination of traces of compounds having very different boiling points or 100 F viscosities. Numerous exceptions to our generalizations have been encountered in the past, but with adequate purification the discrepancy always disappeared. Dr. Denison has cited against this rule unpublished data of API Project No. 42 on the ASTM slope and 100 F viscosity of a recently synthesized hydrocarbon, that is, 1,3-dicyclopentylcyclopentane. I find it difficult to take this criticism seriously until the purity of this hydrocarbon has been fully established. It is a fact that the viscometric data on this compound are not consistent with those relating to 10 other cyclopentyl derivatives listed in the API reports. Until the material has been refereed and published in a scientific journal, it does not prove anything. By this I do not mean in any way to criticize the workers on API Project No. 42, because the purification of such high-boiling and readily subcooled compounds is exceedingly difficult; after all, they have not raised the issue here, and they have not yet considered their data sufficiently final to publish it.

Of course it is not true that only long, slim molecules have low ASTM slopes. In 1950 Murphy and I[4] cited a number of examples of highly branched aliphatic tri- and tetra-esters which have ASTM slopes of 0.7 to 0.8. It was stated that in order to have a low ASTM slope the molecule must be very flexible and able to coil; and if ring structures are present, there must be sufficient flexibility elsewhere in the molecule through the loosening-up effect due to the presence of aliphatic carbon-carbon bonds or else ether, thioether, or Si-O bonded atoms. Unbranched aliphatic hydrocarbons appear to be outstanding among all hydrocarbons with respect to the **joint** properties of low ASTM slope and low 100 F viscosity. This is very probably true for the linear molecules in any homologous series of liquids. As far as available information shows, the silicate esters do not act as exceptions to generalizations or conclusions published by me. Dr. Denison believes that tetra-2-ethylhexyl silicate with an ASTM slope of 0.69 is an exception to our generalizations in our reference,[4] but I beg to differ with him. It is an aliphatic compound which would be expected to have a somewhat lower ASTM slope and lower 100 F viscosity than a tetra alkyl ester or pentaerythritol. This is because of the increased molecular flexibility resulting from the large silicon atom in the center of the molecule instead of a carbon atom, and because no keto oxygen atom is present on each of the four alkyl chains. But it is difficult to discuss the silicate esters further because to my knowledge the data cited by Dr. Denison comes from unpublished and unrefereed work of his own laboratory.

Effect of Aerodynamic

IT remains inevitable for the aircraft designer to continuously expand the boundaries of speed and altitude. This process, when viewed from a time scale, seems to assume a trend line somewhat exponential in nature.

The great achievements in recent years in the field of transonic and supersonic aerodynamics has brought supersonic aircraft design well within reach of the drafting board. The aerodynamic feasibility of achieving ever increasing speeds appears to be only limited by the rate of progress made in the field of engine development. It is perhaps because of this fact that the broad problem of aerodynamic heating has not come into focus until comparatively recently.

Extensive research has been carried out by various agencies to determine steady-state and, to a limited extent, transient-state skin temperatures of such areodynamic shapes as flat plates, wedges, and cones. Much work remains to be done to extend and apply the theory of compressible boundary layer flow to the problem of aerodynamic heating. Much more limited progress has been made in determining heating effects on all the systems that render an airplane operational. A typical modern fighter may contain in excess of 100 distinct systems, not the least important of which is the fuel system. That these systems must operate satisfactorily when projected into the regime of significant aerodynamic heating is self-evident. The degree to which this will be accomplished will largely determine the pace by which the aforementioned boundaries can be advanced.

The scope of this paper has been limited to explore the effects of aerodynamic heating on fuel systems, as this is one of the very basic systems.

In examining the effects of aerodynamic heating differentiation must be made between conditions of steady state and transient state.

Deceleration of the free stream air within the boundary layer of an airplane flying at elevated Mach numbers increases the temperature of the boundary layer. Thus heat is caused to flow into the outer skin, adjacent structure, insulation, and across fuel cells into the fuel proper.

As the time after attaining a given Mach number at a given altitude increases, the temperature of each infinitesimal lamina of structure, insulation, fuel cell, and fuel approaches a limiting value.

When all laminae attain their limiting temperature a steady state is said to exist. In the steady state the rate of heat flow into each lamina must be

THE increasing importance of aerodynamic heating at elevated Mach number in relation to aircraft system design is illustrated in this paper, which is a study of the effects of aerodynamic heating on aircraft fuel systems. The fundamental physical behavior of fuel at elevated temperatures and relative effects on fuel system components are discussed. Differentiation is made between steady-state and transient aerodynamic heating of the fuel.

A broad correlation is established between airplane performance and transient heating, and the general severity of the problem is established as a function of Mach number and fuel system parameters.

JULIUS JONAS has been supervisor of the Thermodynamics Section of Northrop Aircraft, Inc., for the past eight years. He is a graduate of the Royal Polytechnic Institute at Turin, Italy, and the University of Michigan. Mr. Jonas was previously associated with Beech Aircraft and Convair's Fort Worth Plant.

Heating on Fuel Systems

Julius Jonas, Northrop Aircraft, Inc.

This paper was presented at SAE National Aeronautic Meeting, Los Angeles, Oct. 2, 1952.

equal to the rate of heat flowing out.

During the transient state the rate of heat flow into each lamina is larger than the rate of heat flowing out, the difference being equal to the product of the thermal capacity and the rate of change of temperature.

It is important therefore to note that the attainment of a steady-state equilibrium by the fuel would essentially correspond to the quasi-infinite endurance of a given aircraft.

Since there are distinct limitations to the maximum endurance attainable by an optimum aircraft at a given Mach number and altitude, the heating process assumes by necessity the nature of a transient state.

With this concept in mind appropriate general aerodynamic and fuel system parameters may be established for the determination of fuel system temperatures.

Upon establishing the range of fuel temperatures to be expected for a typical aircraft, the effect of elevated temperatures proper upon the fuel system may be determined, and the physical behaviour of fuels as well as individual system component performance under such conditions established.

Transient Heating of Fuel System

When air is brought from a state of motion to a state of rest, as is the case at the stagnation point of an aerodynamic shape, the process is defined by an adiabatic compression. If however, the air is brought to a state of rest by friction, as occurs at the surface of an airfoil, the temperature of the boundary layer is raised to a value that is different from that obtained by adiabatic compression. The ratio of the temperature rise due to friction to the temperature rise due to adiabatic compression is referred to as the recovery factor. Fig. 1 gives effective boundary layer temperatures for the case of a turbulent boundary layer.

For the case of an adiabatic surface convective heat transfer will increase with an increase in Mach number and decrease with an increase in altitude. Radiant heat loss to the outside will vary linearly with the surface emissivity and the fourth power of the absolute temperature. Thus at a given Mach number the surface temperature will decrease with altitude due to the decreasing contribution by the convective heat input. Conversely at a given altitude, radiant heat loss becomes more significant with higher Mach numbers increasing the spread in temperatures between emissivity values of zero and unity.

In turning now to the actual case of a nonadiabatic surface where heat is being transferred to the fuel, the transient nature of this process is evidenced by the fact that aircraft operating with chemical fuel has only a limited range. The fuel within fuel cells, integral tanks, or external tanks, will be exposed to elevated temperatures for only a limited time corresponding to the endurance at a given Mach number.

In determining the thermal response characteristics of the fuel several simplifying or qualifying assumptions were made to limit the scope of this paper. (See Fig. 2.)

1. The only form of heat protection considered was insulation between cell and the structure.
2. The cells are assumed to be vapor-pressurized and thus essentially not subject to evaporative cooling.
3. Heat gain was assumed to occur only through the bottom surface. Fin effects due to adjoining bulkheads were neglected.

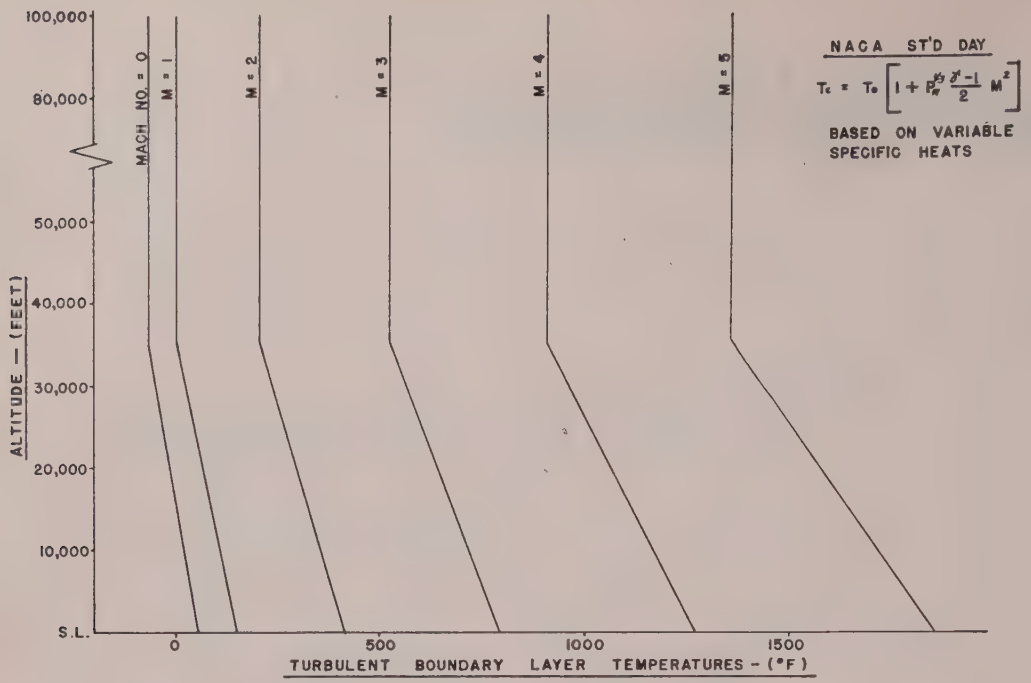


Fig. 1 – Effect of Mach number and altitude on turbulent boundary layer temperatures

4. Chordwise and spanwise heat conduction within the skin structure was neglected.

5. Thermal capacity of the skin structure insulation and cell material was assumed to be negligible.

6. Insulation conductivity, fuel density, and fuel specific heat were assumed to be independent of temperature by selecting mean values. Actually, while both fuel density and specific heat are a function of fuel temperature, the product of these quantities, which appears in the following expressions, is very nearly independent of fuel temperature.

7. The qualifying assumption was made that no thermal gradient exists within the fuel and that it may be treated as an isothermal body as it is heated. This can be substantiated by the following considerations.

When a column of fuel is heated from below, convective currents will not originate off the bottom surface until a critical distribution of density is exceeded. Vertical motion occurs in the form of ascending and descending cellular currents. This may be visually observed by virtue of the variation of the index of refraction.

It has been shown[1] that sustained convection is possible when:

$$\frac{\Delta\gamma_{L-U}}{\gamma_L} > \mathrm{f}\left[\frac{dT_{F_o}}{dl},\ C_{ex},\ l\right]_{\text{critical}}$$

where:

$\frac{dT_{F_o}}{dl}$ = Temperature gradient in preconvective state

C_{ex} = Coefficient of expansion of fuel

l = Vertical distance between fuel boundaries

It may be observed experimentally that the critical value of $\frac{\Delta\gamma_{L-U}}{\gamma_L}$ corresponds to very small preconvective temperature differences. Thus it can be assumed that during the aerodynamic heating process the fuel body will remain isothermal.

Let a fuel cell be considered bounded on the top side by the upper skin and at the lower side by insulation and the lower skin of the aerodynamic boundary.[2]

Let the initial fuel quantity be equal to W_{F_o} and let pW_{F_o} be the fuel consumed during the supersonic flight of duration t_R, whereby p may vary between zero and unity. The zero value corresponds to zero consumption and the unity value to complete exhaustion of the fuel. Thus $\left[W_{F_o}(1-p)\right]$ is the fuel remaining upon completion of the supersonic flight increment and may be termed as subsonic reserve fuel.

A general case is first considered where the fuel level is assumed to decrease linearly from an initial position. The limit case is then considered where the fuel level is assumed to remain constant. (Fig. 2.)

A. *Variable Fuel Level* – The heat balance defining the rise of fuel temperature may be written as:

$$U_{oF}A(T_o - T_F) = W_F C_H \frac{dT_F}{dt} \qquad (1)$$

[1] See Proceedings of the Royal Society of London, Vol. 204, Dec. 22, 1950, pp. 297-309: "On the Stability of a Fluid Heated from Below," by O. G. Sutton.

[2] Royal Aircraft Establishment Technical Note No. 43, October, 1949: "Cooling of Fuel in Integral Tanks," by J. K. Hardy.

where:

$$T_o = \left[1 + (\text{R.F.})\frac{\gamma - 1}{2} M^2\right]$$

$$U_{oF} = \frac{1}{\frac{1}{f_o - f_R} + \sum \frac{x}{R} + \frac{1}{f_F}} \qquad (2)$$

Determination of U_{oF} Value:

a. External unit conductance – f_o:

Reference is made to an Air Force manual[3] and to more recent literature where the external unit conductance f_o as well as recovery factors are extensively treated for the case of the laminar and turbulent boundary layer. Integrated average values of the point unit conductances would be used over the chordwise extent of the surface.

b. Equivalent radiation unit conductance – f_R:

The equivalent radiation unit conductance f_R may be determined from the following considerations.

The net gain of energy by the surface under a fuel cell may be written:

$$Q_R = \left(\sum \alpha A G\right) - e\sigma A T_S^4 \qquad (3)$$

α = Absorptivity

G = Irradiation, Btu/hr/sq ft

e = Emissivity

σ = Stefan constant = 17.3×10^{-10}

It can be shown that for the case of the bottom surface the term $\sum \alpha A G$ consisting of solar and nocturnal irradiation can essentially be neglected. Thus the net loss of energy may be written as:

$$Q_R = e\sigma A T_S^4$$

and:

$$f_R = \frac{e\sigma T_S^4}{T_o - T_S} \qquad (4)$$

from elementary balance considerations:

$$T_S = T_o - \frac{\frac{1}{f_o - f_R}(T_o - T_F)}{\frac{1}{f_o - f_R} + \sum\left(\frac{x}{R}\right) + \frac{1}{f_F}} \qquad (5)$$

The radiation unit conductance f_R may thus be determined by a graphical iteration process from equations 4, 5, and the solution of differential equation 1.

c. Conductivity values of conductive layers:

The value of $\sum\left(\frac{x}{R}\right)$ will depend upon the arrangement of the significant conductive layers. This in turn will depend on the type of installation, both as to type of tank and location. Tanks may be of the bladder-cell, self-sealing or integral-tank type. Location of the tank by virtue of different structural arrangements will be significant as well. Insulation may be used as required.

Only mean values of the conductivities were assumed for the purpose of this paper. In general, however, conductivities will be a function of the bulk temperature. For example, Table 1 shows the variation indicated for 3 lb per cu ft of Refrasil: With some added complexity temperature dependence of conductivities may be included in the analysis.

Table 1 – Temperature Variations

Insulation Temperature, F	Conductivity, Btu/hr/deg F/sq ft
200	0240
400	0373
600	0540

d. Fuel unit conductance – f_F:

Experimental data were obtained by the author to describe the natural convective loss by a heated surface to fuel. The following empirical expression was obtained:

$$f_F = \frac{6.15\,\Delta T_{(S-F)}}{\mu_{S-F}^{0.4}} \qquad (6)$$

f_F values calculated from equation 6 appear to check reasonably well with values calculated from flight test results.[4] Having thus defined the overall heat transfer coefficient U_{oF}, equation 1 may be readily solved.

Let:

$$W_F = W_{F_o}\left(1 - p\frac{t}{t_R}\right) \qquad (7)$$

$$W_{F_o} = Al\gamma \qquad (8)$$

$$\frac{\gamma C_H}{U_{oF}} = \beta \qquad (9)$$

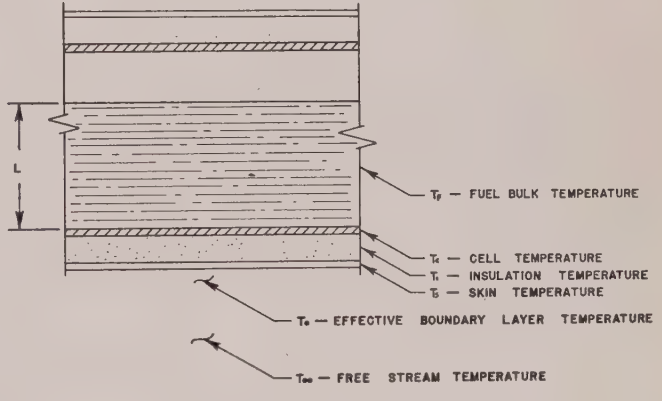


Fig. 2 – Typical section of fuel tank installation

[3] See Army Air Force Technical Report No. 5632: "A Design Manual for Determining the Thermal Characteristics of High-Speed Aircraft." Prepared by University of California, Sept. 10, 1947.

[4] See NACA Technical Note No. 1675, July, 1948, 52 pp.: "Temperature Gradients in the Wing of a High-Speed Airplane during Dives from High Altitudes," by T. Tendeland and B. A. Schlaff.

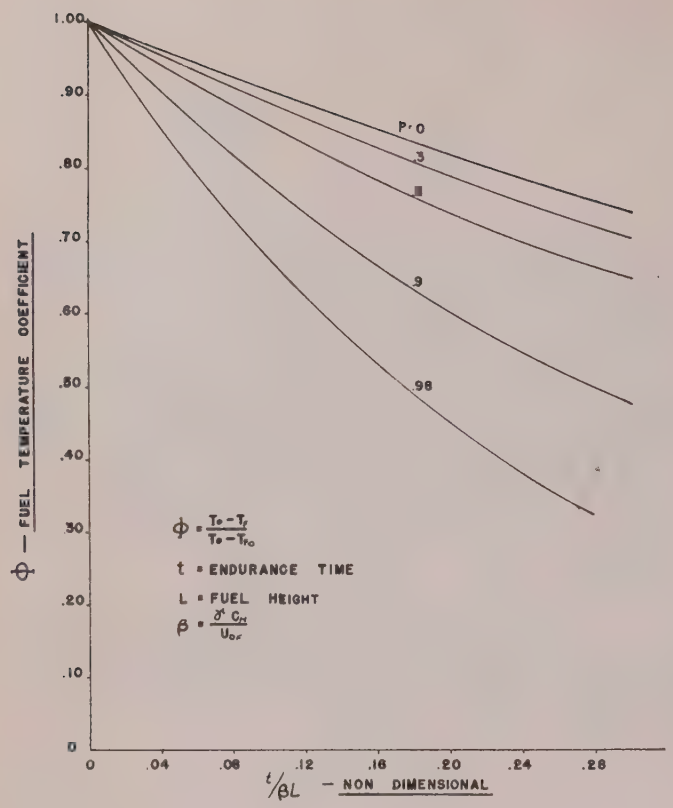


Fig. 3 – Fuel temperature coefficient as a function of nondimensional parameter $t/\beta L$ and $p\%$ fuel consumed during supersonic flight

Thus equation 1 may be written:

$$\frac{dT_F}{T_o - T_F} = \frac{dt}{\beta l\left(1 - p\,\frac{t}{t_R}\right)}$$

Solving and taking the boundary condition:

$$t = 0$$
$$T_F = T_{F_o}$$

$$\frac{T_o - T_F}{T_o - T_{F_o}} = \left[1 - p\,\frac{t}{t_R}\right]^{\frac{t_R}{pl\beta}} \tag{10}$$

B. *Fixed Fuel Level* – Similarly, equation 1 is readily solved for the case where pW_{F_o} is zero, that is, no fuel is consumed out of the tank during the supersonic flight.

Let:

$$W_F = W_{F_o} \tag{7'}$$

$$W_{F_o} = Al\gamma \tag{8}$$

$$\frac{\gamma C_H}{U_{oF}} = \beta \tag{9}$$

Thus equation 1 may be written:

$$\frac{dT_F}{T_o - T_F} = \frac{dt}{\beta l}$$

Solving and taking the boundary condition: $(t = 0,\quad T_F = T_{F_o})$

$$\frac{T_o - T_F}{T_o - T_{F_o}} = e^{-\frac{t}{\beta l}} \tag{10'}$$

Equation 10 and 10′ may be graphically represented by plotting the temperature coefficient, as shown in Fig. 3:

$$\Phi = \frac{T_o - T_F}{T_o - T_{F_o}} \quad \text{versus} \quad \frac{t_R}{\beta l} \quad \text{for} \quad o \leq p \leq 1.0$$

whereby the case of:

$p = o$ corresponds to equation 10′

and:

$o < p \leq 1.0$ corresponds to equation 10

Effect of Aircraft Performance

In order to rationalize the problem of possible fuel temperatures as a function of Mach number it becomes necessary to consider the relationship between maximum range attainable by a given type of aircraft and Mach number. This in effect would define the maximum endurance as a function of Mach number.

The familiar Breguet formula may be written as:

$$R = H_S \eta_T \left(\frac{L}{D}\right) \log_e\left[\frac{W_1}{W_2}\right] \tag{11}$$

where:

R = Range, miles
H_S = Specific energy content, miles
= Btu/lb $\times \frac{778}{5280}$
η_T = Powerplant overall efficiency
W_1 = Initial weight of airplane
W_2 = Final weight of airplane
L/D = Overall lift/drag ratio

In general for conventional fuels H_s may be assumed to be constant. For a given weight class of airplane, $\frac{W_1}{W_2}$ may essentially be assumed to be constant. Thus, in very broad terms it becomes necessary to evaluate the product of powerplant overall efficiency and overall lift/drag ratio.

Simple cycle studies indicate that for the case of the turbojet, with and without afterburning, and for the case of the ram jet, the powerplant overall efficiency varies nearly linearly with Mach number. Typical mean values for an afterburning turbojet would be 20% at $M = 2.0$ and 40% at $M = 4.0$.

Explicit expressions for lift and drag coefficients are quite simple to obtain if the approximations of the thin airfoil theory are used. A typical variation could be $L/D = 8.0$ at $M = 2.0$ and $L/D = 4.0$ at $M = 4.0$. Thus, in very broad terms, it may be concluded that for the purpose of this investigation the maximum range of generic aircraft may be taken to be essentially constant in relation to Mach number variations.

This brings out the very significant point that

the maximum endurance may be expected to decrease essentially linearly with Mach number. Thus, while the fuel heating problem appears aggravated by an increase of Mach number by virtue of increased boundary layer temperatures, a compensating factor is realized by virtue of decreased maximum endurance with increased Mach numbers.

To illustrate the effect of constant range on transient fuel temperatures, let the case of fixed fuel level be reconsidered.

Using equation 10′ and letting:

$$T_{F_o} = T_\infty$$

$$T_e = T_\infty\ (1 + aM^2)$$

$$\frac{t}{\beta l} = \frac{bR}{M}$$

where a and b are constants, one obtains:

$$\left[\frac{T_F}{T_\infty} - 1\right] = aM^2\left[1 - e^{\frac{-bR}{M}}\right] \tag{12}$$

Differentiating equation 12 and putting:

$$\frac{d\left(\frac{T_F}{T_\infty}\right)}{dM} = 0$$

One obtains:

$$e^{\frac{bR}{M}} = 1 + \frac{bR}{2M} \tag{13}$$

Equation 13 is satisfied for:

$$\left[\frac{T_F}{T_\infty}\right]_{\min} \quad \text{at} \quad M \longrightarrow 0$$

and for:

$$\left[\frac{T_F}{T_\infty}\right]_{\max} \quad \text{at} \quad M \longrightarrow \infty$$

Thus, as one might expect, minimum fuel temperatures are obtained at low Mach numbers and maximum fuel temperatures at high Mach numbers.

However, a far more significant conclusion may be realized by examining equation 12. Since the term $\left(\frac{bR}{M}\right) < 1.0$ for Mach numbers larger than 1.0, $e^{\frac{-bR}{M}}$ may be written:

$$e^{\frac{-bR}{M}} = \frac{1}{1 + \frac{bR}{M}} \tag{14}$$

and substituting equation 14 into equation 12, simplifying one obtains:

$$\left(\frac{T_F}{T_\infty} - 1\right) = \frac{abRM^2}{M + bR} \tag{15}$$

Since typical values of bR are in general less than 1.0, $\frac{T_F}{T_\infty}$ will vary nearly linearly with Mach number. Thus the effect of linearly reduced endurance times with increased Mach numbers is to reduce the exponential variation of temperatures with Mach number to a very nearly linear variation.

Effect of Elevated Temperatures on Fuel System

In considering the effect of elevated temperatures on aircraft fuel systems and their components, two distinct groups of problems are apparent. The first one may be associated with the physical behavior of fuels at elevated temperatures, and the second one with the qualification of components operating under high temperatures, particularly in combination with exposure to fuel.

In examining the first group of problems, one of the most significant physical properties of fuel is the wide variation of vapor pressure as a function of temperature. Figs. 4 and 5 were derived from data given in a vapor pressure-temperature monograph.[5]

Fig. 4 gives the vapor pressure as a function of fuel temperature for JP-4 fuel. As a matter of correlation a curve is also shown giving Mach number as a function of effective boundary layer temperature. For the limit case of steady-state heat transfer, the value of fuel temperature would tend to approach that of the boundary layer. Thus, a vapor pressure of 100 psi is realized if the fuel were

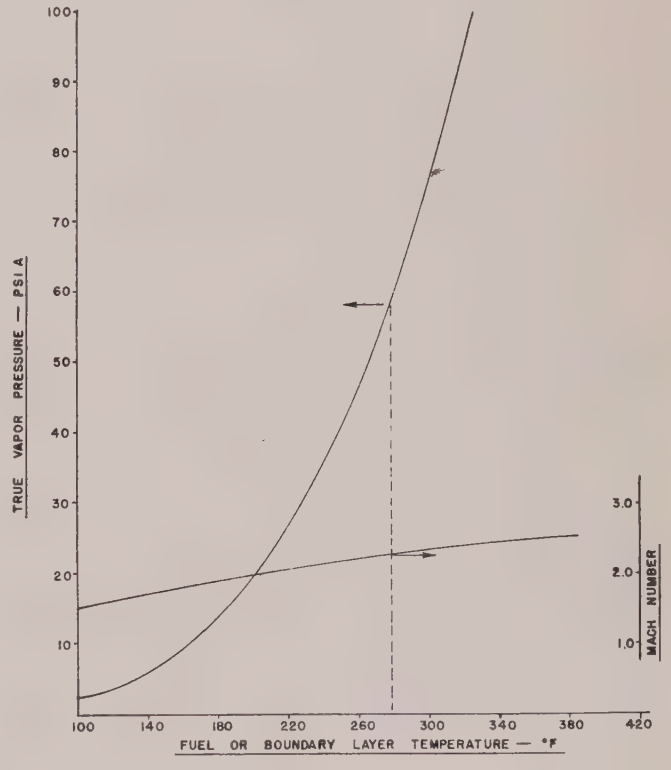


Fig. 4 – Vapor pressure of hydrocarbon of Rvp of 2.5 psi as a function of fuel temperature showing boundary layer temperature variation with Mach number for reference

[5] See *Industrial and Engineering Chemistry*, Vol. 18, March, 1946, pp. 320-323: "Vapor Pressure-Temperature Nomograph," by S. B. Lippincott and Margaret Lyman.

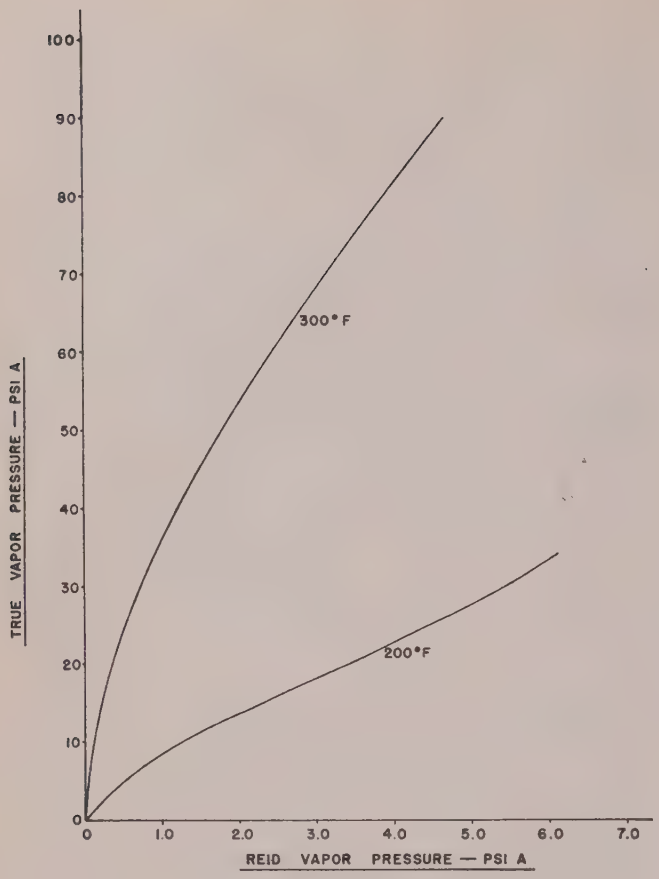


Fig. 5 – Variation of true vapor pressure as a function of Rvp for a constant fuel temperature

to assume a temperature equivalent to that of the boundary layer of a body moving at approximately $M = 2.3$, where fuel cells are bounded by aerodynamically significant external surfaces. Large vapor pressures would determine strength requirements of these surfaces such as to meet aerodynamic waviness criteria.

This clearly points to the conclusion that a steady-state heat transfer process corresponding to an infinite endurance would result in prohibitive weight penalties, thus stressing the importance of the transient nature of the heat transfer process taking place on supersonic aircraft.

To permit a better appreciation of the relationship between Rvp, vapor pressure, and fuel temperature, Fig. 5 was prepared. As can be seen, the higher the fuel temperature, the larger the rate of change of vapor pressure with respect to a change in Reid vapor pressure.

Conversely to limit the vapor pressure at elevated fuel temperatures to reasonable values, unrealistically low Reid vapor pressures would be required.

To limit the vapor pressure to 10 psi at 300 F a Rvp of 0.1 psi would be required.

Since the selection of the Reid vapor pressure is a primary factor in establishing a fuel specification, it might be well to bear in mind that the selection of fuel for military aircraft is predicated heavily upon maximum availability under conditions of emergency, and its ability to start engines under cold weather operation. In general, the higher the vapor pressure selected for a specification fuel the better its availability.

Much controversy exists on the vapor pressure requirements for cold starting. To a limited extent it would appear that combustion-chamber design can compensate for vapor pressure variation under cold starting conditions, and to that extent will permit somewhat lower vapor pressures.

The selection of too heavy a fuel also would aggravate the problem of low-temperature pumpability.

Another property of hydrocarbons of some significance is their ability to dissolve gases in various amounts. Aviation gasoline is known to dissolve air equivalent to 17% of its volume under standard atmospheric conditions. An increase in fuel temperature or a decrease in pressure will decrease the solubility of air and cause it to be evolved from initially saturated gasoline.

Fuel pumpability may not be adversely affected by an increase in fuel temperature, particularly since this is assumed to occur very gradually and during level flight cruise and thus without a significant pressure change, thus causing less air to be driven off at the pump impeller.

Increased volumetric expansion with increased temperature should cause no difficulty particularly where fuel temperatures are limited for structural reasons.

The second class of problems associated with the effects of elevated temperatures pertains to the qualification of components under high temperatures, particularly in combination with exposure to fuel. Components such as fuel cells, seals for components such as check valves, boost pumps, selector valves; switches, such as pressure switches, and float level switches, must be considered.

Fuel cells, like other components, are normally qualified at 165 F. Extension to operation at 200 F should probably be satisfactory. However, at temperatures above 200 F accelerated aging is indicated.

Rubber compounds such as Buna N may retain good physical properties above 200 F, a significant reference point being the vulcanization temperature of approximately 290 F, above which rapid deterioration may be expected. Nylon fabric forming part of the cell material becomes soft at temperatures approaching 320 F.

The deplasticizing effect due to the aromaticity of the fuel, resulting in cracking when exposed to repeated cycles of filling, emptying, and drying, should be further aggravated by the heating process at elevated Mach numbers.

It may also be anticipated that the problem of

fuel leakage in check valves, boost pumps, and selector valves, normally aggravated by the cyclical use of fuels of different aromaticity, will be further complicated by operation at elevated temperatures.

In qualifying various other components such as electrical components, due consideration must be given not only to the maximum temperatures attained but also to the cyclical nature of the temperature variations.

In conclusion it may be stated that careful evaluation of the facts presented in this paper would indicate that the problem of fuel-system operation at elevated Mach numbers for operational aircraft over the next 10 years would come well within the scope of normal engineering processes.

APPENDIX

A = Surface area, sq ft
a, b = Constants
C_H = Fuel specific heat, Btu/lb
C_{ex} = Coefficient of expansion
e = Emissivity
f_o = External unit conductance, Btu/hr/ sq ft/ deg F
f_R = Radiant unit conductance, Btu/hr/sq ft/deg F
f_F = Tank to fuel unit conductance, Btu/hr/sq ft/deg F
G = Irradiation, Btu/hr/sq ft
H_S = Specific energy content of fuel, miles
K = Conductivity, Btu/hr/sq ft/deg F
l = Vertical distance between fuel boundaries, ft
L/D = Overall lift/drag ratio
M = Mach number
PW_{F_o} = Fuel consumed during supersonic flight, %

R = Range, miles
t = Time, hr
R.F. = Recovery factor
t_R = Endurance time, hr
T_∞ = Free stream temperature, F
T_o = Effective boundary layer temperature, F
T_S = Skin temperature, F
T_F = Fuel temperature at time t – deg F
T_{F_o} = Initial fuel temperature, F

dT_{F_o}/dl = Temperature gradient in preconvective state, deg F/ft

ΔT_{S-F} = Temperature difference cell to fuel
U_{oF} = Overall unit conductance, Btu/hr/sq ft/ deg F
W_F = Fuel weight at time, t, lb
W_{F_o} = Initial fuel weight, lb

W_1 = Initial weight of airplane, lb
W_2 = Final weight of airplane, lb
x = Material thickness, ft
α = Absorptivity
β = Defined by equation 9
γ = Fuel specific gravity lb/cu ft
μ = Absolute fuel viscosity, centipoises
η_T = Overall powerplant efficiency
Φ = Temperature coefficient $= \dfrac{T_o - T_F}{T_o - T_F}$

γ_L = Fuel density at lower surface, lb/cu ft
$\Delta\gamma_{L-U}$ = Fuel density difference lower to upper surface, lb/cu ft
σ = Stefan constant $= 17.3 \times 10^{-10}$

DISCUSSION

Fuel Heating Related To Shape of Tank

– J. J. Martin
North American Aviation, Inc.

THIS paper is a good preliminary analysis of the subject of fuel heating in high speed aircraft. No doubt Mr. Jonas has had to simplify some of his work so as not to violate the military security under which he doubtlessly works. He might have pointed out, however, the following refinements which no doubt are open to further refinement.

If one were to assume an arbitrarily shaped fuel tank, there would be some relation, to be found analytically, graphically, or numerically, which would exist between area wetted by fuel and the volume of the wetting fuel at every fuel level, that is:

$$A = A\left(\frac{W_F}{\gamma}\right) = A\,(V_F) \qquad \text{(a)}$$

If this be true and A_o and V_o are initial wetted area and wetting volume, then equation 1 of the paper may be written:

$$\frac{dT_F}{dt} = U_{oF}\,\xi\,(T_o - T_F)\left(\frac{A_o/V_o}{\gamma\,C_H}\right) \qquad \text{(b)}$$

where: $$\xi = \frac{A/A_o}{V_F/V_o}$$

and: $$\frac{A_o/V_o}{\gamma\,C_H} \cong \text{Constant}$$

Now, if U_{oF} and ξ are explicit functions of time, then the integration may be performed and:

$$\Phi = \exp\left\{\frac{A_o/V_o}{\gamma\,C_H}\int_o^t U_{oF}\,(t)\,\xi\,(t)dt\right\} \qquad \text{(c)}$$

However – and this is far more likely the case – if ξ and U_{oF} are available only in graphical form then temperature rise must be calculated iteratively using equation (b) in the form:

$$\Delta T_F = \frac{A_o/V_o}{\gamma\,C_H}\,U_{oF}\,(t')\,\xi\,(t')\left(T_o - T_F - \frac{\Delta T_F}{2}\right)\Delta t \qquad \text{(d)}$$

where: $t' = t + \dfrac{\Delta t}{2}$

One may complain that giving ΔT_F as a function of ΔT_F is unsatisfactory. We may write them:

$$\Delta T_F = \frac{\dfrac{A_o/V_o}{\gamma\,C_H}\,U_{oF}\,(t')\,\xi\,(t')\,(T_o - T_F)\,\Delta t}{1 + \frac{1}{2}\dfrac{A_o/V_o}{\gamma\,C_H}\,U_{oF}\,(t')\,\xi\,(t')\,\Delta t} \qquad \text{(e)}$$

and choose Δt so that: $\frac{1}{2}\dfrac{A_o/V_o}{\gamma\,C_H}\,U_{oF}\,(t')\,\xi\,(t')\,\Delta t << 1$

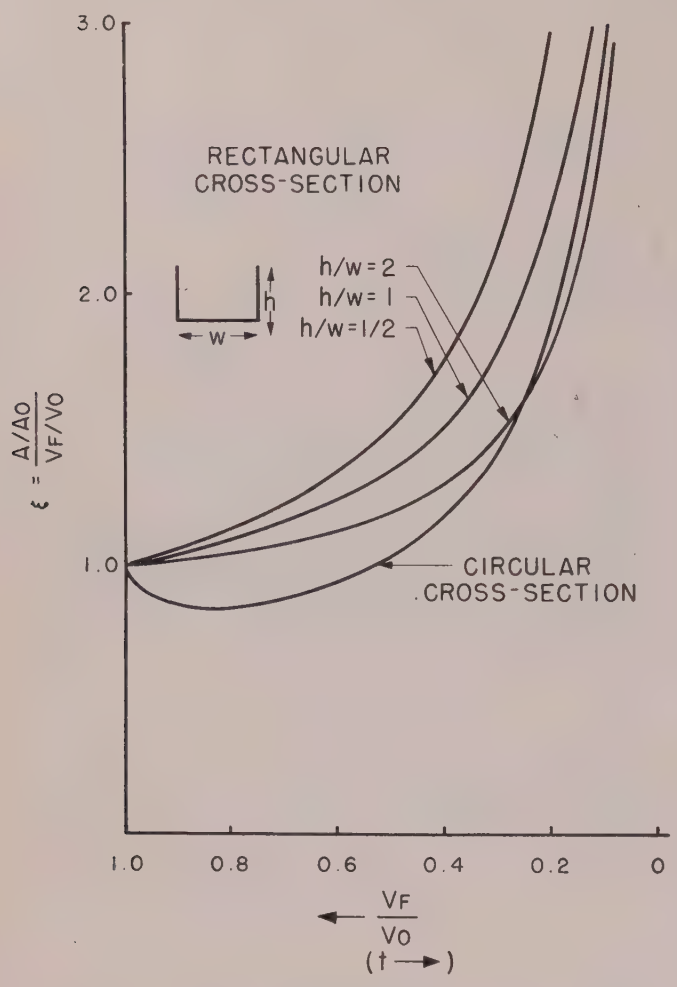


Fig. A – Plot of ξ function for circular-cylindrical and rectangular tanks

Then:

$$\Delta T_F = \frac{A_o/V_o}{\gamma\, C_H}\, U_{oF}\,(t')\,\xi\,(t')\,(T_o - T_F)\,\Delta t \qquad \text{(f)}$$

In our own experience, if $\Delta t = 0.05 t_R$ sufficient accuracy is attained.

As for the variation of ξ, Fig. A shows the plot of this function for circular-cylindrical and rectangular tanks.

Finally, we believe some consideration should be given to radiative heat transfer from the uninsulated inner walls of the fuel tank to the fuel surface. We are not agreed, however, on what basis this consideration be put because of the uncertain nature of the main body of the fuel. The purpose of this comment, therefore, is not to find fault with Mr. Jonas' neglect of this item, but to point out the possibility of an appreciable heat source.

ORAL DISCUSSION

J. A. Quinn, Chance Vought Aircraft: it is my feeling that range of supersonic aircraft would increase with Mach number rather than stay constant, as stated by Mr. Jonas, provided sufficient power is developed by the powerplant.

Mr. Jonas: There have been a number of studies made that indicate that the range remains essentially constant with Mach number in supersonic aircraft, from a range optimization point of view. One cannot limit oneself simply to the Breguet formula; rather one has to consider various weight increments that make up the gross weight, and one has to further insure that such weight increments as related to the powerplant, basic structure, fuel weight, and so forth, are mutually consistent. However, if one were to select different types of aerodynamic shapes the conclusion reached in the subject study may be somwhat modified.

Even if one were to assume that slight deviations existed from the constancy of range versus supersonic Mach numbers it still would not detract from the general conclusion reached in this study. One cannot assume that range can be increased by the simple expedient of increasing propulsive energy without considering the corresponding gross weight increase necessary to carry the increased powerplant and fuel weight.

George Lemke, Consolidated Vultee Aircraft Corp.: I should like to ask Mr. Jonas if he would care to comment on the effect of evaporation on heat transfer and fuel cooling. This effect of evaporation may have been covered in the mathematical derivations, but if so I missed it.

A comparison might well be made to a pressure cooker where the temperature of the liquid remains constant regardless of the amount of heat applied. It appears to me that present day aircraft remain at high Mach number for a relatively short period of time and that, with insulation provided between the wing skin and the fuel, the amount of heat transferred from the skin to the fuel could be readily absorbed by the heat of evaporation from the boiling fuel.

Then there is another problem which concerns me, the upper surface of the skin which does not come into contact with the fuel gets hot quickly when the ram temperature rise becomes large due to high Mach number; the lower surface – when insulated from the fuel – also heats up quickly; however, when integral fuel tanks are used, the spar webs and caps remain cool resulting in additional stresses due to relative expansion between the skin and the caps.

I wondered if it might not be desirable to insulate the spars as well as the skin to reduce differential expansion. Will you comment on that, Mr. Jonas?

Mr. Jonas: As you brought out it is quite natural to think of evaporation as an effective cooling method. However, if one realizes that one pound of fuel contains 18,000 Btu's it becomes apparent that evaporative cooling is a very inefficient utilization of energy. From a weight point of view one would have to compare the weight expenditure of the fuel lost in effecting evaporative cooling versus any other method which might be employed, such as utilization of insulation. Inasmuch as this becomes a consideration for specific installations, this method would have to be evaluated on an individual basis. As to the other questions, the points that you bring out are quite true.

As stated before, it is impossible to cover all the aspects of the problem in this presentation. The fin effects will be there and any heat transfer determination will have to be made on the specific aircraft. However, again it is believed that inclusion of fin effects would not basically alter the order of magnitude of the conclusions reached in this study.

This discussion has been limited to the determination of fuel temperatures. However, depending upon the fuel sequencing arrangements that may exist on the specific aircraft, the exposure of fuel cell material, which becomes uncovered by fuel, may represent a serious problem in determination of material qualifications. Therefore the limitation of fuel temperatures, that may be realized due to limited aircraft ranges, does not necessarily eliminate the problem of elevated temperature qualification of fuel cell material. One would have to resort to the raising of the qualification temperature of material or to some form of heat protection.

Construction and Operation of Brake Testing Dynamometers

Report of SAE Brake Subcommittee No. 3

This paper was presented at the SAE Annual Meeting, Detroit, Jan. 13, 1953.

THERE is always a need for an abbreviated, low-cost exploratory means of evaluating a product. Such a cursory test precedes the final tests prior to ultimate use, but the length of time required for so-called test work either limits the number of items to be tested or introduces an extremely elaborate test program. In the development of brakes for automotive vehicles, the inertia dynamometer has been used for approximately 25 years as a laboratory brake testing machine on which many of the important characteristics of automotive brakes can be observed, as conditions are introduced which simulate operating requirements of the vehicle on the road.

In this capacity the inertia dynamometer is invaluable in permitting observation and recording – in many stages – of the pertinent characteristics of a vehicle brake which indicate its probable performance on the vehicle. In view of the fact that the inertia dynamometer cannot duplicate all road conditions or complications resulting from installation of the brakes on a vehicle, it is limited to the

FORMULATION of a standard test code covering the use of brake testing dynamometers has been completed, approved, and published in the 1953 SAE Handbook as a recommended practice. Increasing use of these brake testing machines has made it apparent that brake evaluation should be reduced to common terms in order to compare performance characteristics. The purpose of this paper is to facilitate use of the code by presenting additional explanatory information.

On the basis of both laboratory and field experience, the working committee has accumulated data covering design features of the dynamometer, dynamometer instrumentation, general features of the test code, and correlation of dynamometer and road tests.

The working committee which formulated the code – and wrote the report given here – includes the following members:

R. K. Super (chairman), Timken-Detroit Axle Co.; D. J. Bonawit, Marshall-Eclipse Axle Corp.; D. P. Dyer, Budd Co.; R. A. Goepfrich, Bendix Products Corp.; G. H. Hunt, Motor Wheel Corp.; J. F. Johnson, Raybestos-Manhattan, Inc.; A. E. Kimberly, Chrysler Corp.; G. K. McCann, Ford Motor Co.; E. O. Reynolds, American Brakeblok Co.; W. S. Rigby, Wagner Electric Corp.; and E. H. Wells, Jr., Johns-Manville Corp.

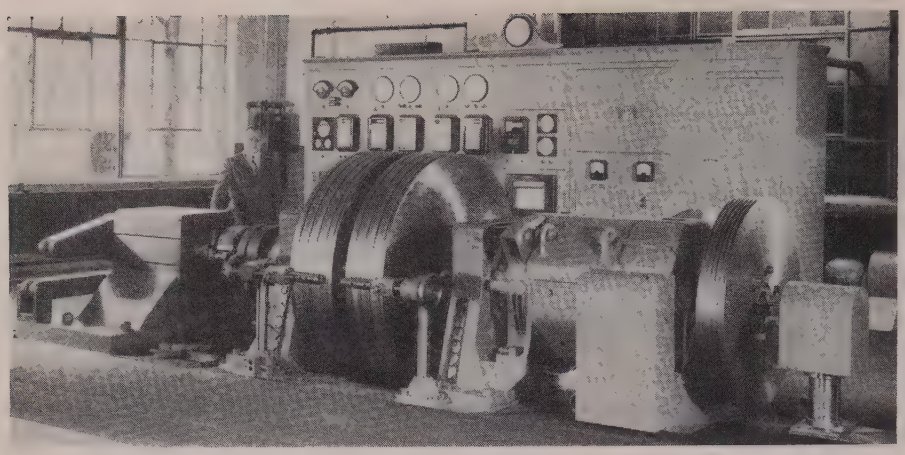


Fig. 1 – Inertia dynamometer for brake testing – electric-drive type

role of screening or providing data for comparisons of brake operating characteristics measurable within its scope. The inertia machine cannot eliminate the test work required to evaluate brakes on the vehicle, but it can supplement and shorten the amount of road testing necessary.

Increasing use of these brake test machines by vehicle manufacturers, brake manufacturers, and lining and drum suppliers, has made it apparent that the basis for evaluating brakes should be reduced to common terms so that performance characteristics could be more readily compared. With this objective, the SAE Brake Committee organized Brake Subcommittee No. 3 to formulate a standard test code for brake dynamometers. Late in 1952 this was completed by the working committee, approved by the Technical Board, and released for publication in the 1953 SAE Handbook as a recommended practice.

It will be impossible to follow this code to the letter, as the design characteristics of the dynamometers, the brake designs, and end use, will necessitate variations in procedure, with subsequent deviations in the code. The code, nevertheless, will serve as a guide in securing information on operational characteristics of brakes and component parts; and by establishing a uniform method of test preparation, procedure, and compilation of data, a proper comparison of brake performance can be made. The recommended data forms and curve sheets are patterned after those which have been in use for years in automotive engine testing where standardized forms list engine performance characteristics.

The purpose of presenting this paper is to elaborate on the basic information given in the published code. Accumulation of this supplementary data relating specifically to inertia brake dynamometers was believed to be highly valuable and of particular importance to all users of the code. It is intended that this paper provide available information which could not be included in the code because of limited space or time. The code has been in use in a number of laboratories during the past two years. Revisions and modifications have been made on the basis of actual laboratory experience.

To provide a semblance of grouping, the subject has been divided as follows:

1. Brake dynamometer – design features.
2. Brake dynamometer – instrumentation.
3. SAE test code – general features.
4. Correlation of dynamometer and road tests.

The basis on which the inertia dynamometer has been adapted to the testing of brakes is predicated on its ability to simulate the vehicle kinetic energy, road speed, and in many instances, cooling conditions. The kinetic energy of that part of the weight of the vehicle to be retarded or stopped by the brake is given by the formula:

$$\text{KE (vehicle)} = \frac{WV^2}{64.4}$$

where:

W = Weight on wheel

and:

V = Vehicle speed in fps

The equivalent kinetic energy of the dynamometer is obtained by equating the above KE to be handled by the brake to the kinetic energy of the dynamometer flywheel, given by the formula:

$$\text{KE (flywheel)} = \tfrac{1}{2} I\omega^2$$

where:

I = Moment of inertia of flywheel

and:

ω = Rotative speed of tire on vehicle

These formulas are covered in detail in the test code data sheets.

This energy and speed relationship between the dynamometer setup and the installation of the brake on the vehicle, will explain the function of the inertia dynamometer and the value of its basic design features in tests of this type.

Brake Dynamometer – Design Features

In general, the inertia dynamometer is fundamentally a carefully supported laminated flywheel

which, when revolved at any desired speed, possesses a known amount of kinetic energy. In bringing the revolving mass to a stop by a braking mechanism, the kinetic energy of rotation is converted into heat energy by the brake and subsequently dissipated through this device. Basically, a brake is an energy-converting mechanism. (See Figs. 1 to 5.)

This concept of the energy relationship between the flywheel and brake indicates that the flywheel is the most important element of the inertia dynamometer; consequently, its design, construction, and control should receive careful attention. The supplementary components of the dynamometer are also of vital concern. Therefore, in the construction of inertia dynamometers the following essential elements should be adequately considered:

1. Foundation.
2. Laminated flywheel and supports.
3. Brake support.
4. Driving mechanism.
5. Control and recording instruments.
6. Blower or cooling equipment.

1. The foundation for an inertia dynamometer provides and maintains accurate alignment of the main pedestal bearings, the driving mechanism, and the brake support element. Most foundations consist of a cast metal bedplate set on a concrete foundation. Such a bedplate serves as a well-aligned solid reference face for the construction of the machine superstructure.

2. The laminated flywheel generally consists of a master plate, keyed or otherwise, fixed to a heavy, rigid shaft. A group of balanced discs may be added to this master plate to provide a rotating mass having total energy equivalent to the load to be handled by the brake on the vehicle. These added discs are secured in accurate relationship to the master plate by through-bolts. The entire shaft and flywheel assembly is supported on trunnion bearings which are self-aligning and of the best antifriction type. The brake drum is mounted on a flange at one end of the main shaft and the dynamometer driving mechanism is normally connected at the opposite end of the shaft. Some of the larger dynamometers are using flywheels in excess of 60 in. in diameter with total weights around 15,000 to 20,000 lbs.

3. The brake support provides a mounting for the brake assembly. This is usually a flanged through-shaft carried on antifriction bearings in a pedestal or tailstock. On current dynamometers, either the shaft can be moved longitudinally to withdraw the brake assembly from the drum, or the entire pedestal or tailstock can be moved along the dynamometer base for this purpose. With this construction, the torque arm is attached to the end of the through-shaft opposite to that on which the brake is mounted. The rotative force in the torque arm is measured by various devices such as hydraulic pistons, strain gages, and platform scales, and indicates the output of the brake. (See Fig. 3.)

4. Two types of flywheel driving devices are in general use. One is a directly connected electric motor (see Fig. 1) and the other is an oil gear type (see Fig. 4) wherein an electric motor drives an oil pump, which in turn propels an oil driving unit directly connected to the flywheel main shaft. These propelling devices vary in size, depending on requirements for torque and speed, up to 150 hp. High torque capacity with a minimum drag on the main shaft after cutoff are prerequisites of these units. Rotative speeds vary from 1000 rpm for testing wheel brakes, to upwards of 4000 rpm for the smaller machines on which propeller shaft brakes may be tested.

5. The control and recording devices differ widely, depending on the type of brake test being conducted. Many of these tests are time cycle tests wherein a synchronous motor maintains a stop-start brake application cycle through a series of cams and switches. A second type of test places the control of the dynamometer on a temperature basis so that the starting of the cycle is a function of the brake temperature. This type of control has been found necessary in obtaining consistent results.

The recording of the stopping time, brake torque, torque characteristic, temperature, noise, and actuation unit travel or input effort requirement, necessitate instrumentation of the best type. This is covered in a later section of the paper. (See Figs. 2 and 5.)

6. Control of the cooling air to the brake, so that the test conditions simulate as closely as possible the cooling existing on the brake as it operates on the highway, has been a difficult problem. Many of the original dynamometers were built with an

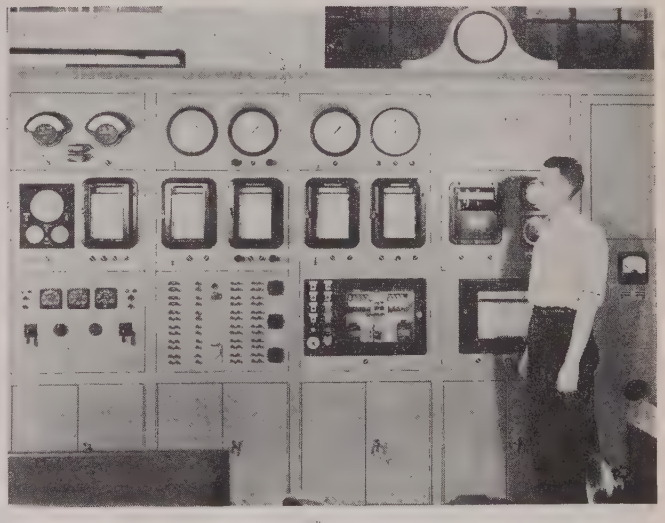


Fig. 2 – Instrument and control panel for electric-drive dynamometer shown in Fig. 1

exhaust fan as the only means of circulating air over the brake drum. Actually, these were for the purpose of carrying off smoke and dust. The more recent dynamometers have both exhaust blowers and pressure fans connected to a housing over the entire brake assembly. Air speeds up to 2200 fpm, as recommended in the code, are within the capacity of this type of installation.

Inertia dynamometers for testing vehicle brakes have grown in size and special design so that brakes for all types of vehicles, as well as railroad train brakes, can be handled. The machines vary in cost up to $150,000 for the latest installations. These are valuable and complicated tools for testing brakes and require the best in technical know-how for operation to make them worth this initial expenditure.

Brake Dynamometer – Instrumentation

A. Purpose of instrumentation.

1. Measure brake effectiveness:

(a) *Brake Torque* – A brake test machine must have instruments for measuring and maintaining a desired output, since it must not only simulate service conditions but it must measure and control work of the brake. With a torque recorder, a written record is obtained of brake output which, of course, is one of the main measures of brake effectiveness. In addition, a record of the torque required to maintain the brake output at desired levels is secured.

(b) *Stopping Time* – As with brake torque, the stopping time is likewise a measure of brake effectiveness, since it can be correlated directly with deceleration rates and stopping distances on test vehicles. These foregoing are accepted criteria for establishing brake friction level, and the measure of stopping time – whether by clock or by measurement of the time coordinate on an output chart – is a necessary item in laboratory evaluation of friction materials.

2. Measure brake input:

(a) *Input Pressure* – The full evaluation of a dynamometer brake test must take into account not only the controlled torque but the amount of input required to maintain this torque in much the same manner as deceleration is matched with pedal pressure on road tests. Effectiveness cannot be measured with output values alone, nor can input values be discarded when a test is run with the output controlled. The interdependence of output and input is the most important relationship in evaluation of brake dynamometer work and for that reason, whether the measurement is visual or recorded, input values are closely watched throughout tests.

(b) *Brake Speed* – The obvious effect of brake speed on brake dynamometer evaluation is brought to light when considering the purpose of laboratory brake work, that is, correlation with conditions met in service. The energy absorbed in stopping a vehicle from 60 mph is four times as great as that required to stop the same vehicle from 30 mph. This kinetic energy-velocity relationship is basic, and it is the important factor in fade test work where temperatures are raised by making the brakes absorb a good deal of energy in a short period of time. Speed control is, therefore, not only important for this basic brake fade information, but it is also important in the evaluation of brake effectiveness at different speeds. Tachometer data in drum revolutions per minute, or converted to vehicle miles per hour, not only aids in bringing brake analysis from the field into the laboratory,

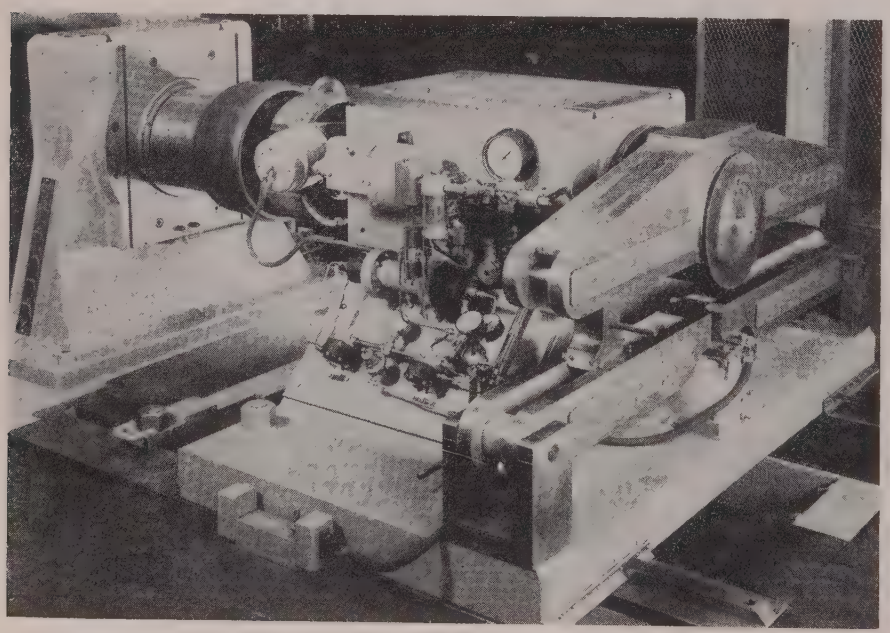


Fig. 3 – Brake mounting and torque control mechanism for electric-drive dynamometer shown in Fig. 1

but it allows closer control than can be maintained in the field.

3. Measure brake characteristics:

(a) *Temperature* – It is not fully correct to call temperature a brake characteristic, as it is more a tool for analysis and, as such, is an important aid in the correlation of data between the field and the laboratory. The energy converting rate of a vehicle brake and a dynamometer brake can be made alike by recording the equilibrium brake temperatures at which they settle; also, tests can be run in the laboratory at the same temperatures as in the field by means of temperature measuring devices. Regardless of the position of the thermocouple, as long as it is in the same position in the dynamometer brake as it is in the brake on test on the road, a correlation of road test and laboratory test data is possible. In addition, temperature data can be matched with input and output data to determine breakoff points where a brake may start fading, or other such behavior traits where the temperature is the cause of the change. These temperature effects are, therefore, of basic importance in overall brake design.

(b) *Wear* – The durability of the lining on a brake, as well as other component parts, is of particular importance to overall brake design, since it can be measured when conditions are controlled. This problem of maximum brake performance with longer life is basic in the industry. Micrometer checks before and after the controlled brake dynamometer runs are the accepted method for checking wear.

(c) *Other Characteristics – distortion and noise* – Characteristics which are not so readily measured, such as distortion and noise, are at times the most perplexing of all.

It is common knowledge that brake noise, although it does not alter such main brake characteristics as brake effectiveness and life, can be a major source of field complaints. In order, therefore, to measure noises and work toward bringing them to a commercial level, decibel counters are used to measure the noise level, and assorted devices indicate the frequency or pitch of noise. It is well to point out that instruments such as these are not widely used in brake dynamometer work for the human ear is considered a satisfactory noise-measuring device. In addition, it is more practical in most cases to test for noise in the field.

Distortion measurement in the brake elements can be made with strain gages or with dial indicators and a stroboscope, and the instantaneous distortion at definite positions can be observed.

4. Test Control:

(a) *Time* – It may be noted that during the foregoing discussions on the importance of certain measurable brake performance characteristics, the word control was used numerous times. Such measurements as input, output, and temperature are almost meaningless unless procedures can be dupli-

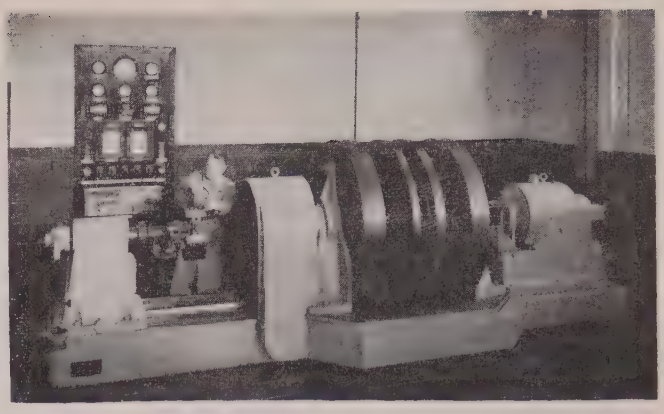


Fig. 4 – Inertia dynamometer—oil gear drive

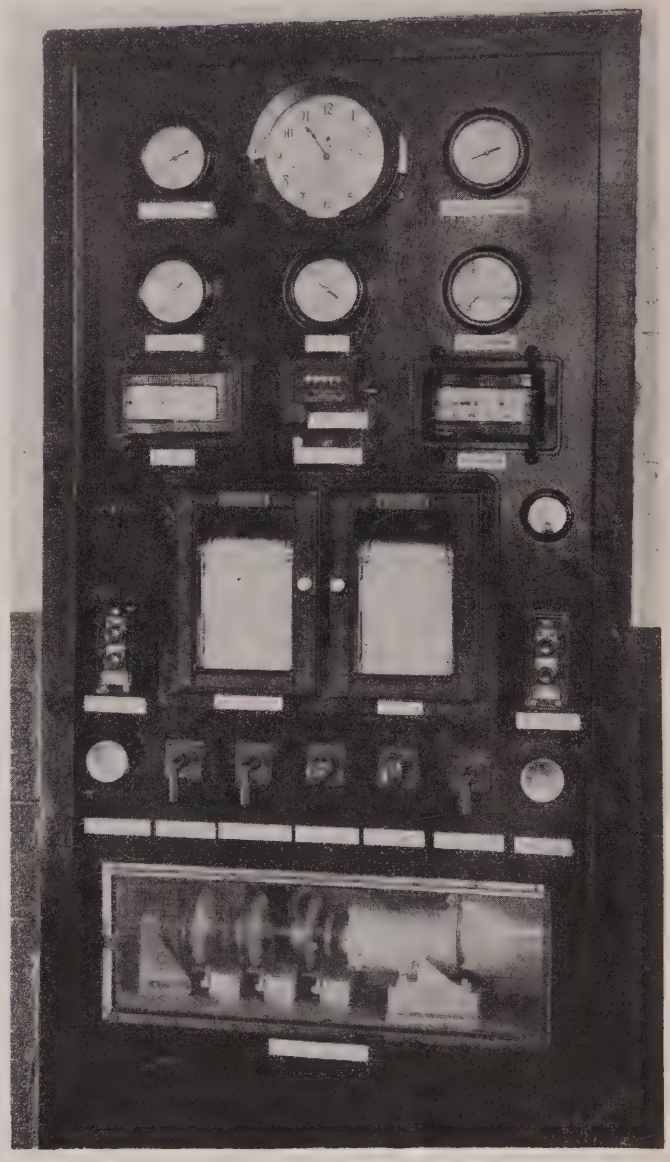


Fig. 5 – Control and recording devices for dynamometer shown in Fig. 4

cated from one test to the next, and this is done by controlling the interval between application. The importance of time control goes back to basic temperature-energy relationships. Thus, if a shorter interval is maintained between stops at a predetermined output level, there will be less time to dissipate the brake heat, and the brake will level off or have a final brake temperature higher than the test where the longer interval is maintained. Past test work has shown that the same brake lining will show more wear doing the same total work at a higher temperature than at a lower temperature, and for reasons such as these, we must maintain close control over the time cycle.

(b) *Number of Cycles* – The overall severity of a test depends on the total amount of work done, and such individual items as fade runs should go to the same number of stops from one test to the next. The need for controls to set the number of cycles and for counters to actually count the number of stops is obvious.

(c) *Torque Control* – A need which has always been recognized but which has not always been satisfied is for a device which will automatically control the torque developed by a brake. There is no problem connected with running input controlled tests, where the input load is maintained or varied according to schedule and the output varies not only in proportion to the input but also according to the friction characteristics of the brake. To make measurements of expected life and to maintain brake energy outputs at the same level for fade work, a brake must do the same work from one test to the next. There are, of course, sections of a test where brake effectiveness is determined by holding definite inputs and observing the output rate. However, for fade, wear, and overall friction behavior, controlled torque tests must be used since the actual behavior in the field is based on output. Various means, including manual manipulation, have been used for controlling torque on brake dynamometers, but this control has not lent itself to standardization.

B. Description of dynamometer instrumentation.

1. Torque and time measurements:

(a) *Graphic Recorder for Torque Measurement* – The measuring and recording of torque can best be done on a strip chart of the Esterline-Angus graphic recorder type. These recorders can be motor or spring-driven at a speed which runs one coordinate at a certain distance per unit time, while the torque is normally plotted by a pen carried by a bourdon element. As a rule, these recorders are signalled by pressure to the bourdon elements, but they may be actuated electrically for remote measurement or for strain gage measurement. The range and accuracy required for this instrument would have to be determined in the overall layout of each dynamometer. The range would depend on the maximum loads which the brake would stop, and upon the accuracy on the full-scale accuracy rating of the manufacturer (the standard accuracies being in the neighborhood of 1 to 2% of full scale).

(b) *Time Clock and Graphic Recorders for Time Measurement* – There are two instruments which can be used for measuring the time for stopping the flywheel on an inertia brake dynamometer. A time clock can measure stopping time (which is inversely proportional to what would be the equivalent deceleration rate) by being actuated electrically by a pressure switch connected to the torque measuring system. In addition, the time coordinate can be measured on the graphic torque recorder and from it the time for stopping computed. The clock will give much greater accuracy, but the graphic recorder serves as a written record. As noted in an earlier section of this report, stopping time is a measure of brake effectiveness and gives the best correlation with stopping distances (fifth wheel – feet to stop) and deceleration rates. For a standard, an electric time clock of the Standard electric clock type is in wide use and, as previously noted, it should be actuated by a torque-operated switch. For recorder time values, the strip recorders of the Esterline-Angus type are recommended.

2. Input pressure and speed instruments:

(a) *Graphic Recorder for Input Pressure* – The basic strip recorder of the Esterline-Angus type previously described in the section on torque recorders is recommended for input measurement and recording, since, as with the torque recorder, it does a satisfactory job and gives a written record of input. Range considerations should be considered in the original design. A recorder with the accuracy normally maintained by the manufacturer is entirely satisfactory (usually 1 to 2% of full scale).

(b) *Electric Tachometer* – The necessity for accurate speed measurement to determine the total energy output of the brake is obvious. The general layout of the tachometer can be varied to fit individual design, with the installation gear or belt driven from the main shaft, or with a direct connection or clutch-type friction connection in line with the shaft. The unit is simply a mechanically-driven electrical generator of very small capacity which signals a galvanometer instrument calibrated directly into rpm. Range and accuracy are, of course, subject to individual design and the manufacturers' specifications. The type normally used, and entirely satisfactory for most dynamometers, is a visual instrument, but a strip chart of the Esterline-Angus type is available where a written record is desired. Tachometers of the Weston and Reliance type are satisfactory for visual speed measurement.

3. Brake characteristics:

(a) *Temperature Recording Potentiometer* – Various layouts for thermocouple positioning have been adopted by those operating brake dynamom-

eters, and as this subject lies more with the actual physical design of the machine rather than with its instrumentation, it will not be discussed in this section. The temperature-recording potentiometer should be a direct-reading recorder instrument with a standard cell. It is important that temperature measurement balancing system be used for temperature measurement, since these potentiometer circuits give the advantage of a circuit in which the end result is not affected by length of lead, change in air temperatures near the thermocouple leads, changes in running contact as with slip rings, and so forth. Also, it is felt that a multiple-point recorder can be used to advantage in brake dynamometer work, since with it observation can be made of simultaneous temperatures at different positions in the brake assembly. The overall range for most brake work is 0-1400 F, but where special tests for such items as aircraft brakes are concerned, higher ranges must be used.

An important item in temperature measurement is the subject of response speed. When measurements are made at two temperature levels with a 2-point recorder, it is important that the recording element move back and forth between points rapidly enough to mark the point accurately and plot lines that do not have their points too far apart. In order to do a satisfactory job on this score, a response speed of 2½ sec for full-scale measurement should be used.

For the foregoing reasons, potentiometers of the Leeds and Northrup Speedomax, Brown Electronik, and Bristol Dynamaster type are recommended. It may be well to point out in this section that these potentiometers may be used to control routine wear cycles where temperature control is desired.

Summarizing operating and construction features necessary for this instrument, the following are needed: (1) a balance-type potentiometer with standard cell; (2) a multiple-point recorder with a response speed of 2.5 sec for full-scale measurement, and (3) range 0-1400 F for all normal applications (higher ranges for special work).

(b) *Discussion of Wear* – As was noted earlier in this report, the measurement of brake and lining wear is an important part of brake dynamometer research, but no standard instrument has been accepted for wear measurement during test. Devices for checking pedal reserve and the like have been incorporated in some dynamometers, but the most universal method for determining wear remains micrometer measurement.

(c) *Discussion of Other Brake Characteristics* – The full analysis of braking problems is up to the experimental section doing the investigating, but characteristics other than the measurable quantities, such as input, output, temperature, wear, and so forth, are sometimes of a very important nature. The problems of noise and distortion were discussed in the preliminary section on the purpose of brake dynamometer instrumentation. As will be noted by referring to that section, no standardization of the special devices for that work is possible.

4. Test Control:

(a) *Timer Control* – Where temperature control is used for routine wear cycles, a recording potentiometer of the type described in the section on potentiometers will be used. However, for the many sections of a routine schedule where time interval governs the cycle, a simple timer such as the Automatic temperature control or Eagle Signal Flexopulse Timer is recommended.

(b) *Cycle Controller and Cycle Counter* – A cycle controller or counter is used in conjunction with either the temperature control for routine wear cycles or the timer control for performance tests. A unit of the Eagle signal type for control may be used for this in conjunction with a Veeder-Root type counter.

(c) *Constant Torque Control* – A device such as the Uni-Tork control valve originally put in use by Raybestos-Manhattan Division is recommended for torque control work. There are many variations on this valve, particularly on the signaling side. The basic valve can be either hydraulically or air-operated. That is, the device measuring the torque may be either an air unit similar to the Hagan ThrusTorq, or it may be the standard hydraulic system, since the actual behavior of the valve is independent of the fluid used.

Essentially, the valve has two sides – both usually diaphragms. One side sets up the desired pressure, and the other receives a signal of what the (output) pressure actually is. The unit is made so that if the actual output pressure is less than desired, more fluid will be applied on the input side, thus raising output, or, if the output is too high, the valve cuts off the input to the brake. Basically, that is the description of the Uni-Tork valve.

General Features of Test Code

The complete text of the new SAE Brake Test Code-Dynamometer, as it appears in the 1953 Handbook, is included as the Appendix of this paper. Throughout the discussion which follows reference will be made to this code.

As outlined in the code, a complete standard dynamometer brake test includes the determination, at various speeds and line pressures, of the following points:

1. Torque output and deceleration (various speeds and brake input).
2. Evaluation of lining life (wear tests).
3. Brake characteristics (fade and recovery tests, noise, chatter, grabbiness).
4. Drum evaluation (wear, heat-checking, scoring).

From these determinations the following graphs are plotted and charts compiled on standard forms:

1. Torque output versus brake input, and decel-

eration versus brake input graphs (effectiveness tests).

2. Characteristic torque graphs from effectiveness tests at maximum line pressures (individual stops).

3. Lining fade and recovery graphs (deceleration and temperature versus stop number).

4. Lining-wear tabulation – wear per test phase and total test.

5. Drum – statement of condition (and wear, if any).

From the foregoing, it will be seen that the test procedure outlined in the code will provide all of the information which is generally required to determine the operating characteristics of a brake.

A. *Preliminary Preparations* – The code designates that certain preparatory work be completed on the brake and its components prior to the start of the actual test procedure. This includes brake inspection, thermocouple installation, drum measurement, bleeding, and so forth. Forms are provided for recording measurements of drum diameters and profilometer readings.

B. *Initial Lining Break-In* – After the preliminary preparations and measurements have been completed, the brake is mounted on the dynamometer and operated through a break-in cycle, so as to obtain a minimum of 80% lining contact, uniformly distributed. This break-in cycle is run with a flywheel speed of 30 mph, and a line pressure producing a deceleration of approximately 11 ft per sec per sec. Flywheel speed for 30 mph is calculated from the tire rolling radius of the vehicle on which the brake is used. Deceleration is calculated from the time elapsed between the initial brake application and the time at which the brake brings the flywheel to a standstill.

During the break-in runs, blower air velocity and stop cycles are regulated to maintain a drum temperature of 125 to 150 F at the start of each cycle, so as to avoid damage to the linings or drum. Upon completion of break-in, the shoes are removed and measurements are taken of shoe and lining thickness at 6 to 10 points on each shoe, in accordance with Fig. 1 in the code. After inspection, the brake is reassembled, the shoes adjusted, and the clearance measured and recorded.

The various remaining phases of the test may be summarized as to speeds required:

C. First effectiveness test – 30, 50, and 70 mph, or 30, 50, and 60 mph for heavy-duty brakes, with 10-mph increments optional.

D. First fade and recovery test – 10 50-mph fade stops at 15 ft per sec per sec, at 1-min intervals; 10 30-mph recovery stops at 15 ft per sec per sec, at 1-min intervals.

E. Second fade and recovery test – 5 70-mph fade stops at 15 ft per sec per sec, at 1-min intervals; 10 30-mph recovery stops at 15 ft per sec per sec, at 1-min intervals.

F. Second effectiveness test – 30, 50, and 70 mph, or 30, 50, and 60 mph for heavy-duty brakes, with 10-mph increments optional.

G. Lining-wear test – 500 30-mph stops at 11 ft per sec per sec, and cycle starting temperature of 200 F.

H. Lining-wear test – 300 40-mph stops at 11 ft per sec per sec, and cycle starting temperature of 300 F.

I. Third effectiveness test – 30, 50, and 70 mph, or 30, 50, and 60 mph for heavy-duty brakes, with 10-mph increments optional.

J. Third fade and recovery test – 10 50-mph fade stops at 15 ft per sec per sec, at 1-min intervals; 10 30-mph recovery stops at 15 ft per sec per sec, at 1-min intervals.

K. Fourth fade and recovery test – 5 70-mph fade stops at 15 ft per sec per sec, at 1-min intervals; 10 30-mph recovery stops at 15 ft per sec per sec, at 1-min intervals.

L. Lining-wear tests – 100 60-mph stops at 11 ft per sec per sec, and cycle starting temperature of 350 F.

Explanatory notes and discussion of each of the foregoing phases of the complete dynamometer brake test are given in the following paragraphs.

C. *First Effectiveness Test* – After the initial lining break-in schedule is completed it is desirable to determine the effectiveness of the brake when new so as to establish a base line for comparison after the fade and recovery and the lining-wear tests. Speeds of 30, 50, and 70 mph, with 10-mph increments optional, are specified for passenger cars. Thus, if a certain organization conducts brake road tests at intermediate speeds, such as 40 and 60 mph, they can correlate their road results with dynamometer tests run at these intermediate speeds. For heavy-duty brakes, because of the lower operating speeds usually encountered, the 70-mph effectiveness test was replaced by one at 60 mph.

Since the effectiveness and line pressure requirements of the various types of brakes vary widely, the line pressures are specified to be increased in increments of 100 psi or multiples thereof, selected so as to provide a minimum of 5 points on the curve through the operating range.

During all tests throughout this and the succeeding phases, the blower is regulated to provide an air velocity of 2200 fpm (25 mph) at the brake, distributed uniformly and continuously over the brake projected area during the test, using air at room temperature. On the first effectiveness test, the brake drum is allowed to cool to 125 or 150 deg between successive stops, with the flywheel rotating, in order to avoid any undue lining fade.

D. *First Fade and Recovery Test (50 mph)* – After the completion of the first effectiveness test, the brake is then subjected to the first fade and recovery test. Prior to the start of the actual fade stops, line pressures are established which provide

a deceleration of 15 ft per sec per sec at both 30 and 50 mph, with drums cooled to 125 F between trial stops. Ten successive 50-mph fade stops are then made, using the established line pressure, at intervals of 1 min, with 15 sec application time and the flywheel operating at 50 mph between stops. The blower is operated continuously during this test, using an air velocity of 2200 fps (25 mph). After the 10 fade stops, the brake is allowed to cool for 3 min with the flywheel operating at 30 mph, after which 10 successive 30-mph recovery stops are made at 1-min intervals, using the previously established line pressure. In the event that the lining has not completely recovered at the end of 10 stops, the test is continued to 15 stops. Drums and linings are inspected at the completion of the test.

E. *Second Fade and Recovery Test (70 mph)* – In this more severe fade test the same procedure is used as in the preceding test, except that only 5 stops are made from 70 mph, at 1-min intervals, with 15 sec application time.

Recovering stops at 30 mph are then made at 1-min intervals in the same manner as in test D.

For heavy-duty brakes the fade tests are run at 60 instead of 70 mph.

MASTER SHEET NO. 1

SAE BRAKE TEST—DYNAMOMETER

Test No. 1234
Date 1/8/53

SUBJECT

ABC 11" x 2" x 3/16" Type "VA" Brake

OBJECTIVE

Complete Dynamometer Test

SPECIAL INSTRUCTIONS

VEHICLE

Manufacturer ABC Model CBA Tire Size 7.60 x 15

	Wt Static	Wt Dynamic	Test Load
Total	lb	lb	lb
Front	lb	lb	1219 lb
Rear	lb	lb	lb

(r) Rolling Radius 13.7 in. at 30 mph
BRAKE x Front Rear

Size 11" x 2" x 3/16" Type VA Dwg No. FF 3730
Anchor Location 72 % of Radius 8 Degrees from Centerline
Shoes 2 Pc. Steel Dwg No. FD 8301 & 4
Drum Centrifuse Mfr XYZ Dwg No. 33036

WEIGHTS

Complete Assembly 9 1/4 lb
Forward Shoe and Lining 1 3/4 lb
Reverse Shoe and Lining 1 3/4 lb
Backing Plate Complete 5 3/4 lb
Drum 9 1/4 lb

LINING

Forward		Shoe	Reverse	
DEF		Mfr	DEF	
S-100		Code	S-100	
FC 10027		Dwg No.	FC 10030	
12.48 in. 130	deg	Length	9.6 in. 100	deg
33	deg	Toe Angle	33	deg
17	deg	Heel Angle	17	deg
2	in.	Width	2	in.
3/16	in.	Thickness	3/16	in.
24.93	sq in.	Area	19.2	sq in.
44.15	sq in.	Total Area		

Method of Affixing Rivet Grind Dia 10.990 in.

ACTUATION

Hydraulic
Cylinder Dia 1"
Dwg No. FD 9457
Material Cast Iron
Piston Material Aluminum
Cup Dwg No. FC - 1199
Mfr
Compound
Fluid ARC 21-11

Electric
Voltage
Max Amp
Dwg No.

Air
Cylinder Dia
Dwg No.
Slack Adj Length
Stroke
Cam

Fig 6 – First master sheet for brake test

SAE BRAKE TEST—DYNAMOMETER

MASTER SHEET NO. 2

Test No. 1234
Date 1/8/53

DYNAMOMETER

Dynamometer Wheel Test Load 1219 lb 37.9 (W) slugs (lb/32.2)
Flywheel Moment Inertia 49.4 (I) slug ft² ($WR^2/144$)
Discs Used 1/2" Master Disc + Two 1/2" Discs + Three 1/2" Kidney Weights
Torque Arm 12 in.
Torque Cylinder Diameter 2.015 in. Torque Constant 3.19

Miles Per Hour	20	30	40	50	60	70
Rpm.....(N).....	244	366	489	610	733	855
* K E/Stop—lb-ft.....	16,150	35,900	64,300	99,600	144,500	196,000

* K.E.= 1/2 I W^2 = .00548 I N^2 Lb.Ft.

Fig. 7 – Second master sheet for brake test

SAE BRAKE TEST—DYNAMOMETER

LINING- AND DRUM-WEAR DATA Complete Test Schedule Test No. 1234
Test Laboratory ABC Date 1/8/53
Brake 11" x 2" x 3/16" Type "VA" Lining DEF S-100
Drum Centrifuse XYZ #33036

Forward Shoe

	DRUM SIDE					SHIELD SIDE				
Check Point	Toe		Center		Heel	Toe		Center		Heel
After Test B, break-in.....	.3000		.3000		.3000	.3000		.3000		.3000
After Tests C, D, E, and F....	.2996		.2992		.2994	.2992		.2994		.2994
After Test G, 30-mph wear.....	.2986		.2977		.2984	.2976		.2983		.2984
After Test H.....	.2984		.2963		.2984	.2967		.2975		.2984
After Tests I, J, and K.....	.2980		.2955		.2978	.2959		.2969		.2978
After Test L, 60-mph wear.....	.2895		.2708		.2844	.2841		.2820		.2905
Total.....	.0105		.0292		.0156	.0159		.0180		.0095

Reverse Shoe

	DRUM SIDE					SHIELD SIDE				
Check Point	Toe		Center		Heel	Toe		Center		Heel
After Test B, break-in.....	.3000		.3000		.3000	.3000		.3000		.3000
After Tests C, D, E, and F....	.2998		.2996		.2997	.2995		.2998		.2997
After Test G, 30-mph wear.....	.2958		.2982		.2996	.2993		.2994		.2996
After Test H.....	.2948		.2966		.2991	.2980		.2965		.2992
After Tests I, J, and K.....	.2945		.2961		.2989	.2984		.2994		.2989
After Test L, 60-mph wear.....	.2885		.2875		.2975	.2891		.2909		.2988
Total.....	.0115		.0125		.0025	.0109		.0091		.0012

Lining Condition Smooth - No Evidence of Surface Cracks

Drum Wear and Condition Profilometer Before Tests: 16-21; After Test: 22-40.
Heat Discoloration.

Fig. 8 – Lining-wear and drum-wear data sheet

F. *Second Effectiveness Test* – At the completion of the 50 and 70-mph fade and recovery tests, it is desirable to determine the influence of these tests on brake effectiveness. Accordingly, effectiveness tests are then conducted, using the same procedure as on the first effectiveness test.

G. *Lining-Wear Test – Glazing (30 mph)* – Prior to the start of this first wear test, the drum is removed and lining thickness at the exposed edges of the shoes is measured and recorded. Five hundred 30-mph stops at 11 ft per sec per sec are then made, with drums cooled to 200 F before each stop is made.

The torque balancing mechanism which was described earlier in the paper is used to maintain required deceleration throughout the 500 stops. If such equipment is not available, line pressures are manually adjusted periodically in order to maintain the 11 ft per sec per sec deceleration. On each tenth stop a line pressure graph is obtained and drum temperature is observed and recorded.

H. *Lining-Wear Test (40 mph)* – This is a more severe wear test than the preceding one, as 300 40-mph stops are made at a deceleration of 11 ft per sec per sec, with a cycle starting temperature of 300 F. The same procedure is used as in test G. Lining thickness at the exposed edges of the shoes is measured and recorded at the beginning and end of this wear test.

I. *Third Effectiveness Test* – After the first two wear tests it is desirable to determine the effect of these tests on brake output. Hence, effectiveness tests are tnen conducted, following the same procedure as on the first effectiveness test. At the completion of this test, brake linings and drum surfaces are inspected.

J. *Third Fade and Recovery Test (50 mph)* – Following the conditioning of drum and linings by the preceding wear tests, it is then of interest to repeat the fade and recovery tests. Accordingly, the 50-mph fade and 30-mph recovery test is repeated, using the same procedure as in test D.

K. *Fourth Fade and Recovery Test (70 mph)* – The 70-mph fade and 30-mph recovery tests described in test E are then repeated. Brake linings and drum surface are inspected and observations recorded at the completion of this test.

L. *Lining-Wear Test (60 mph)* – Since some brakes exhibit appreciable wear only at higher speeds and temperatures, the dynamometer brake test code includes as a final test a wear evaluation at 60 mph. The same procedure is followed as in the 30-mph lining-wear test, except that 100 stops are made from 60 mph at 11 ft per sec per sec, using a cycle starting temperature of 350 F.

M. *Lining-Wear Evaluation* – The dynamometer test of the brake has now been completed. The shoes are removed from the assembly, and shoes and linings are measured at the designated points which were established prior to the start of the tests. These measurements are then subtracted from initial measurements to obtain total lining wear.

N. *Drum Evaluation* – The drum is inspected and calipered at the completion of the test to determine wear and any other irregularities, such as scoring, heat-checking, and so forth, which have occurred during the test. Drum run-out is measured prior to removal from the dynamometer.

Data Sheets

The first master sheet provides space for recording all pertinent information required to describe the test brake, together with the vehicle on which the brake is used. Figures in the weight dynamic column take into account the weight shift onto the front wheels during braking. This is, of course, dependent on position of center of gravity of vehicle, weight distribution, and vehicle speed. A typical filled-in master sheet is shown in Fig. 6.

The second master sheet provides space for recording dynamometer constants, as well as calculated inertia, kinetic energy, and dynamometer speeds for various phases of the test. A filled-in sheet is shown in Fig. 7. Flywheel moment of inertia (I) is calculated using the formula:

$$I = Wr^{-2}/144$$

where:

I = Flywheel moment of inertia, slug ft^2
W = Dynamometer wheel test load, slugs
r = Rolling radius, in.

Kinetic energy is calculated from the formula:

$$KE = \tfrac{1}{2} I\omega^2 = 0.00548\, I\, N^2$$

where:

KE = Kinetic energy, ft lb
N = Flywheel rpm

The lining-wear and drum-wear data sheet provides space for recording measurements of drum and linings before and after various phases of the complete dynamometer test. A typical filled-in sheet is shown in Fig. 8.

Curve Sheets

The first curve sheet shows a plot of brake input versus average torque output for the effectiveness tests. One curve is plotted for each effectiveness test. A typical curve sheet for this plot is shown in Fig. 9.

The second curve sheet (Fig. 10) shows a plot of brake input versus deceleration for the effectiveness tests. Here again one curve sheet is used for each effectiveness test. This of course is another method of representing brake effectiveness, and is plotted from the same data taken in the effectiveness tests shown graphically on the first curve sheet.

The third curve sheet (Fig. 11) provides for plotting torque characteristic curves as obtained on the various effectiveness tests. Such curves are often useful in comparing the changes which take place in the brake and its components during the course of the complete dynamometer test.

The fourth curve sheet (Fig. 12) shows a plot of

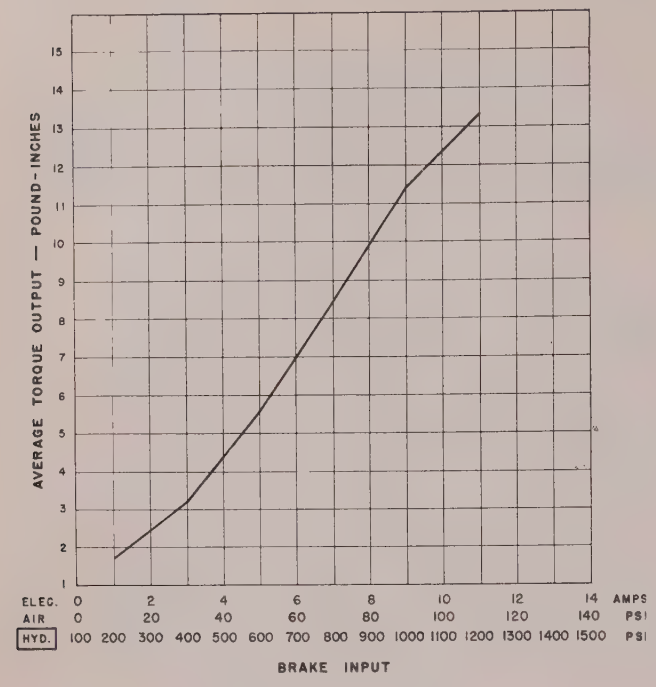


Fig. 9 – Torque output versus brake input recorded in first effectiveness test (30 mph). (See Fig. 6 for data on brake type, drum, and lining)

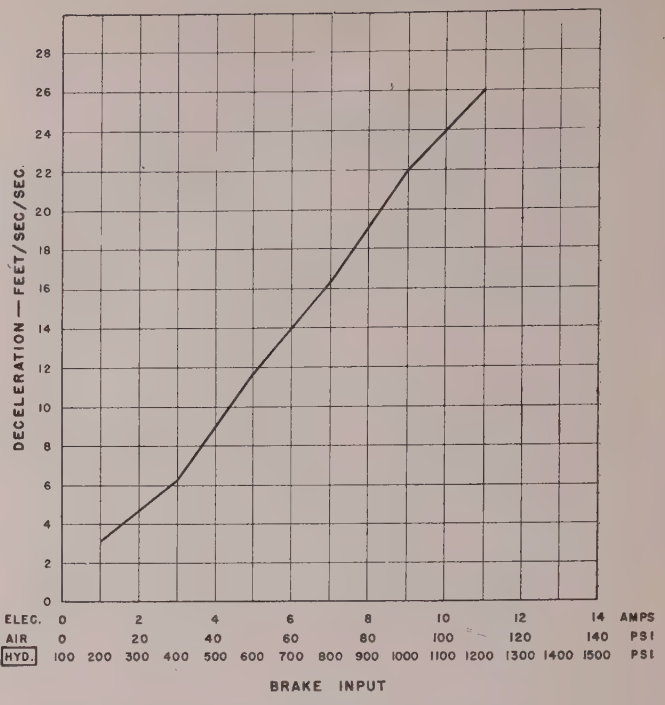


Fig. 10 – Deceleration versus brake input recorded in first effectiveness test (30 mph). (Conditions same as in Fig. 6)

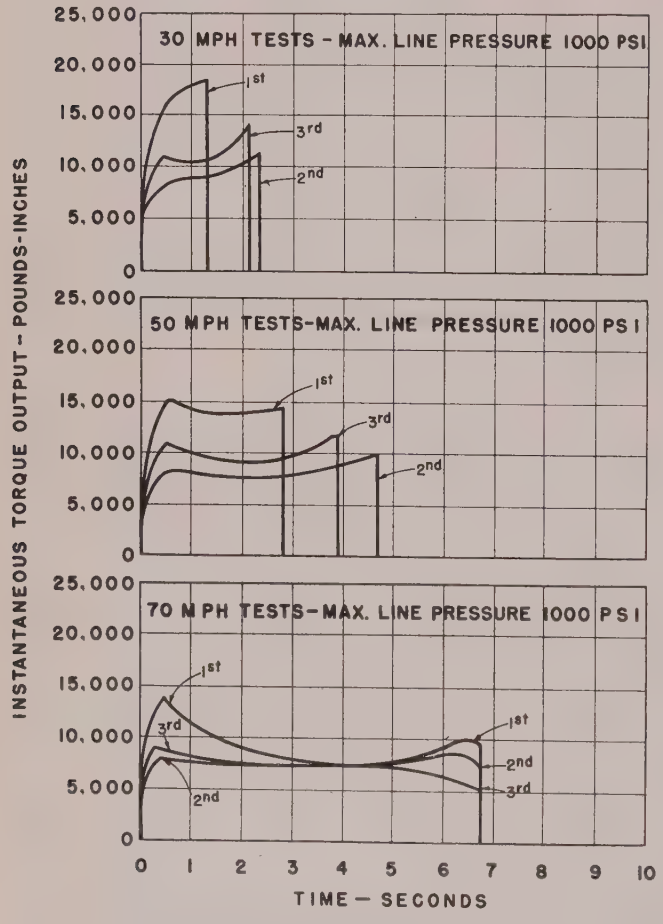


Fig. 11 – Torque characteristic curves in first, second, and third effectiveness tests. (Conditions same as in Fig. 6)

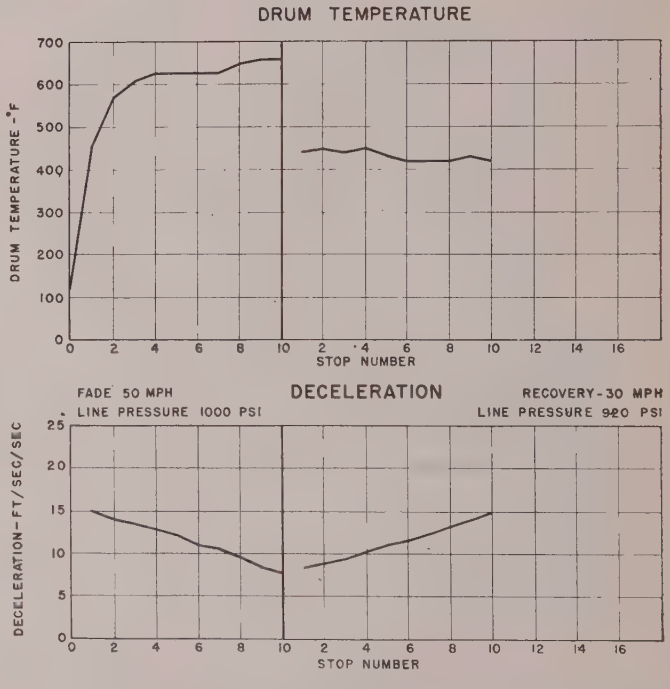


Fig. 12 – Deceleration and drum temperature versus stop number in third (50 mph) fade and (30 mph) recovery test. (Conditions same as in Fig. 6)

deceleration and drum temperature versus stop number for the various fade and recovery tests. One curve sheet is used for each fade and recovery test.

Correlation of Dynamometer and Road Tests

The brake testing dynamometer is the most effective and economical means to evaluate or screen brake assemblies, components, and friction materials. Brake dynamometer tests are conducted under controlled conditions of temperature, speed, load, and on some dynamometers, humidity. The brake dynamometer is completely instrumented to determine many of the required performance characteristics of the brake assembly and components.

The brake test code includes tests to indicate effectiveness, fade and recovery, lining and drum wear, together with other measurements or observations, such as distortion, noise, and so forth. Properly conducted road tests, of course, while made under less favorably controlled conditions, also provide information on the foregoing brake performance characteristics.

Due to conditions which will be discussed subsequently, considerable judgment and experience are required to predict from dynamometer test results the actual vehicle performance involving two or more brakes.

The dynamometer permits special studies with instrumentation which would be very difficult to apply to a vehicle, and also makes possible visual studies of brakes and components in operation, which would be impracticable to do on a vehicle.

In general, the various uses of a brake dynamometer may be listed as follows:

1. General brake evaluation.
2. Lining and attachment evaluation.
3. Drum evaluation.
4. Special studies:
    A. Deflection and distortion.
    B. Chatter.
    C. Noise.
    D. Temperature gradients and cooling.
    E. Automatic adjusters.

In the following paragraphs the function of the brake dynamometer to supplement road tests during brake development will be discussed in more detail for each of the foregoing uses.

1. *General Brake Evaluation* – By comparing the results of road tests and dynamometer tests on a given brake, a considerable correlation can be observed. Many deficiencies in the operation of the brake which are revealed on road tests will also be duplicated on dynamometer tests.

Thus, if an experimental brake design exhibits such shortcomings as lack of effectiveness, poor fade characteristics, rapid lining wear, or overrecovery after fade, these items can be studied individually on the dynamometer. Many modifications can be accomplished and tested on the dynamometer in the time required to test one such change on a vehicle. Since values are relative, improvement in one of these characteristics on the dynamometer will usually be realized on the vehicle.

On the dynamometer the brake torque can be accurately measured by direct-reading instruments. In a vehicle such measurements would be extremely difficult, and the brake torque must usually be calculated from other factors such as load on the wheel, deceleration, and initial speed and stopping distance. At wheel slide it is dependent on the coefficients of adhesion between the tire and road, at best an extremely variable factor.

Also, a brake can be made to develop sufficient torque on the dynamometer to cause a failure; this is often impossible on a vehicle, where the torque is limited to wheel slide. On the other hand, it is sometimes possible to obtain shock loads on the vehicle, such as produced by wheel hop, which are impossible to duplicate on the dynamometer.

A standardized brake test code makes it possible to compare under duplicate test conditions the performance of a new brake design versus an existing design, and to correlate those results with tests conducted on the same brake on other dynamometers.

Since dynamometers are capable of testing only one brake at a time, it is impossible to determine the distribution of front and rear braking on the dynamometer. In a vehicle the balance between front and rear braking would be affected by the load, its distribution on the axles, wheel base, the height of the center of gravity, and the load transfer caused by deceleration. The final brake distribution between front and rear is usually a compromise to satisfy the above factors, and can only be arrived at by testing on a vehicle. Many vehicles today employ different designs of brakes on the front and rear axles.

2. *Lining and Attachment Evaluation* – Research and development of brake linings continues to be a never-ending process, since brake engineers are demanding more and more from these materials. Consequently, the evaluation of various brake-lining materials utilizes a large percentage of the available test time on many brake-testing dynamometers. In the rather short time required to conduct a dynamometer test according to the brake test code, much information can be gained on the various characteristics of the lining being investigated. Hence, it is possible to screen many linings and to select the most likely for subsequent vehicle road tests without using the time-consuming procedure of testing each lining on the vehicle.

Dynamometer controls make it possible to conduct wear tests of a desired cycle and duration to determine wear patterns under controlled conditions of load and pressure application. This gives comparative results between brake structures and friction materials. It is impracticable, however, on

a brake-testing dynamometer to simulate all conditions of cooling, moisture, dirt, and other foreign materials which prevail on the vehicle. Consequently it is difficult to obtain absolute correlation of lining and drum wear obtained on a vehicle and on a dynamometer.

By means of special severe tests the brake dynamometer also is a valuable tool in developing brake-lining bonding cements or rivets. Cemented or riveted shoes and brake linings can be aged on an accelerated basis in an oven and then subjected to a dynamometer pull-off test, which might prove hazardous on a vehicle. Various cements and rivets can thus be screened on the dynamometer, and the most likely selected for final vehicle tests.

3. *Drum Evaluation* – Brake drums are also under continuous development in order to keep pace with the demands of modern braking systems. Brake drums must be tailored to fit the particular type of brake and brake lining used, and it is often necessary to screen many drum sections in order to arrive at the optimum design.

Here again the brake dynamometer permits the evaluation of many drums, tested under identical conditions, so that the best designs can be designated for subsequent vehicle tests. Such road test drum problems as scoring, spotting, heat-checking, permanent distortion, cracking, and so forth, are very often reproduced on dynamometer tests.

4. *Special Studies* – A. *Deflection and Distortion* – Because of the relatively large loads imposed on modern brakes, deflection or distortion of the various parts is inevitable. These deflections or distortions often are the source of erratic and other undesirable brake action on the vehicle. The brake dynamometer provides a means for studying deflection or distortion of various parts of the brake mechanism, visually and with instrumentation which would be most difficult to apply to the vehicle. As a result of such studies, improvements can be developed by testing the brake on the dynamometer, and then proof testing on the vehicle.

B. *Chatter* – Chatter is one of the most elusive of the many brake ills which confront the engineer. Often, studies conducted on the brake dynamometer can provide clues which lead to important improvements to reduce chatter. These are then checked by subsequent road tests.

The conventional brake mounting on the dynamometer does not include many of the other vehicle components which contribute to brake chatter on the vehicle, so that chatter obtained during road tests may not be reproduced on the dynamometer.

C. *Noise* – Brake noises are often experienced on a dynamometer, but frequently they are of a different character than are experienced on a vehicle; this is probably accounted for by the variations in mounting between the dynamometer and the vehicle.

In some instances, however, the brake dynamometer has proved useful in noise investigations when used in conjunction with instrumentation which indicates the characteristics of vibration of the various components of the brake assembly. By such studies it is sometimes possible to isolate one particular part, which by proper treatment can be damped or redesigned so as to eliminate the noise. Road tests, of course, must be the final basis for acceptance of a correction.

D. *Temperature Gradients and Cooling* – The increasing demands of horsepower absorption of modern braking systems, as brought about by higher speeds, styling trends, and diminished help from the engine on grades, have spotlighted the need for increased attention to brake component temperatures, and to brake cooling in general.

The brake dynamometer provides a useful tool for instrumented study of brake temperature gradients throughout the mechanism, as well as the effectiveness of modifications to reduce temperatures. Dynamometer tests, of course, must be supplemented by vehicle tests to give the final answer on such modifications.

Although the cooling of one brake versus another on dynamometer tests can be accurately controlled, it is extremely difficult to simulate the exact cooling of the front or the rear brakes on a vehicle, where this cooling is greatly influenced by the type of shrouding around the wheels. This also affects the brake effort distribution between the front and rear brakes, and can only be evaluated on the vehicle.

E. *Automatic Adjusters* – The operating characteristics of automatic brake adjusters can often be studied to advantage on the brake dynamometer by using the wear cycles of the dynamometer test code, whereby the effectiveness of the adjuster in maintaining the shoe-to-drum clearance can be evaluated for various increments of lining wear. The dynamometer also permits a study of the resistance to wear of the drum-sensing member of certain types of automatic adjusters.

In summary, there is a degree of correlation possible between dynamometer and vehicle testing which must be arrived at by calculation. Such data as brake input, brake output, drum speed, and brake fluid requirements, which are obtained on one brake on the dynamometer, must be applied to four brakes on the vehicle. To do this, the weight and its distribution on the axles must be known. Weight transfer due to deceleration must be calculated or estimated. By using the rolling radius of the tires, brake torque may be translated into deceleration. Pedal travel may be calculated from dynamometer fluid requirement data by taking into account the master-cylinder displacement, the pedal ratio, and allowances for system expansion.

The brake dynamometer is a most valuable piece of equipment for many different varieties of brake investigations, and provides the only logical approach to the selection of a newly-designed braking combination, subject to verification by road testing on the actual vehicle.

APPENDIX

Brake Test Code—Dynamometer

SAE Recommended Practice

Report of Brake Technical Committee approved April 1952.

Rules and Directions—Hydraulically Actuated Brakes

A complete standard dynamometer brake test includes the determination at various speeds and line pressures (see Procedure) of the following:

1. Torque output and deceleration (various speeds and brake input)
2. Evaluation of lining life (wear tests)
3. Brake characteristics (fade and recovery tests, noise, chatter, grabbiness)
4. Drum evaluation (wear, heat checking, scoring)

From these determinations the following graphs are plotted and charts compiled on standard forms:

1. Torque output versus brake input and deceleration versus brake input graphs (effectiveness tests) (Figs. 5 and 6)
2. Characteristic torque graphs from effectiveness tests at maximum line pressures (individual stops) (Fig. 7)
3. Lining fade and recovery graphs (deceleration and temperature versus stop number) (Fig. 8)
4. Lining-wear tabulation—wear per test phase and total test (Fig. 9)
5. Drum—statement of condition (and wear, if any) (Fig. 9)

Test Conditions—Actual tests for performance shall be started when preliminary preparations and lining "break-in" have been completed as given in sections on Preliminary Preparations and Initial Lining Break-In under Procedure.

Flywheel loading shall correspond with the work load imposed on the particular brake as it is to be used in the specific vehicle.

Instrumentation—Tests shall be run on an inertia-type machine capable of speed control within ± 2% of designated speeds (under Procedure) and flywheel loading within ± 5% of that designated under Test Conditions.

The machine shall be equipped with accurate means for determining torque output of the brake. These shall include both direct torque-recording instruments and devices for indicating the average flywheel deceleration which can be translated into torque output. Means shall be provided for accurate control and determination of input pressure (line pressure) to the brake wheel cylinder.

The machine shall also be equipped with a blower for control of drum temperature within the limits specified and a pyrometer which records drum temperature as described under Procedure. A direct-reading pyrometer may be used for observation of brake-lining temperature.

It is desirable that the machine shall be equipped with an accurate displacement meter for determination of input requirement of the brake.

It is also desirable that the machine be equipped with a torque-balancing device or mechanism to maintain specified deceleration rates during wear.

Procedure

A—Preliminary Preparations—Before the brake unit is as-

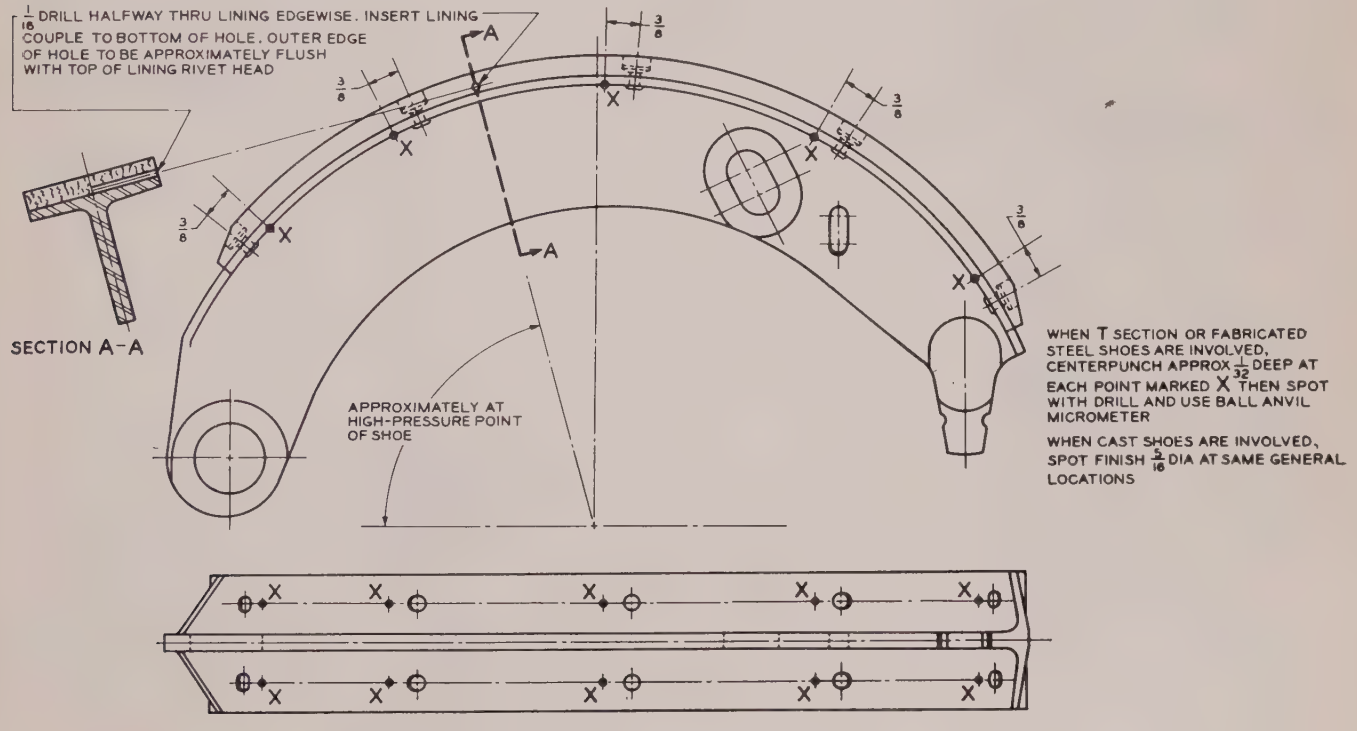


FIG. 1—MARKING OF REFERENCE POINTS FOR LINING-WEAR MEASUREMENTS

sembled to the dynamometer for test, it shall be corrected for any obvious defects or errors of omission and checked to make certain that the brake lining is free from grease, paint, or other foreign material. The shoes shall be removed from the assembly and spot finished or marked with a prick punch on the under side of table, in accordance with Fig. 1, for the purpose of making accurate measurements for lining wear. The brake lining may be drilled and thermocouple installed in accordance with Fig. 1. The brake drum shall also be washed thoroughly on the braking surface with acetone or equivalent solvent to remove any rust film. The drum must also be prepared for thermocouple installation in accordance with Fig. 2.

The foregoing preparatory work completed, the brake and drum are now mounted on the dynamometer and centered and runout recorded. (The drum may be calipered at open and closed end, the maximum and minimum diameters recorded and permanently marked.) The drum surface condition shall be observed and recorded (profilometer reading preferred). Proper pressure-line and thermocouple connections shall then be made to the instrument panel. The hydraulic system shall be thoroughly bled to free it of air, which completes preliminary preparation for tests.

All subsequent procedure presupposes previous calculations of flywheel load and speeds and recording of same on master forms No. 1 and No. 2 (Figs. 3 and 4) together with other pertinent data.

B—Initial Lining Break-In—Adjust brake shoes to drum clearance recommended by the manufacturer. Record clearance. Establish flywheel speed for 30 mph and line pressure to obtain a deceleration of approximately 11 fps per sec. Make a sufficient number of stops to obtain a minimum of 80% lining contact uniformly distributed radially. Regulate the blower air velocity and stop cycle to maintain a starting drum temperature of 125–150 F. After completion of break-in, remove shoes without disturbing adjustments and take initial lining measurements at points established for this purpose. Record results and general condition of lining. Reassemble shoes, adjust, and record clearance.

Note: Readjust to initial clearance when a maximum clearance of 0.030 in. for passenger cars and 0.050 in. for trucks is reached.

C—First Effectiveness Test—Run effectiveness tests at 30, 50, and 70 mph (10-mph increments optional) on passenger-car brakes, and 30, 50, and 60 mph (10-mph increments optional) on heavy-duty brakes. Make one stop at each speed and pressure. Use line pressures in increments of 100 psi or multiples thereof, using a minimum of five points through the range to determine the characteristics. Blower shall be regulated to provide air velocity of 2200 fpm (25 mph) at the brake, distributed uniformly and continuously over the projected area during test, using air at room temperature. With flywheel rotating, cool the brake drum to 125–150 F between successive stops. Inspect brake linings and drum surface, and record observations.

D—First Fade and Recovery Test (50 mph)—Establish line pressures to give a deceleration of 15 fps per sec at both 30 and 50 mph prior to start of fade test. Cool drums to 125 F between

(Text continued after Fig. 4.)

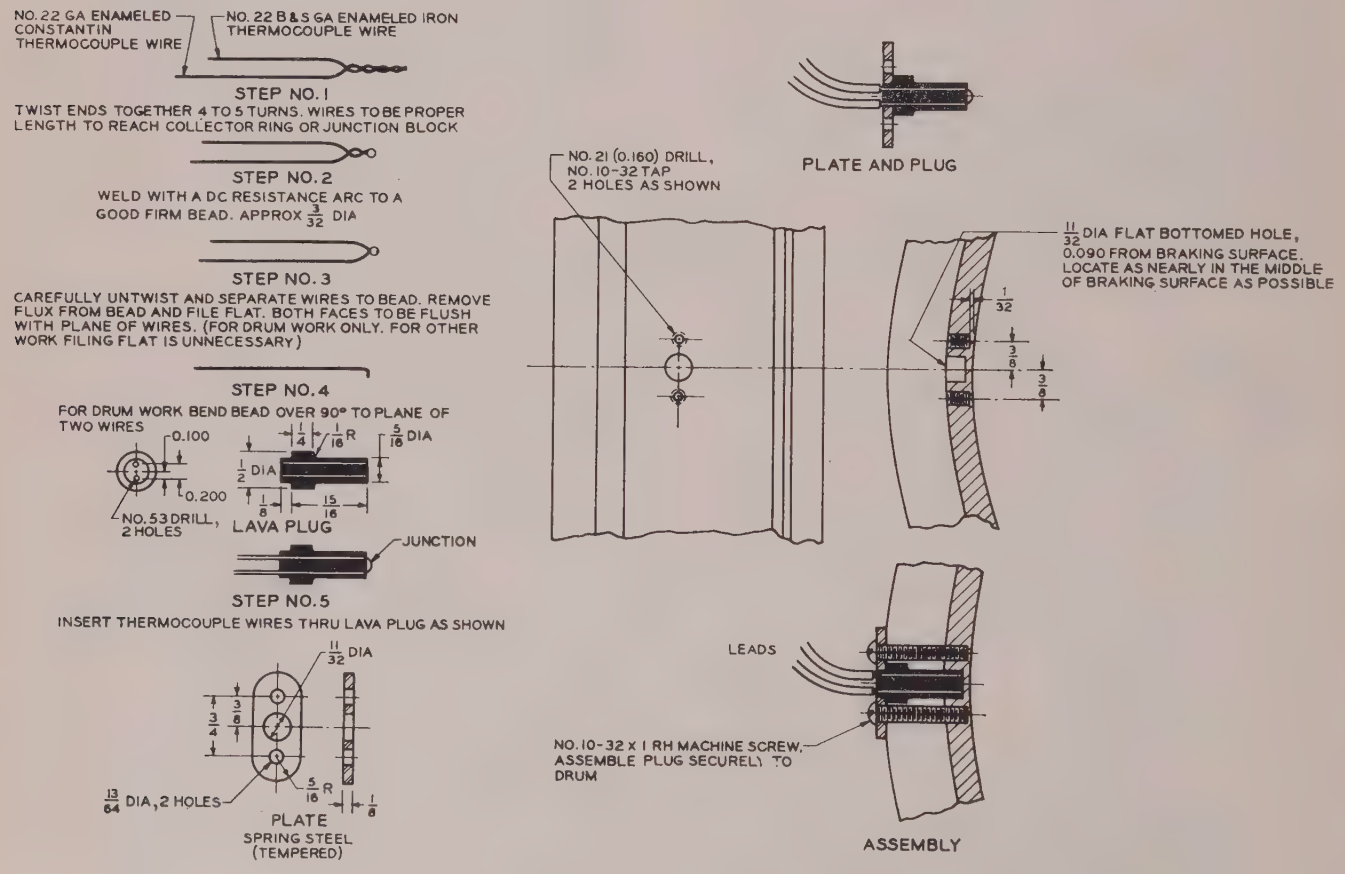


Fig. 2—Preparation of Drum for Thermocouple Installation

SAE BRAKE TEST—DYNAMOMETER

MASTER SHEET NO. 1

Test No._____

Date_____

SUBJECT

OBJECTIVE

SPECIAL INSTRUCTIONS

VEHICLE

Manufacturer_____ Model_____ Tire Size_____

	Wt Static	Wt Dynamic	Test Load
Total	lb	lb	lb
Front	lb	lb	lb
Rear	lb	lb	lb

Rolling Radius_____ in. at_____ mph

BRAKE Front Rear

Size_____ Type_____ Dwg No._____

Anchor Location_____ % of Radius_____ Degrees from Centerline_____

Shoes_____ Dwg No._____

Drum_____ Mfr_____ Dwg No._____

WEIGHTS

Complete Assembly_____ lb

_____ Shoe and Lining_____ lb

_____ Shoe and Lining_____ lb

Backing Plate Complete_____ lb

Drum_____ lb

LINING

		Shoe		
		Mfr		
		Code		
		Dwg No.		
in.	deg	Length	in.	deg
	deg	Toe Angle		deg
	deg	Heel Angle		deg
	in.	Width		in.
	in.	Thickness		in.
	sq in.	Area		sq in.
	sq in.	Total Area		

Method of Affixing_____ Grind Dia_____ in.

ACTUATION

Hydraulic

Cylinder Dia_____

Dwg No._____

Material_____

Piston Material_____

Cup Dwg No._____

Mfr_____

Compound_____

Fluid_____

Electric

Voltage_____

Max Amp_____

Dwg No._____

Air

Cylinder Dia_____

Dwg No._____

Slack Adj Length_____

Stroke_____

Cam_____

FIG. 3—MASTER SHEET NO. 1

SAE BRAKE TEST—DYNAMOMETER

MASTER SHEET NO. 2

Test No._____
Date_____

DYNAMOMETER

Dynamometer Wheel Test Load_____lb _____slugs (lb/32.2)
Flywheel Moment Inertia_____slug ft^2
Discs Used_____
Torque Arm_____in.
Torque Cylinder Diameter_____in. Torque Constant_____

Miles Per Hour	20	30	40	50	60	70
Rpm						
K E/Stop—lb-ft						

FIG. 4—MASTER SHEET NO. 2

trial stops. Make 10 successive 50-mph stops, using established line pressures, at intervals of 1 min, with 15-sec application time (time allowed for stop) and flywheel operating at 50 mph between stops. Record deceleration for each stop. Blower shall be regulated to provide air velocity of 2200 fpm (25 mph) at the brake, distributed uniformly and continuously over the projected area during test, using air at room temperature. Reset line pressure to 30-mph value (previously established) and, after 3-min cooling time with flywheel operating at 30-mph equivalent, make 10 successive 30-mph stops at 1-min intervals (flywheel operating at 30 mph between stops). If lining does not show definite recovery in 10 stops at 1-min intervals, continue to 15 stops. Inspect brake linings and drum surface, and record observations.

E—Second Fade and Recovery Test (70 mph)—Use same procedure as in Test D, establishing line pressure for both 30 and 70 mph. Make five stops only at 1-min intervals and 15-sec application time. Recovery is same as in Test D.

Note: In case of heavy-duty brakes, fade test should be run at 60 mph and recovery at 30 mph. Inspect brake linings and drum surface and record observations.

F—Second Effectiveness Test—Use same procedure as in Test C. Inspect brake lining and drum surface, and record observations.

G—Lining-Wear Test, Glazing (30 mph)—Measure and record lining thickness at exposed edge of shoe at beginning and end of test. Make 500 stops at 30 mph. Establish line pressure for a deceleration of approximately 11 fps per sec, and regulate torque-balancing mechanism to maintain this deceleration. Obtain line-pressure graph and record drum temperature every 10 stops. Use blower air velocity and stop cycle to maintain cycle starting temperature of 200 F.

H—Lining-Wear Test (40 mph)—Use procedure as in Test G, except that there shall be 300 stops at 40 mph and cycle starting temperature of 300 F. Measure and record lining thickness at exposed edges of shoes at beginning and end of test.

I—Third Effectiveness Test—Use procedure as in Test C. Inspect brake linings and drum surface, and record observations.

J—Third Fade and Recovery Test (50 mph)—Repeat fade and recovery test as in Test D.

K—Fourth Fade and Recovery Test (70 mph)—Repeat fade and recovery test as in Test E.

L—Lining-Wear Test (60 mph)—Use procedure as in Test G, except that there shall be 100 stops at 60 mph and cycle starting temperature of 350 F. Inspect and record observations.

M—Lining-Wear Evaluation—Complete test. Remove shoes from assembly. Measure linings at designated points. Subtract from initial measurements, and record total wear.

N—Drum Evaluation—Caliper drum to determine wear. Record together with statement of condition of braking surface.

Rules and Directions—Air Actuated Brakes

All rules and directions, as well as procedure, pertaining to hydraulically actuated brakes shall apply to air brakes.

Rules and Directions—Electrically Actuated Brakes

All rules and directions, as well as procedure, pertaining to hydraulically actuated brakes shall apply to electrically actuated brakes except that amperes are substituted for line pressure.

(*Figs. 5–9 begin on next page.*)

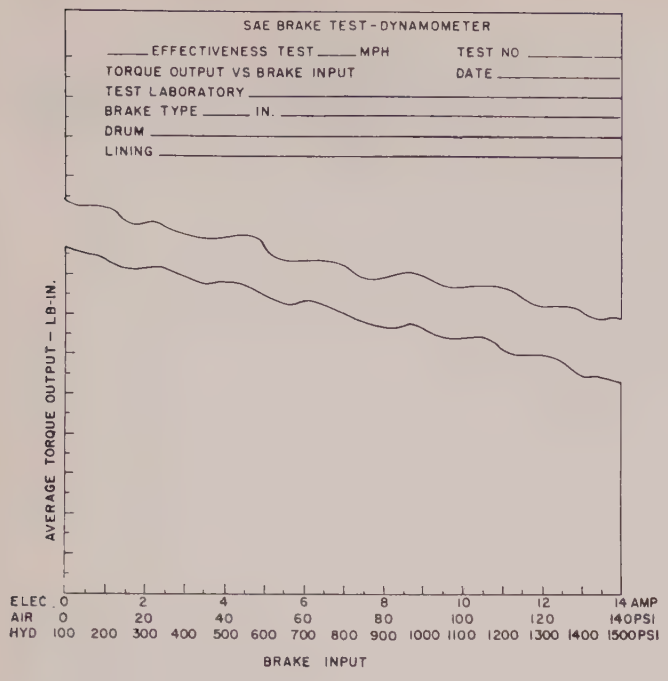


FIG. 5—TORQUE OUTPUT VERSUS BRAKE INPUT

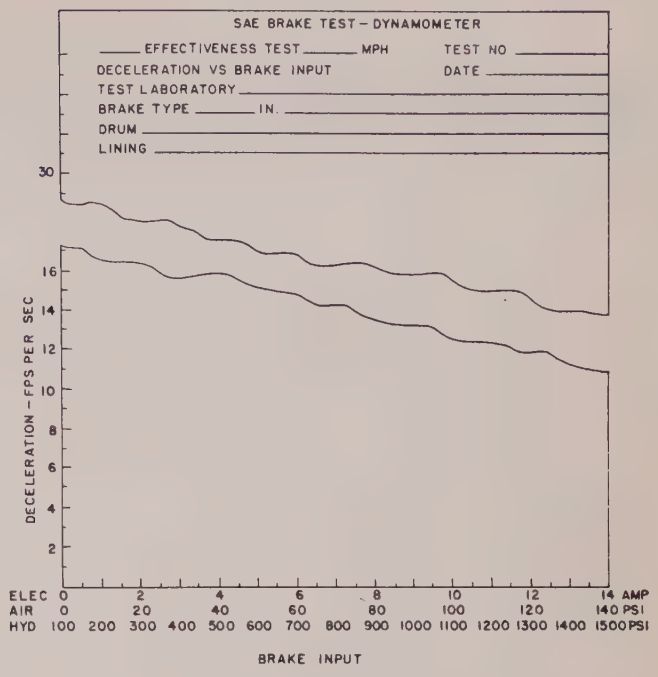


FIG. 6—DECELERATION VERSUS BRAKE INPUT

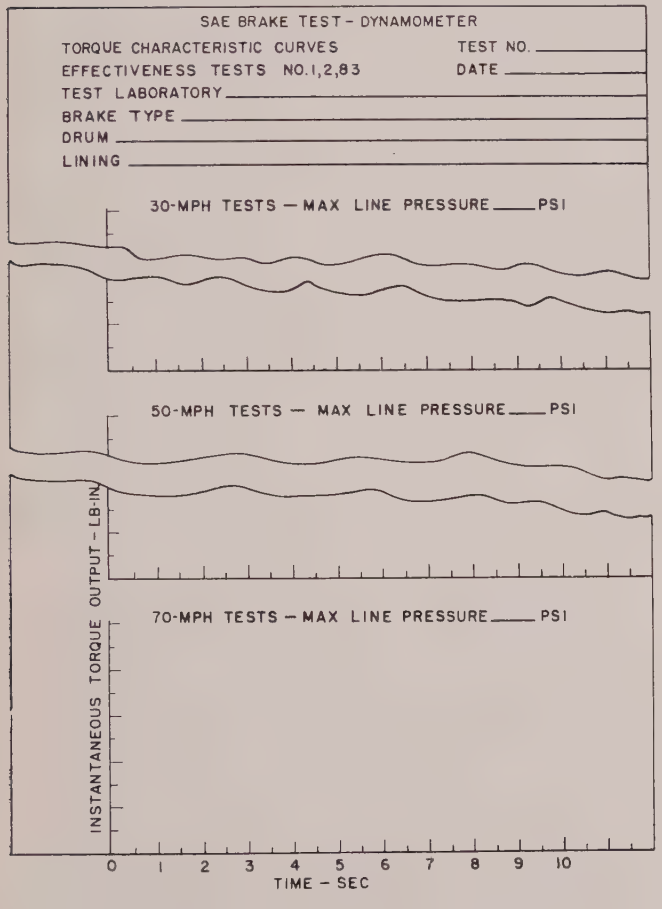


FIG. 7—TORQUE CHARACTERISTIC CURVES

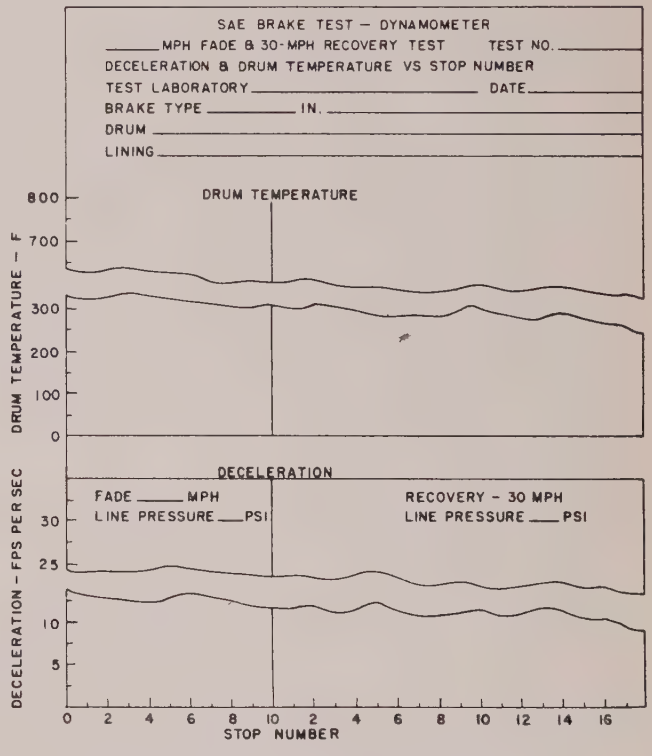


FIG. 8—DECELERATION AND DRUM TEMPERATURE VERSUS STOP NUMBER

(*Continued on next page.*)

SAE BRAKE TEST—DYNAMOMETER

LINING- AND DRUM-WEAR DATA _____ Test No. _____
Test Laboratory _____ Date _____
Brake _____ Lining _____
Drum _____

_____ Shoe

Check Point	DRUM SIDE					SHIELD SIDE				
	Toe		Center		Heel	Toe		Center		Heel
After Test B, break-in										
After Tests C, D, E, and F										
After Test G, 30-mph wear										
After Test H										
After Tests I, J, and K										
After Test L, 60-mph wear										
Total										

_____ Shoe

Check Point	DRUM SIDE					SHIELD SIDE				
	Toe		Center		Heel	Toe		Center		Heel
After Test B, break-in										
After Tests C, D, E, and F										
After Test G, 30-mph wear										
After Test H										
After Tests I, J, and K										
After Test L, 60-mph wear										
Total										

Lining Condition _____

Drum Wear and Condition _____

Fig. 9—Lining- and Drum-Wear Data

Coating Steel by the Aldip Process

D. K. Hanink and A. L. Boegehold,
Research Laboratories Division, GMC

This paper was presented at the SAE Annual Meeting, Detroit, Jan. 12, 1953.

THE potential advantages to be gained from coating ferrous articles with aluminum has been recognized for many years. One of the first patents on such a process was issued in 1893, only seven years after the Charles Hall discovery of a commercial method for producing aluminum. During the intervening years other processes were developed, most of which were based on precoating the steel with tin, zinc, lead, or alloys of these metals. All of these methods had the common disadvantage that the precoat metal contaminated the molten aluminum bath during extended periods of operation and thus prevented large-scale commercial adoption.

Other commercial methods for coating sheet steel have been developed but these methods do not readily lend themselves to coating prefabricated assemblies for various reasons. For example, in one process the aluminum is applied in a hydrogen atmosphere which presents handling problems in draining off excess aluminum.

This paper describes a process for coating fabricated assemblies without the disadvantages of the methods previously encountered. This patented process called Aldip (Patent No. 2,569,097) has been in commercial operation for two years and successful field experience has been achieved for applications at room and elevated temperature in various types of atmospheres.

The Aldip process can best be described by referring to the flow sheet in Fig. 1 outlining the various operations necessary to successfully clad steel parts or assemblies with aluminum.

Preliminary Cleaning – There are a number of alternate steps that can be taken in preliminary cleaning of steel and choice depends largely on type of material and surface conditions regarding grease and dirt, or rust and scale.

CLADDING of steel parts or assemblies with aluminum, through the use of a recently developed process which has been in commercial operation for two years, is described in this paper. Successful plant and field experience indicates that it can be applied at room and elevated temperatures in various types of atmospheres.

This new process was originally designed to offer corrosion resistance caused by the collection of leaded fuel gas condensates in exhaust systems, and for other parts requiring high-temperature oxidation resistance up to 1600 F. However, it has since been used in a wide variety of applications.

D. K. HANINK is senior metallurgist with the Research Laboratories Division, General Motors Corp. He received his B.S. degree in metallurgical engineering from the University of Michigan. Mr. Hanink joined GMC in 1945.

A. L. BOEGEHOLD (M '44) is assistant to the general manager of the Research Laboratories Division, General Motors Corp. He graduated from Cornell University, and prior to his present position he was head of the metallurgy department of the GMC Research Laboratories Division for over 25 years.

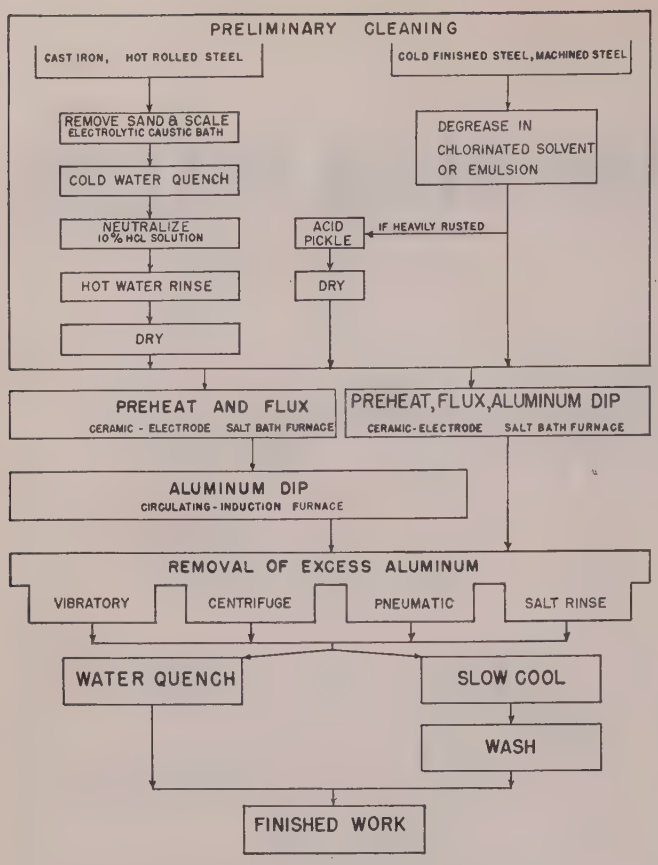


Fig. 1. – Flow sheet of aluminum coating process

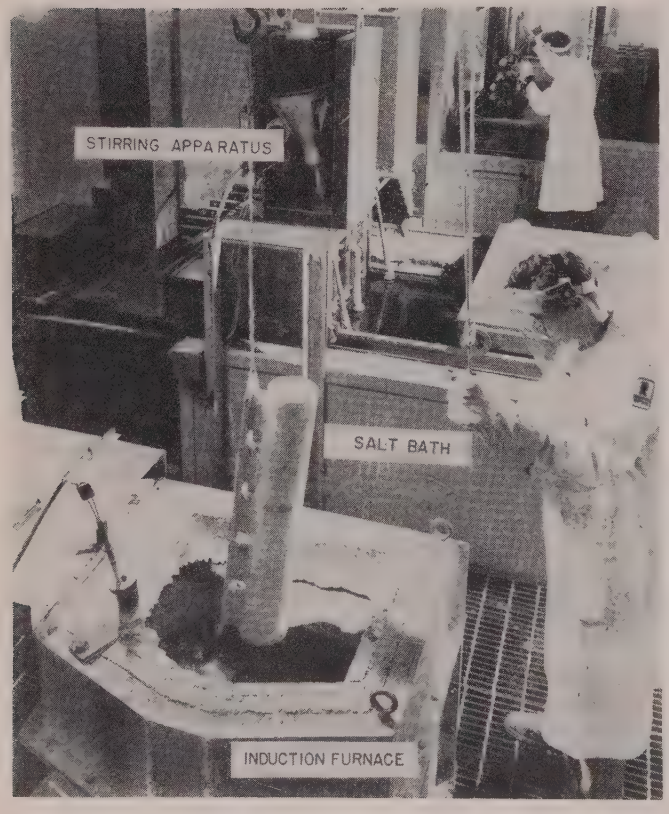


Fig. 2 – Aldip research installation at General Motors

Cast iron with typical carbon smear on the surface or hot-rolled steel with imbedded mill scale is cleaned in an electrolytic molten caustic salt at 900 F, followed by neutralizing in acid and water washing. Welded structures with very thick oxide or entrapped welding-rod flux can also be cleaned in this manner if the weld location is inaccessible for sand blast treatment or if such treatment would be harmful to the part.

Other fabricated parts which are relatively clean can be degreased and aluminum-coated without preliminary treatment. Heavily rusted steel must be acid-pickled, but superficial rust formation which may occur either before or after preliminary pickling will be dissolved in the preheating and fluxing bath.

Preheat and Flux – The molten salt bath for pre-

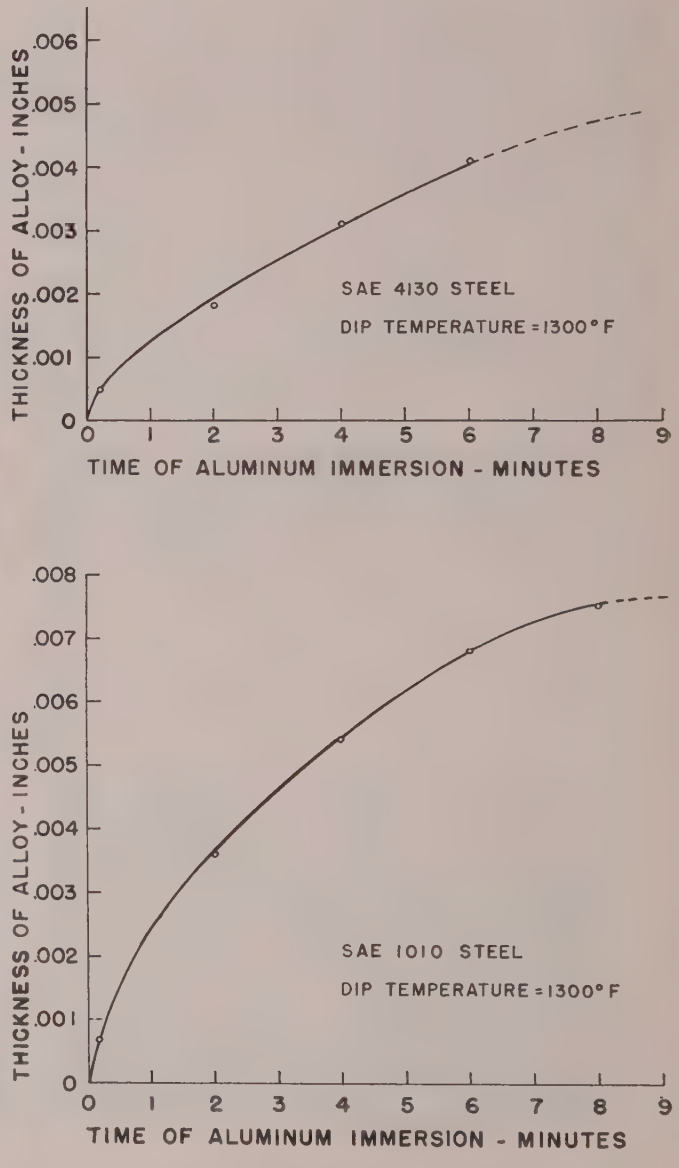


Fig. 3 – Aluminum-iron alloy formation

heating and fluxing steel is maintained between 1300 F and 1325 F in an electrode furnace lined with special ceramic brick and containing molten aluminum at the bottom of the pot. The photograph in Fig. 2 shows the salt bath furnace in the immediate background with stirring apparatus mounted on the furnace structure. This stirrer is operated intermittently to maintain temperature of molten aluminum by circulation of hot salt at the metal and salt interface.

Preheated and fluxed parts can be dipped in either the molten aluminum beneath the salt in this electrode furnace or transferred to a low-frequency induction furnace containing aluminum. This furnace is shown in the foreground of Fig. 2.

Aluminum Dip – When steel is fluxed and preheated to 1300 F in the special salt mixture, subsequent dipping in aluminum forms an instantaneous and continuous coating regardless of steel cross-sectional thickness. For parts which operate at temperatures exceeding 1000 F, close control of both time and temperature of the aluminum dip is necessary. This control of the aluminum dip and relationship to recommended post-diffusion heat-treatments are covered in another section of this paper.

Removal of excess aluminum coating on parts can be accomplished in a variety of ways, and the preferred combination is a salt rinse in the preheat flux followed by an air blast at selected locations on the part where metal may accumulate.

Engineering specifications designed to produce uniformity of the coated product require background data on the type of surface alloying obtained beneath the aluminum overlay, and its relationship with the type of alloy produced in subsequent heat-treatment to provide optimum high-temperature service. For this reason, data were obtained on:

1. Thickness of intermediate iron-aluminum compound for different dip times in aluminum.
2. Characteristics of the alloy layer produced by subsequent diffusion at different temperatures and times with respect to:
   a. Depth of alloy.
   b. Approximate aluminum content.
   c. Resistance to oxidation.

The charts in Fig. 3 show the thickness of iron-aluminum compound with increase in aluminum dip time. Depth of alloy can be controlled as closely as other metallurgical processes such as nitriding and carburizing. Standard metallographic techniques satisfactorily define the microstructure as shown in photomicrographs, Figs. 4 through 7. These show the typical microstructure through both aluminum coating and intermediate iron-aluminum compound layer on Aldip-coated SAE 1010 and 4130 steel.

The longer dip time of 2 min represents parts which are to be conditioned for service by diffusion heat-treatment after coating. The 15-sec dip is the

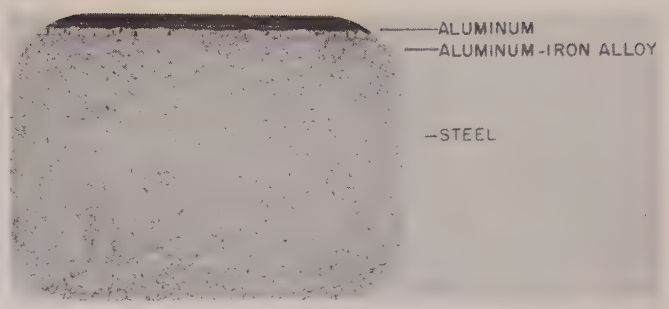


Fig. 4 – SAE 1010 steel, aluminum-coated. Dipping time 15 sec at 1300 F. Nital etch. Reduced from microphotographs taken at 100x

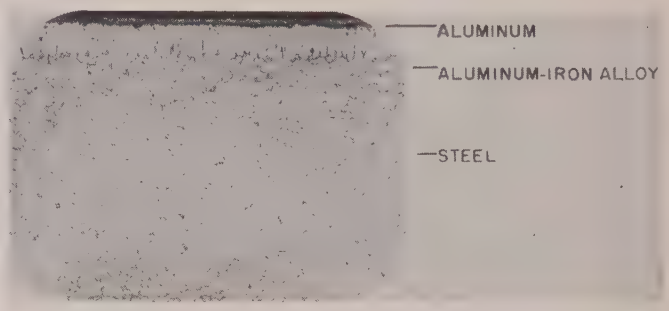


Fig. 5 – SAE 1010 steel aluminum-coated. Dipping time 2 min at 1300 F. Nital etch. Reduced from microphotographs taken at 100x

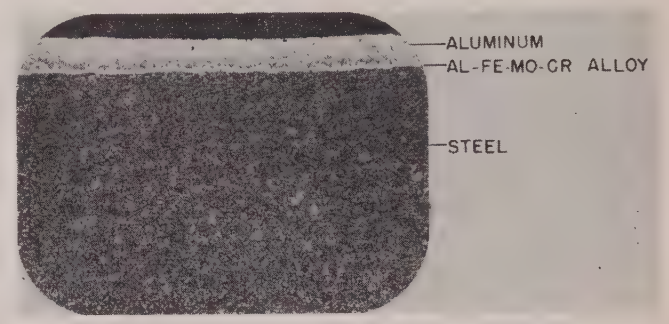


Fig. 6 – SAE 4130 steel aluminum-coated. Dipping time 15 sec at 1300 F. Nital etch. Reduced from microphotographs taken at 100x

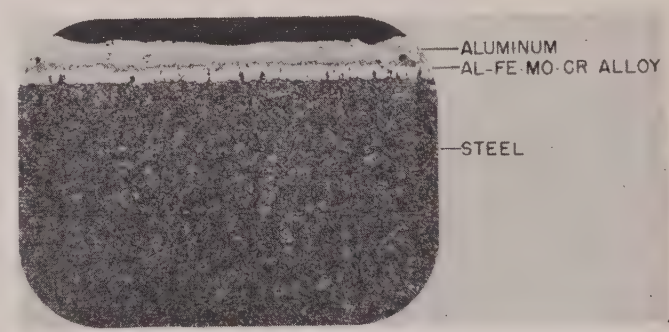


Fig. 7 – SAE 4130 steel, aluminum-coated. Dipping time 2 min at 1300 F. Nital etch. Reduced from microphotographs taken at 100x

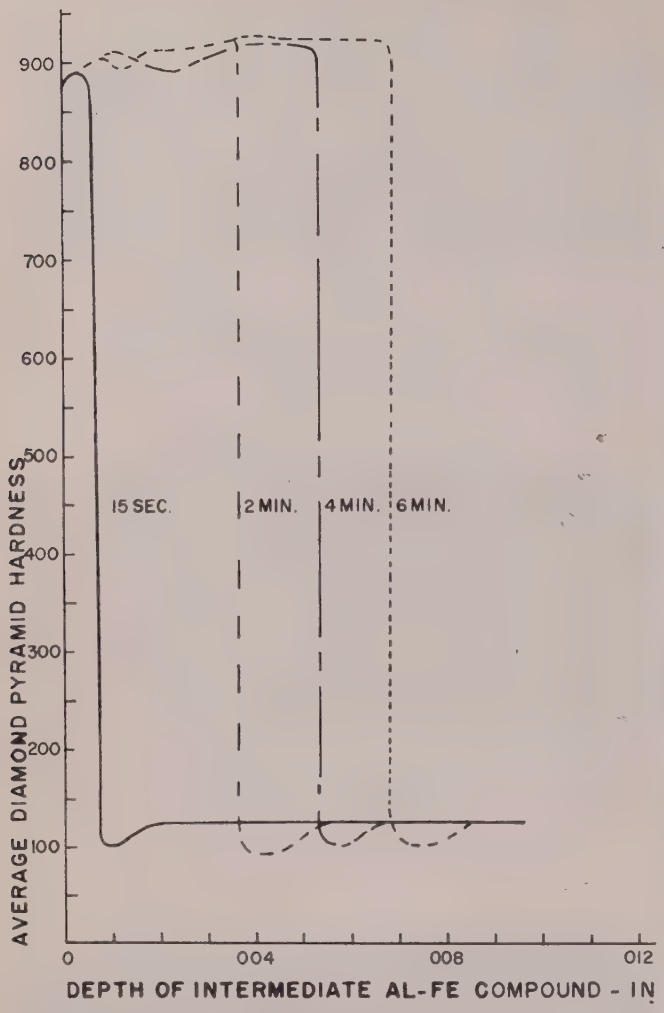


Fig. 8 – Microhardness traverse across Aldip aluminum-coated SAE 1010 steel at 1300 F

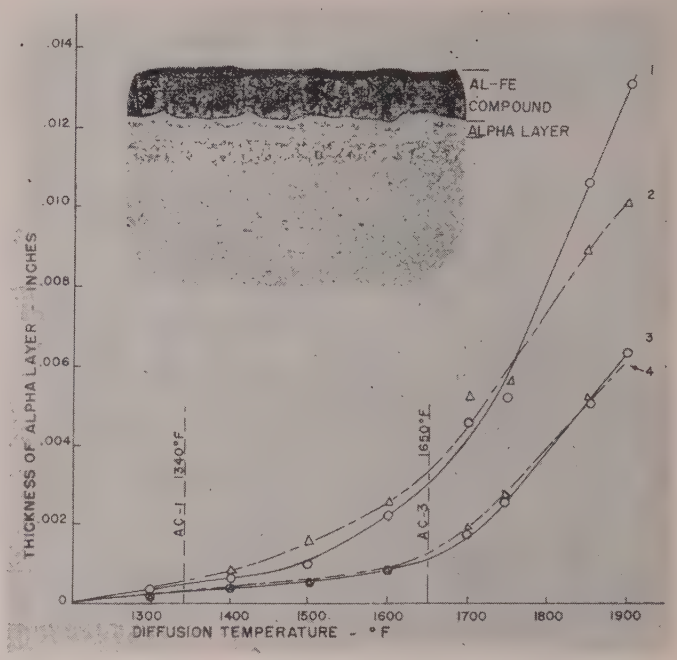


Fig. 9 – Formation of aluminum-iron alpha solid solution at different temperatures for Aldip-coated SAE 1010 steel

Curve	Dip Time at 1300 F, min	Diffusion Time, hr
1	8	4
2	2	4
3	2	1
4	8	1

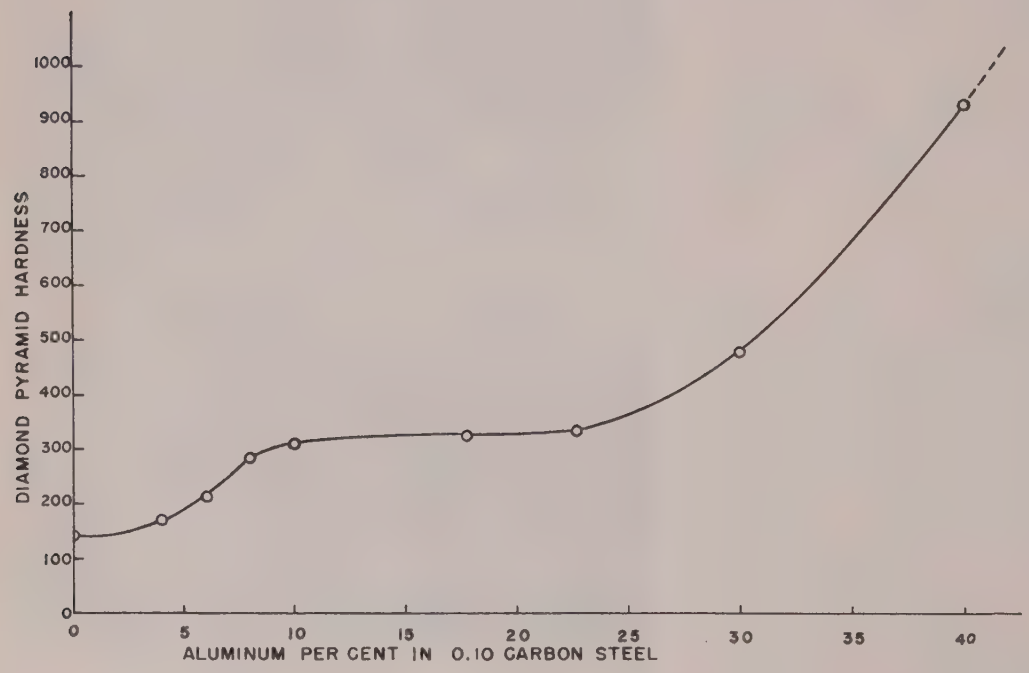


Fig. 10 – Effect of aluminum concentration in 0.10% carbon steel on hardness

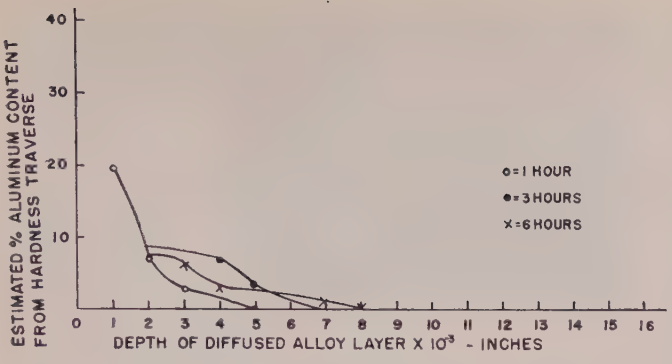


Fig. 11 – Fifteen-sec Aldip at 1300 F; 1775-1800 F diffusion heat-treatment

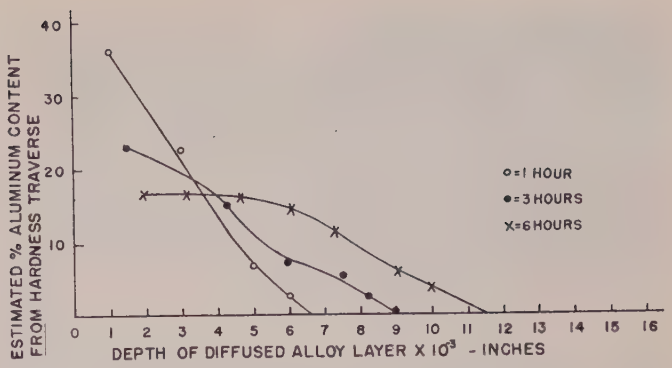


Fig. 12 – Two-min Aldip at 1300 F; 1775-1800 F diffusion heat-treatment

usual treatment for parts which operate under service conditions which do not require heat-treatment. A comparison of the two steels on the basis of aluminum dip time shows effect of carbon and chromium in reducing aluminum penetration.

Microhardness traverses in Fig. 8, which were taken through the series of complex iron-aluminum compounds, indicate that average hardness in this intermediate alloy is not greatly affected by dip time. Extremely high hardness and brittleness are a fundamental characteristic of the compounds formed during immersion of steel in pure molten aluminum and according to the equilibrium diagram the compounds contain about 55% aluminum.

Small additions of silicon and beryllium to the aluminum dip metal can lower the alloy hardness and increase ductility but not without damage to corrosion resistance properties under room and high-temperature conditions.

Diffusion Heat-Treatment of Aluminum-Coated Steel – Preliminary high-temperature tests under cyclic thermal shock conditions such as rapid heating to 1650 F followed by air-blast cooling conclusively showed the protective properties of an aluminum coating on the surface of steel but at the same time confirmed the need for a conditioning heat-treatment that would eliminate spalling and cracking of the surface alloy protective layer.

The three variables of dip time, diffusion time, and diffusion temperature were investigated to determine their relative influence or effect of different variable combinations. Data developed in connection with a study of these factors are plotted in Fig. 9, showing the rate of alpha phase or solid solution phase formation as a result of aluminum diffusion into steel at different temperatures. Steel samples used in this preliminary study represent a 2-min and 8-min dip in aluminum followed by heat-treatment at the indicated temperatures for 1 and 4 hr, respectively.

The important observations relative to this family of curves are:

1. Rate of diffusion is increased substantially when the base steel is austenitic above the AC_3, 1650 F.

2. The diffusion rate is not affected by aluminum dip time as long as there is excess iron-aluminum alloy available.

Selection of diffusion heat-treatments required consideration of susceptibility to cracking and nonadherence of the diffused alloy layer to the base steel. This spalling tendency based upon early experience in field operation depends upon the aluminum distribution and content of the alloy layer.

It was recognized, however, that although high aluminum content contributes to spalling, it is also required to be a maximum for optimum oxidation resistance.

For this reason, a study was conducted on the relationship between heat-treat diffusion and final aluminum content of the surface alloy with respect to spalling susceptibility and oxidation resistance.

Aluminum Content of the Diffused Alloy Layer – Difficulty in machining the hard iron-aluminum alloy layer led to work which correlated hardness of low-carbon steels containing various amounts of aluminum. These data are presented in the chart, Fig. 10. The relationship provides a method of rapid determination of approximate aluminum content across aluminum-iron diffused alloy layers by simply completing microhardness surveys and comparing hardness with the per cent by weight aluminum shown in the curves.

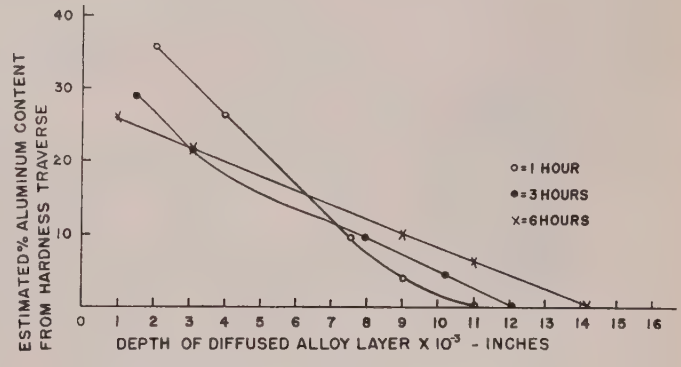


Fig. 13 – Six-min Aldip at 1300 F; 1775-1800 F diffusion heat-treatment

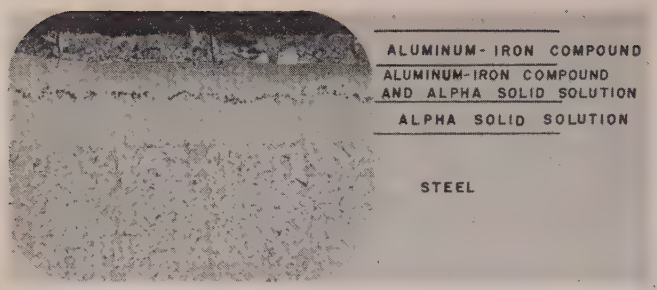


Fig. 14 – Aldip-coated SAE 1010 steel; 6-min dip at 1300 F; 1-hr diffusion at 1800 F. Nital etch. Reduced from microphotographs taken at 100x

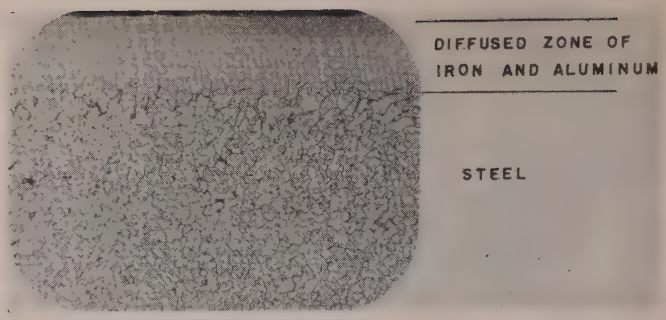


Fig. 15 – Aldip-coated SAE 1010 steel; 2-min dip at 1300 F; 1-hr diffusion at 1800 F

In this way, aluminum concentration gradients were determined on a number of different aluminum dip and heat-treat combinations.

The data shown in Figs. 11 through 13 represent aluminum content at different depths in the diffused aluminum-iron alloy layers. It is evident from the curves that a longer diffusion time is necessary to produce equivalent aluminum concentration gradients when the dip time is increased. For any specified aluminum dip, therefore, it is expedient to also specify the diffusion heat-treatment, since the combination of these control variables will determine the final amount and distribution of aluminum in the surface alloy layer.

Further evidence of dip time effect is apparent in the photomicrographs, Figs. 14 and 15.

The 6-min dip at 1300 F, followed by diffusion for 1 hr at 1800 F, resulted in formation of an alloy layer with high concentration iron-aluminum compounds at the surface and a mixture of solid solution and compound phases near the base steel.

However, the microstructure representing the short dip time shows a relatively uniform phase throughout the aluminum-rich alloy layer. The difference is attributed to the shorter dip time prior to diffusion. Therefore, in terms of aluminum concentration, the available aluminum for diffusion after a 6-min dip is too high for the 1-hr diffusion time, and experience has shown that a steel part so treated is subject to spalling under thermal shock at elevated temperatures. Referring again to Figs. 14 and 15, case depth of aluminum-dipped and diffused steel will hereafter be defined as the surface alloy layer containing iron-aluminum alloy which is measured from the surface to unaffected steel core.

Determining the Aluminum Dip and Post-Diffusion Heat-Treatment – Diffusion data are shown in Figs. 16 through 22 for various heat-treatments with different aluminum dip times and outlining the proper range of heat-treatment with respect to desired case depth showing treatments that can be used to prevent spalling.

The zones as outlined on the curves represent

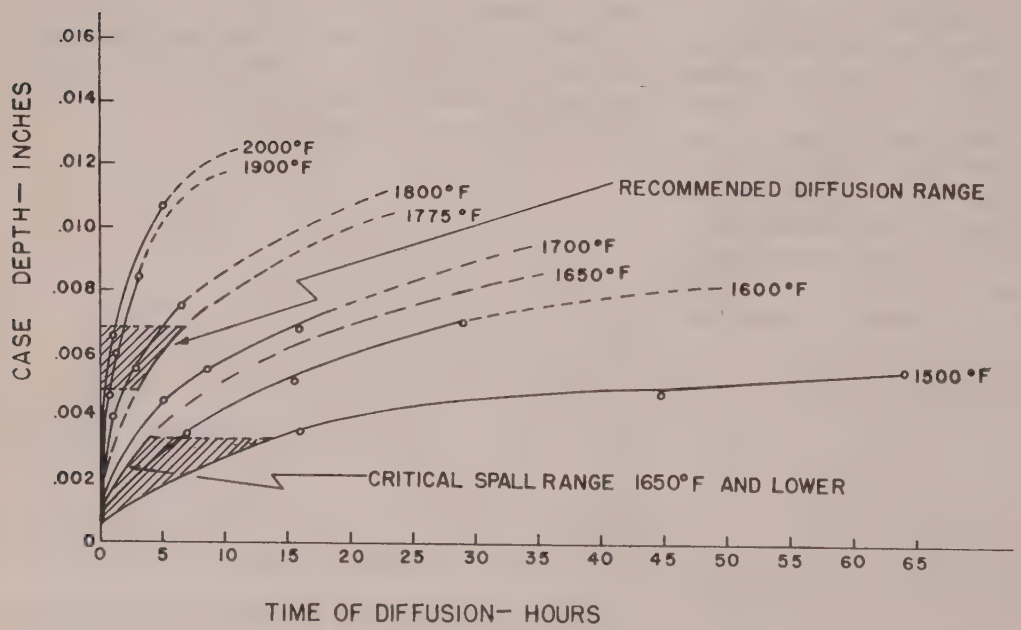


Fig. 16 – SAE 1010 steel, aluminum-dipped 15 sec at 1300 F and diffused at different times and temperatures

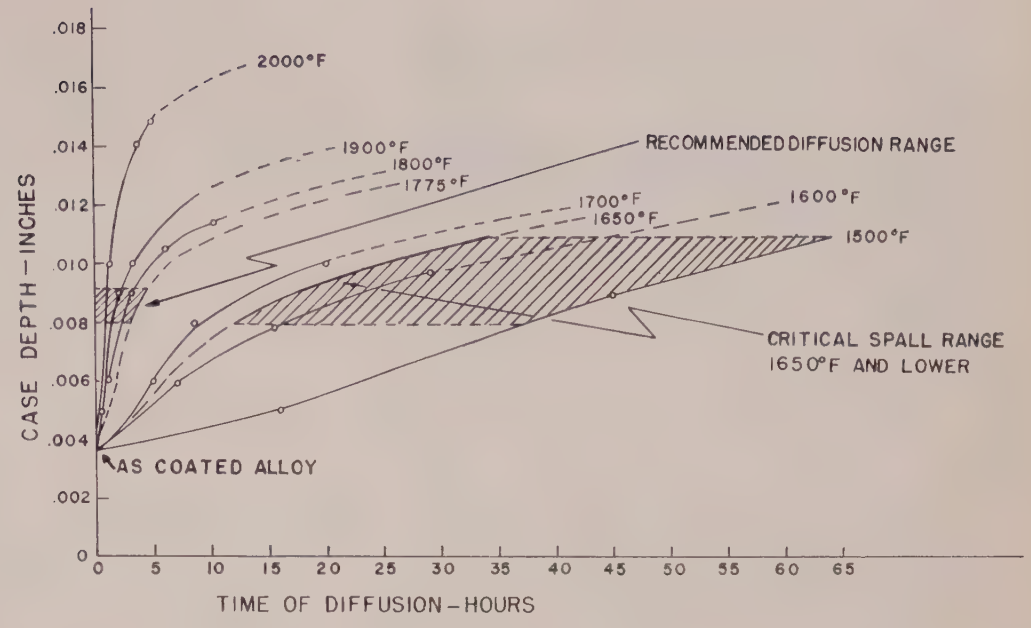


Fig. 17 – SAE 1010 steel, aluminum-dipped 2 min at 1300 F and diffused at different times and temperatures

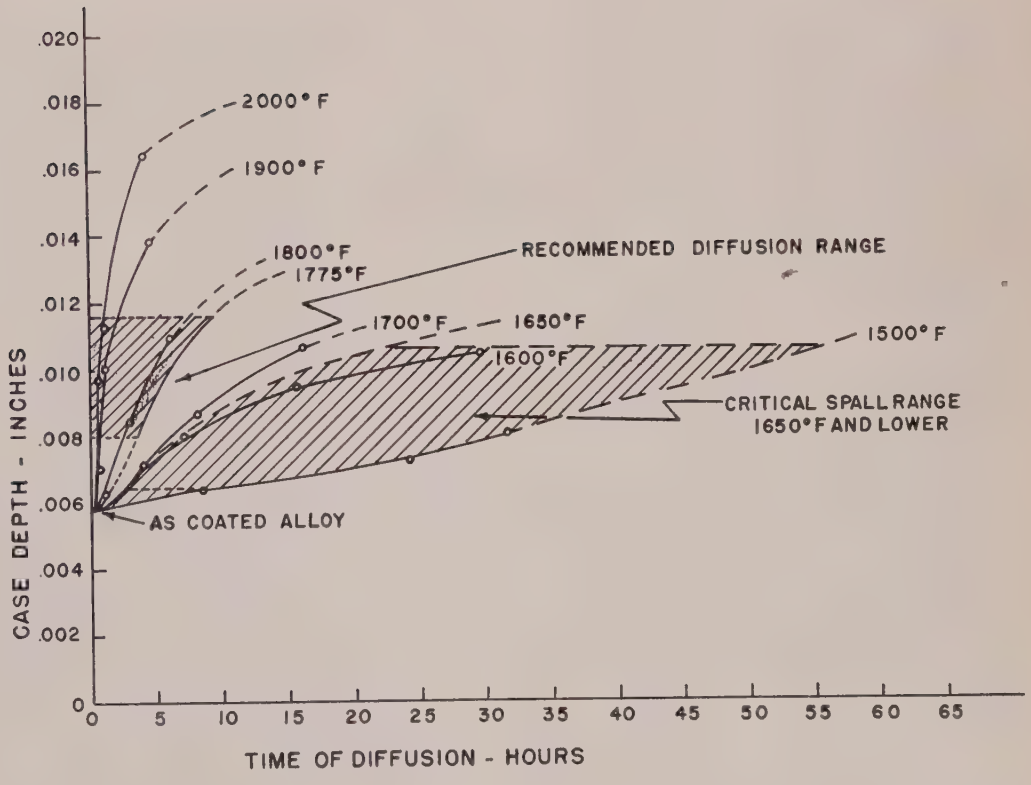


Fig. 18 – SAE 1010 steel, aluminum-dipped 4 min at 1300 F and diffused at different times and temperatures

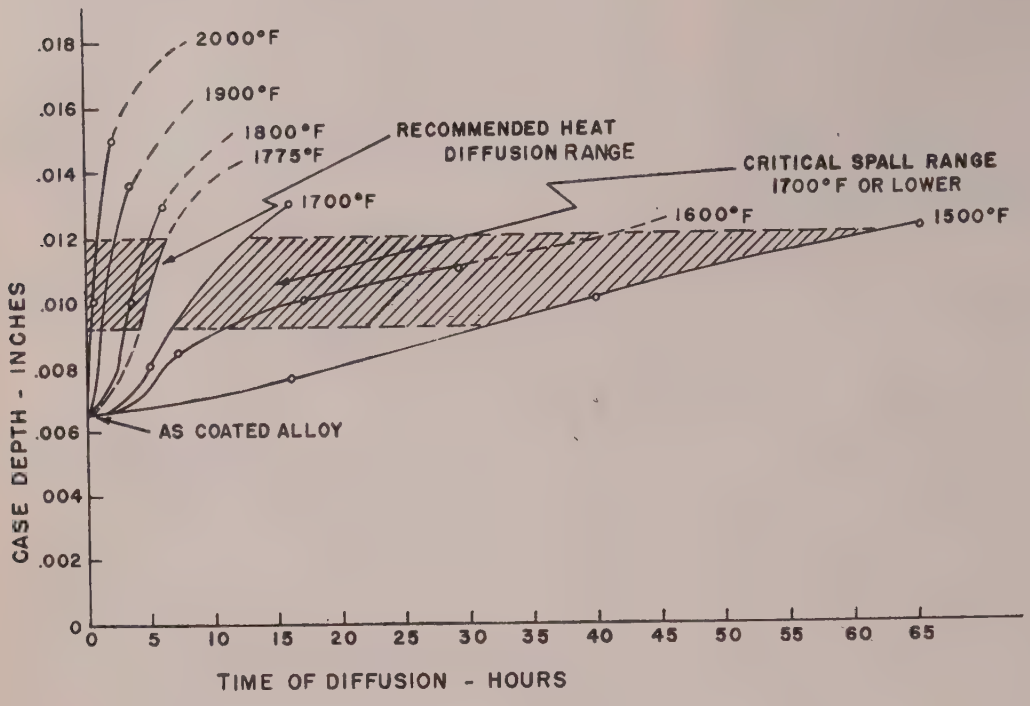


Fig. 19 – SAE 1010 steel, aluminum-dipped 6 min at 1300 F and diffused at different times and temperatures

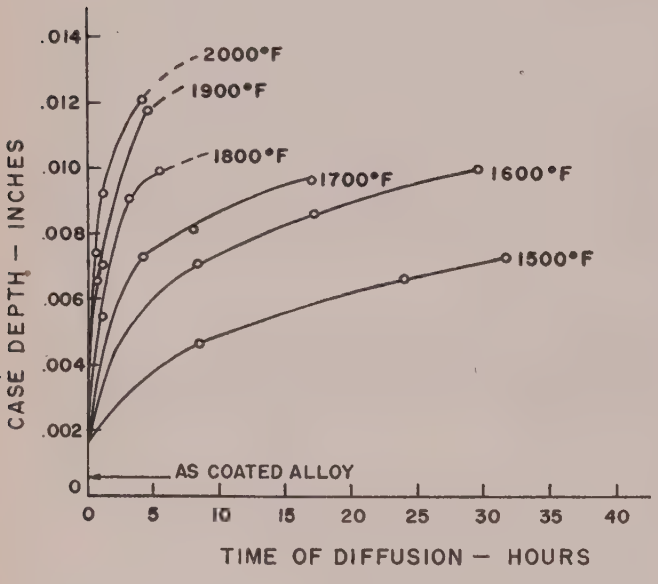


Fig. 20 – SAE 4130 steel, aluminum-dipped 15 sec at 1300 F and diffused at different times and temperatures

Table 1—Summary of Diffusion Data

Dip Time	Recommended Case Depth, in
15 sec	0.0048 – 0.0065
2 min	0.0080 – 0.0085
4 min	0.0080 – 0.0116
6 min	0.0085 – 0.0120

both production experience and laboratory tests on steel panels.

These data can be summarized for the two steels, SAE 1010 and 4130, diffused at the minimum temperature of 1775 F as given in Table 1.

The tabulation (Table 1) specifies the minimum case depth to be obtained at the minimum diffusion temperature of 1775 F which will produce the required aluminum level for resistance to thermal shock heating and cooling. The recommended maximum case depth will also insure sufficient aluminum concentration in the surface alloy for optimum heat resistance.

Oxidation Resistance – Resistance of aluminum-coated steel to oxide penetration at elevated temperatures is fundamentally due to the formation of aluminum oxides and complex iron-aluminum oxides (light gray to reddish in color) which provide a protective, adherent refractory scale. As oxygen slowly penetrates the scale to form additional oxides, resistance to further penetration becomes greater. It should be recognized, however, that at the maximum temperature of recommended operation, 1600 F, a finite life of a part may be reached after a long period of time when oxides are forming in the lower aluminum content subsurface strata. The scale thus formed is primarily iron oxide which offers less protection, and scale adherence becomes poor. Aluminum concentrations of less than 10% by weight in the alloy layer are too low for adequate protection from oxide penetration at temperatures of 1600 F or higher.

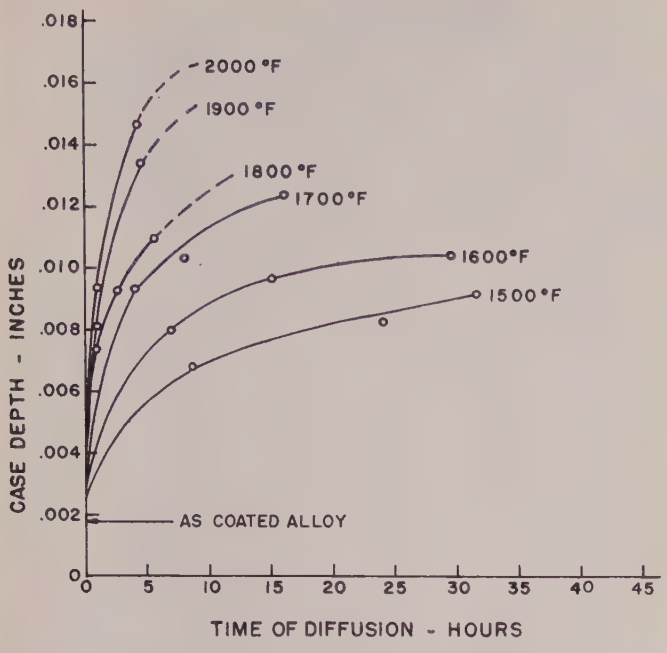


Fig. 21 – SAE 4130 steel, aluminum-dipped 2 min at 1300 F and diffused at different times and temperatures

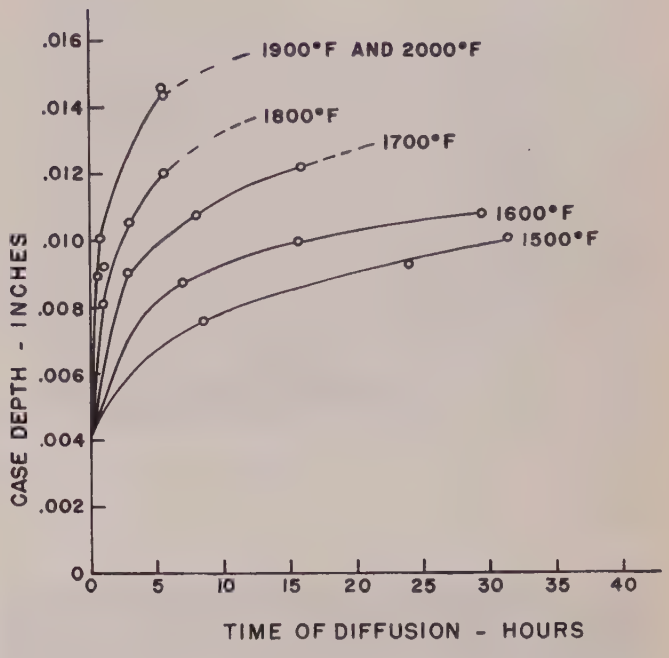


Fig. 22 – SAE 4130 steel, aluminum-dipped 6 min at 1300 F and diffused at different times and temperatures

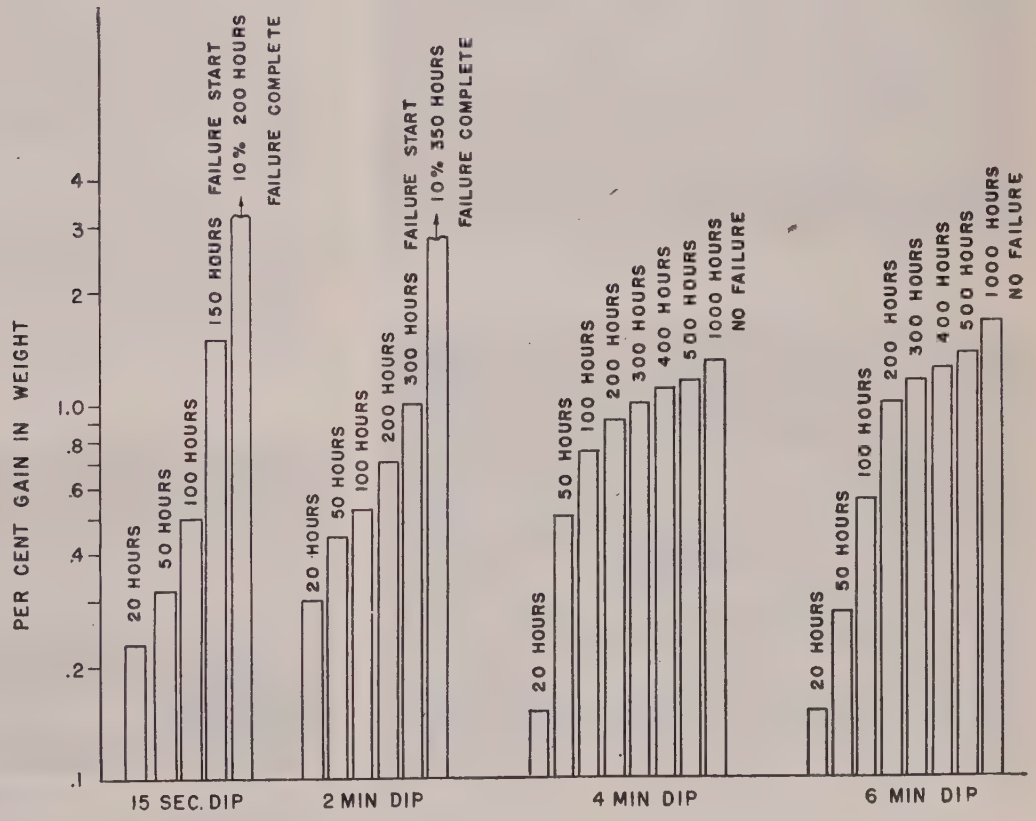


Fig. 23 – SAE 1010 steel sheet, 0.036 in. thick, Aldip aluminum-coated and diffusion heat-treated; oxidation test 1650 F, air atmosphere, forced air-cooled to room temperature; 20-hr cycle

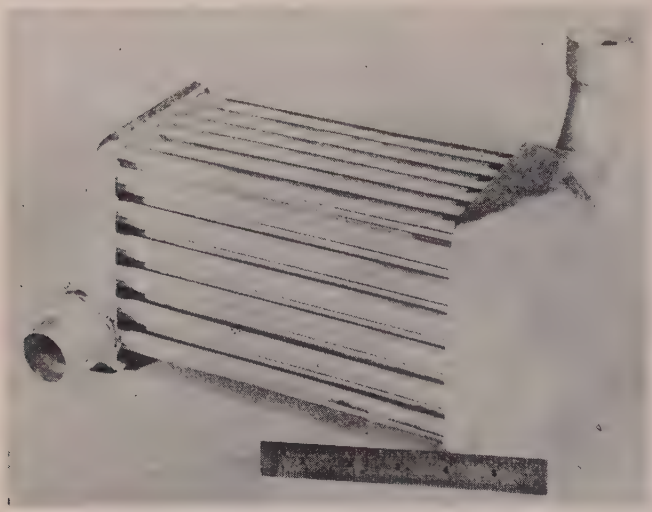


Fig. 24 – Heat-exchanger, military tank. Application requirements, oxidation resistance up to 1000 F and corrosion resistance to exhaust gas condensates. Weight of assembly, 12 lb. Previous material, type 321 stainless steel; present material, Aldip-coated carbon steel; alloy conservation per heat-exchanger, 1.2 lb nickel, 2.2 lb chromium

Fig. 26 – Carburizing spacer. Service conditions, gas carburizing at 1700 F; application requirements, resistance to carbon penetration; present material, Aldip-coated SAE 1010 steel pipe sections

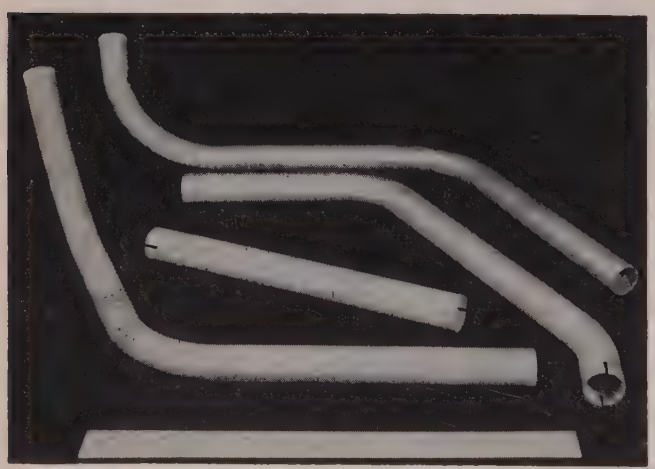


Fig. 27 – Heat-exchanger tubing. Application requirements, oxidation resistance to 1000 F and corrosion resistance to exhaust gas condensates; weight of tubing per assembly, 9.6 lb; alternate material, type 321 stainless steel; present material, Aldip-coated carbon steel; alloy conservation per assembly, 0.8 lb nickel, 1.7 lb chromium

Fig. 25 – Exhaust pipe adaptor, truck duty. Application requirements, oxidation and corrosion resistance to exhaust gas condensates. Present material, Aldip-coated and diffusion-treated carbon steel

Fig. 28 – Diesel-engine exhaust manifold. Application requirements, oxidation resistance to 900 F and corrosion resistance to exhaust gas condensates. Weight of assembly, 25 lb; previous material, type 321 stainless steel; present material, Aldip-coated carbon steel; alloy conservation per assembly, 2.5 lb nickel, 4.5 lb chromium

The bar chart in Fig. 23 compares the average life of Aldip-coated and diffusion-treated SAE 1010 steel panels representing four aluminum dip times at 1300 F. The tests were conducted in air atmosphere at 1650 F on steel which was diffusion heat-treated according to recommended specifications to produce optimum aluminum concentration in the case prior to test. All test panels were air-cooled after each 20-hr period at 1650 F. These were then weighed to determine weight gain from oxygen absorption. It is concluded from these tests that 1600 F should be the upper limit of service temperature when extremely long life is required.

This maximum service temperature could be raised if it were practical to aluminum dip for times longer than 6 min at 1300 F. In production long dip times are not specified because of iron solution in the dip metal, which if contaminated beyond 4% iron decreases fluidity and creates some problems in the operation of the equipment.

A balance between iron solution and aluminum drag-out as a coating on parts becomes possible when dipping time does not exceed 6 min at 1300 F.

Development of an economical and commercial process for aluminum-coating steel assemblies has led to the engineering acceptance of a variety of parts. Work has progressed along the lines for which Aldip was originally designed, namely corrosion resistance in exhaust systems at locations where leaded fuel exhaust gas condensate tends to collect, and for other parts requiring high-temperature oxidation resistance up to 1600 F. In many of the applications covered by these two broad categories, stainless steel grades which are usually specified can be successfully replaced by Aldip-coated steel without sacrifice in service performance.

Fabricated SAE 1010 manifolds of the type used in diesel engines are proving satisfactory in the field and replace type 321 stainless steel. An exhaust manifold of this type is shown in Fig. 28, saving 2.5 lb of nickel and 4.5 lb of chromium.

In general, it is concluded that for many service applications, aluminum-coated steel provides an opportunity to conserve on the expensive and also strategic materials.

Typical of such parts is the military tank heat-exchanger shown in Fig. 24, which operates at a maximum temperature of 900 F. Aldip-coated assemblies have performed satisfactorily in field service resisting corrosion at the exit tank ends where exhaust gas condensate collects. Replacing type 321 stainless steel in this assembly produces a considerable saving in strategic chromium and nickel with additional cost savings realized in the case of fabrication of SAE 1010 steel.

Other typical examples of maintenance or replacement parts requiring a combination of oxidation and corrosion resistance are Aldip-coated and diffusion heat-treated exhaust pipe adapters for trucks, one of which is shown in Fig. 25. The adapter is installed in the truck exhaust system in front of the muffler where temperatures are high and exhaust gas condensate collects and requires the post-diffusion heat-treatment for optimum performance.

Heat-treat fixtures such as liners for furnace trays, hooks for salt bath heat-treating, or the spacer supports for carburizing side gears which are shown in Fig. 26 perform satisfactorily under this type of service condition.

Parts requiring similar service for tank operation are the heat-exchanger tailpipe tubes in Fig. 27 which have been given military acceptance. The aluminum-coated pipes comprise only part of the full assembly which utilizes high exhaust gas temperatures to heat the tank compartment.

DISCUSSION

Develop Coating Designed To Overcome Spalling

– M. G. Whitfield
Whitfield & Sheshunoff, Inc.

WE believe that the authors of this paper have done an excellent job in summarizing and tabulating their work with aluminum-dip coatings. The facts brought out indicating the conservation of critical alloying materials are most striking.

Aluminum-dip coating has been practiced successfully for a number of years without the use of a flux bath or flux coating by our own direct knowledge and experience as well as that of others. A wide variety of objects ranging from straight tubes, bars, and sheets to more complicated cast-iron shapes, such as stove grates of all designs and sizes, burner pipes, muffler assemblies, and a number of other parts, have been coated and found satisfactory.

We believe that fatigue data for both plain coatings and diffused coatings would constitute a valuable addition to the data collected by the authors. Recently published material[a] carries some fatigue data, but only on undiffused aluminum coatings.

As a result of our work on the subject of spalling of aluminum coatings when exposed to intermittent service at elevated temperatures, we have developed a type of alloy surface layer which is exceptionally free of spalling difficulties. The typically hard alloy coating also may be useful for applications where a surface containing minute porosity has merit; that is, for cylinder walls, and so forth (patents pending).

With regard to the question of dipping time versus iron buildup in the molten aluminum-coating bath, a considerable amount of work has been done along this line already by a number of workers, and further work is undoubtedly under way. Among other methods, alloy additions to the coating bath are successfully used in some instances. One of the processes we are using experimentally holds the promise of eliminating this problem (patents pending). Any practical method of controlling solution of the base metal, in effect, also assures longer life at a given service temperature – which should open up additional fields of use for aluminum-coated products of all types.

[a] See *Iron Age*, Vol. 162, July 1, 1948, pp. 88-93: "Bonding Aluminum to Ferrous Metals," by M. G. Whitfield and V. Sheshunoff.

Research Indicates Special Needs for New Product

– K. Oganowski and T. F. Olt
Armco Steel Corp.

AS producers of flat-rolled aluminum-coated steel, we have long recognized not only the unique properties of such a product, but also the large field of applications where flat-rolled precoated sheet could not be used successfully due to the size or shape of the final product. The Aldip process appears to fulfill the requirements of this field.

The complexity of the reactions between iron and aluminum, and the relationship between iron-aluminum-alloy formation during dipping and subsequent diffusion, and the final quality of the product in respect to ductility and heat resistance of the coating, are very ably and clearly presented.

It is interesting, however, that in this presentation the authors do not refer to the influence of the unalloyed coating surface layer obtained initially during dipping operation. This layer undoubtedly participates in the diffusion stage. Would finishing methods employed in the Aldip process be amenable to regulating the thickness of the free aluminum layer, thus allowing increased thickness of the diffused coating without a necessity for extended dipping time beyond a critical 6 min?

The authors point out that the diffused coating offers satisfactory protection from scaling for long life requirements up to 1600 F. In most applications at these temperatures, mild steel as a base metal would not be satisfactory due to lack of strength, and alloyed steels as a base would have to be used. What types of low-alloy steels other than 4130 can be successfully coated by Aldip process?

ORAL DISCUSSION

Reported by Walter Mincho
Great Lakes Steel Corp.

D. K. Hanink: In the Aldip process we are attempting to control thickness of intermediate iron-aluminum layer, particularly on parts which are subsequently diffusion heat-treated, because we feel this is a controlled source of aluminum for diffusion. If dependent on aluminum overlay we could not obtain uniformity of resulting diffused alloy because of run-down during heating of the part up to the diffusion temperature. Although the aluminum overlay participates in a minor way to some alloy formation, the majority of it is oxidized during the diffusion treatment in the air atmosphere which is used.

We can aluminum-coat any steel regardless of alloy content as long as iron is the base metal.

H. E. Staehle, Surface Combustion Corp.: On higher alloy steel, in ranges from 1500 to 1600 F, for aircraft heating chambers, how successful would aluminum coating be, using Inconel and type 309? I understand in another aluminum coating process it is not practical to coat with aluminum using high nickel alloy, as nickel dissolves so fast the coating will not stick. Would the Aldip process be successful in this application?

In the Aldip process do you get full coverage without pin holes? If there are such discontinuities, is the pitting severe or does lateral diffusion prevent this?

Mr. Hanink: Inconel can be coated but occasionally, experience has shown that nickel-aluminum surface alloys are formed which do not adhere to base Inconel because of difference in expansion characteristics. However, iron alloy base materials are readily coated and no experience similar to Inconel is evident.

We experienced some difficulty with coating discontinuities in early investigations. Lateral diffusion does afford some protection. In present refined practice, however, a continuous surface coating is obtained.

A. T. Colwell, Thompson Products: How high have you been able to go in alloy content in steel containing chromium and nickel? What balance have you found that you cannot go beyond?

Mr. Hanink: Type 310 stainless steel is the present experience. It coated satisfactorily.

M. G. Whitfield, Whitfield and Sheshunoff, Inc.: What happens to surface where a little air or salt is trapped? What is percentage of this occurrence?

Mr. Hanink: We examine the geometry of part before proceeding to coat. We generally do not work with parts that are designed so that we cannot get the salt in contact with surfaces which are desired to be coated. A vibratory motion in the salt bath can be used to eliminate air traps. When we process parts, suitable procedures are used to guarantee absence of air traps.

R. Schumaker, Kolene Corp.: Can aluminum be brazed to aluminum-coated parts? Have you coated meat handling hooks and such equipment?

Mr. Hanink: In brazing, aluminum-coated material is treated as if made of aluminum. Specific items you mentioned have not been aluminum-coated.

H. E. Weber, Thompson Products: What report is available on effect of manganese on alloy layer at the surface? What are characteristics of failure of any dip process coating?

Mr. Hanink: Aluminum will diffuse in austenitic grade steel. With regard to cracking or spalling, the part must undergo diffusion heat-treatment so the part is not subject to such defects. Diffusion treatment as in 1010 or 4130 covered by this paper requires lowering of aluminum content of surface alloy layer. Temperature and time of diffusion treatment may be entirely different for stainless steel than for plain carbon or low alloy steel.

J. B. Russell, Kaiser Aluminum and Chemical Corp.: Have you been successful in being able to clean internal threads so that nuts can be screwed on by hand in development of pole line hardware?

Have you been successful in controlling iron content of bath by control cooling method?

Your silicon and beryllium finds are contrary to our finds. Aluminum-beryllium-silicon have better corrosion resistance than aluminum or aluminum-silicon type coatings.

Equipment is expensive as described. What is cost of this process that may be attributed to use of flux? Is it suitable for small operator?

Mr. Hanink: Cleaning of threads is a major problem in such hardware.

We control aluminum dipping time so that aluminum drag-out is sufficiently high that when fresh aluminum is added, iron content remains about same, 2½ to 3%.

We have found silicon and beryllium to be detrimental to corrosion separately but in combination we have had no experience. In any event, under corrosive conditions, after aluminum overlay has disappeared due to formation of aluminum hydroxide, aluminum-iron alloy must then be resistant to corrosion. Since silicon and beryllium reduce the thickness of this alloy layer, corrosion resistance would then be less after aluminum overlay has been eliminated.

The small operator who may be interested in the Aldip process must face high initial investment costs, but beyond that we have a relatively inexpensive process, because flux will not cost appreciably more than average heat-treating salts.

Combustion-Chamber Deposition and Knock

H. J. Gibson, C. A. Hall, and D. A. Hirschler, Ethyl Corp.

This paper was presented at SAE National Fuels & Lubricants Meeting, Tulsa, Nov. 6, 1952.

THE increase in antiknock requirement of engines which results from the accumulation of combustion-chamber deposits is not a new problem. However, it has become of greater importance as the antiknock quality of fuels and the antiknock requirements of engines have reached higher levels. Previous papers on this subject[1, 2, 3, 4] have indicated that an average value for the increase due to deposits in passenger cars is about 10 octane units in the 80-90 octane number range. From the standpoint of the automotive industry, the efficiency of engines could be improved by the benefits from compression ratios approximately one unit higher if the effects of combustion-chamber deposits on requirement could be eliminated. Superimposed on the problem of increase in ordinary knock is that of autoignition induced by deposits.[5] This latter problem is a subject in

[1] See SAE *Quarterly* Transactions, Vol. 3, December, 1949, pp. 557-570: "Factors Affecting Octane Number Requirement," by H. J. Gibson.

[2] See SAE *Quarterly* Transactions, Vol. 5, October, 1951, pp. 565-576: "Possible Mechanisms by Which Combustion-Chamber Deposits Accumulate and Influence Knock," by L. F. Dumont.

[3] See SAE *Quarterly* Transactions, Vol. 5, October, 1951, pp. 577-583: "Effects of Combustion-Chamber Deposits on Octane Requirement and Engine Power Output," by J. B. Duckworth.

[4] "An Engine Laboratory Study of Motor Oil Characteristics As One of the Factors Influencing Combustion-Chamber Deposits," by D. E. Carr, L. A. McReynolds, S. C. Britton, and R. E. Linnard. Paper presented at SAE Annual Meeting, Detroit, January 17, 1952.

[5] See SAE *Quarterly* Transactions, Vol. 6, October, 1952, pp. 724-752: "Flame Photographs of Autoignition Induced by Combustion-Chamber Deposits," by L. L. Withrow and F. W. Bowditch.

THIS paper discusses a number of factors involved in the problem of octane-number requirement increase due to combustion-chamber deposits. A laboratory single-cylinder engine test procedure, which evaluates the effects of various fuel and oil factors, is presented with data showing its correlation with passenger-car operation under light-duty, city-driving conditions.

The influence of engine operating conditions during accumulation of deposits and the importance of engine conditions selected to evaluate the magnitude of the requirement increase are illustrated. It is indicated that organic materials formed from both fuel and oil are of major importance in deposit formation.

Data are presented which show that tel added to pure hydrocarbons of different chemical types may have different effects. It is shown that the carbon/hydrogen ratio of leaded pure hydrocarbons influences the amount and composition of the deposit formed. Among leaded commercial gasolines there appears to be little effect of changes in hydrocarbon composition, although there is a trend toward smaller requirement increase as fuel endpoint is reduced.

Similarly, removal of heavy ends from lubricating oils tends to lessen requirement increase, and the magnitude of this effect is dependent on the contribution of the fuel to the organic portion of the deposit. Maximum effects are observed when clean-burning leaded isooctane is used as the fuel.

Combustion-chamber design and deposit location influence the magnitude of the requirement increase.

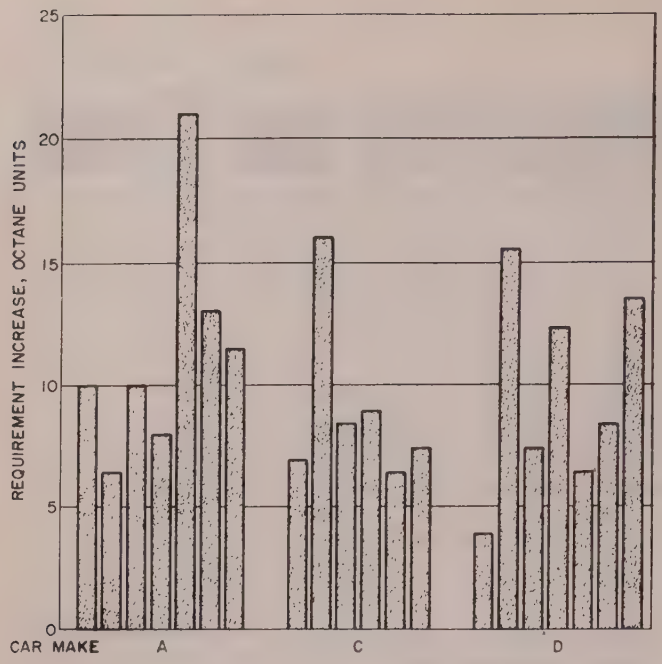


Fig. 1 – Requirement increase due to deposits in several passenger cars of three makes.

Fuel: Catalytically cracked gasoline, 3-A, containing 1.0 ml tel per gal.
Oil: Premium, high V.I., 24-A
Operating Conditions: 12,000 miles, city-suburban operation

itself and, while it cannot be separated completely from the increase in ordinary knock due to deposits, will not be discussed in this paper.

This paper discusses the difficulties involved in studying the requirement increase problem, which is generally acknowledged to be most severe in vehicles operated in crowded metropolitan areas. It presents a laboratory test procedure based on a single-cylinder engine and shows its correlation with passenger-car operation under light-duty, city-driving conditions. Single-cylinder and multicylinder engine data are used to illustrate the effects of factors such as fuel type and volatility, tel concentration, oil type and volatility, and engine type and operating conditions on increase in antiknock requirement due to deposits. A hypothesis as to why deposits increase antiknock requirement is presented.

Why Requirement Increase?

Two excellent papers,[2, 5] published in the last few years, offered explanations as to why deposits increase antiknock requirement. The authors of these papers believe that deposit effects can be ascribed largely to:

1. Increase in compression ratio caused by the physical volume of the deposits.
2. Heating of the intake charge by the deposits.

The change in compression ratio caused by deposits is easily obtained by measurement of combustion-chamber clearance volume before and after deposit removal. A sonic device with which volume can be measured without disturbing the cylinder head is suitable for this purpose. Such measurements have indicated that deposits in multicylinder engines increase the compression ratio as much as 0.3 unit. This can be considered as equivalent to an increase in requirement of about two units in the 80-90 octane number range.

Heating of the intake charge results from the heating capacity of the deposits. This term is used to describe the ability of deposits to absorb heat during the combustion and exhaust strokes of one cycle and to release a portion of this heat to the fresh charge during the following cycle. The magnitude of this effect depends on a number of deposit factors such as amount and location, the physical characteristics of the deposit layer as well as the condition of its surface, and the amount of heat transferred to the wall of the combustion chamber.

A previous paper[6] has pointed out that heating of the charge reduces air consumption to a significant degree in some engines. Reduction in charge density tends to reduce knocking tendency, whereas heating of the charge by deposits increases knocking tendency. Although these two effects are opposite in direction, the resultant generally increases the knocking tendency. Different engine makes show differing deposit effects.

Work done by the Ethyl laboratories has shown that it is possible to duplicate the thermal effects of deposits on octane number requirement by:

1. Inserting metal disks in the combustion chamber.
2. Building up a synthetic deposit of fused, foamy sodium silicate.
3. Increasing the jacket temperature.

All three of the above means involve heating of the intake charge. If each is adjusted so as to reduce the weight of the incoming charge, due to heating effects, to the same degree as by a normal set of deposits, then the effect on octane number requirement is approximately the same as the normal deposits. These experiments give emphasis to the importance of the heating effects of deposits in this problem. While catalytic effects of some deposit constituents cannot be entirely ruled out, it would appear that such effects, if they do exist, are

Table 1 – Octane Number Requirement as Affected by Deposit Condition

After 10,000 miles of deposit accumulation	91
After operation at high speed and full throttle	87
After deposit removal	77

[6] See SAE *Quarterly* Transactions, Vol. 6, October, 1952, pp. 594-605: "Combustion-Chamber Deposits and Power Loss," by H. J. Gibson, C. A. Hall, and A. E. Huffman.

minor in comparison with the volume and heating effects of the deposits.

Evaluating the Problem

There are two principal phases in any deposit study which must receive equal attention. They are (1) the phase during which deposits are accumulated, and (2) the phase during which their effects are evaluated.

Deposit Accumulation Phase

The wide variability of deposit effects from engine to engine is well-known, particularly in vehicles driven in ordinary road service without careful control. Fig. 1, based on data from an earlier paper,[1] illustrates this point by showing that the requirement increase between two cars of the same make may vary by as much as 15 octane units. The cars were in normal service in a utility fleet and all used the same fuel and lubricating oil. The reasons for the wide variability in deposit effects are not known, but it is believed that the following factors were involved:

1. The average severity of driving during deposit accumulation.
2. The type of driving to which the vehicle was subjected just prior to the test observations.
3. The oil consumption rate.
4. The fuel consumption rate.
5. The ignition timing – engine speed combination at which maximum knock occurred.
6. The engine cooling-jacket temperature.

It is obvious that when variability such as that shown in Fig. 1 exists in vehicle tests, either enough tests must be run to obtain a sound average value (as was the case above) or individual vehicle tests must be run under carefully controlled conditions to minimize variability. While variability is not nearly as great in tests run in multicylinder engines in the laboratory as on the road, care must be taken to keep all variables under rigid control. Some of these factors will be covered in more detail later in this paper.

Deposit Evaluation Phase

Of great importance are the conditions under which the effects of a given set of deposits, from either laboratory or road test, are evaluated. It is obvious that care should be taken to avoid modification or disturbance of deposits during this evaluation. Combustion chamber clean-up can be minimized by the use of operating conditions which are as mild as possible relative to the conditions under which deposits were accumulated. The importance of this item alone is demonstrated by the data in Table 1. The data show the effect on antiknock requirement of severe operating conditions after deposits had been accumulated during 10,000 miles of road operation of a modern V-8 engine in simulated city-suburban service. Primary reference fuels were used for these observations.

The Authors

H. J. GIBSON (M '32) is a research supervisor in the Ethyl Corp. Research Laboratories. He joined Ethyl immediately after graduating from the University of Michigan in 1930, and has spent the intervening years in conducting and directing research on the interrelationship of fuels, lubricants, and engines.

C. A. HALL (M '43) is an assistant research supervisor in the Ethyl Corp. Research Laboratories. He received his B.S.M.E. from Ohio State University in 1937, and after a year with the Frigidaire Division of General Motors and four years as an instructor in mechanical engineering at Rice Institute, he obtained his M.S. degree from the University of Michigan. Since 1942 he has been with Ethyl Corp., engaged in research on fuels and lubricants.

D. A. HIRSCHLER (M '50) is a research engineer with the Ethyl Corp., where his activities have been devoted largely to the study of engine deposit problems. Mr. Hirschler has been associated with Ethyl since 1940, except for the period from 1941 to 1946 when he served as an engineering officer with the Air Force. He received his B.A. from the College of Emporia, and his B.S.M.E. from the University of Kansas.

The effects of atmospheric conditions on octane number requirement have been presented previously[1] but are mentioned here as a reminder that deposit effects should be determined under atmospheric conditions as uniform as possible. Correction factors for changes in atmospheric temperature, pressure, and humidity for a given engine can

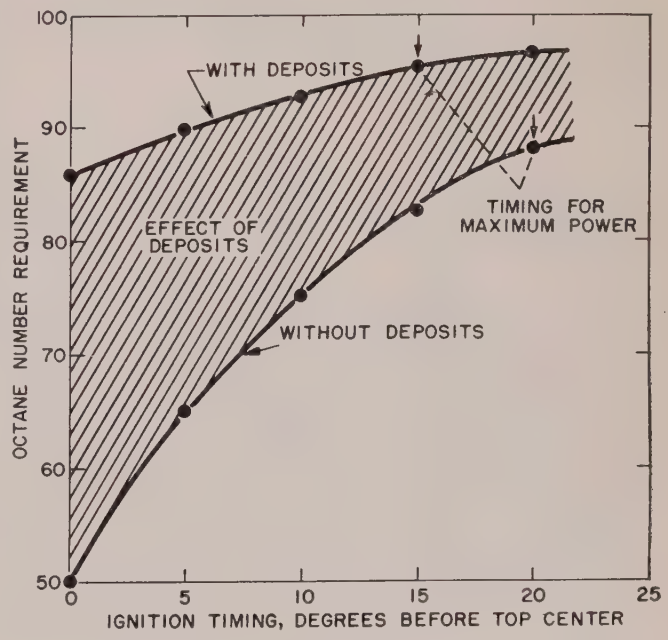


Fig. 2 – Relation of octane number requirement to ignition timing, with and without combustion-chamber deposits.

Engine: Single-cylinder, L-head
Deposits accumulated at tdc ignition timing

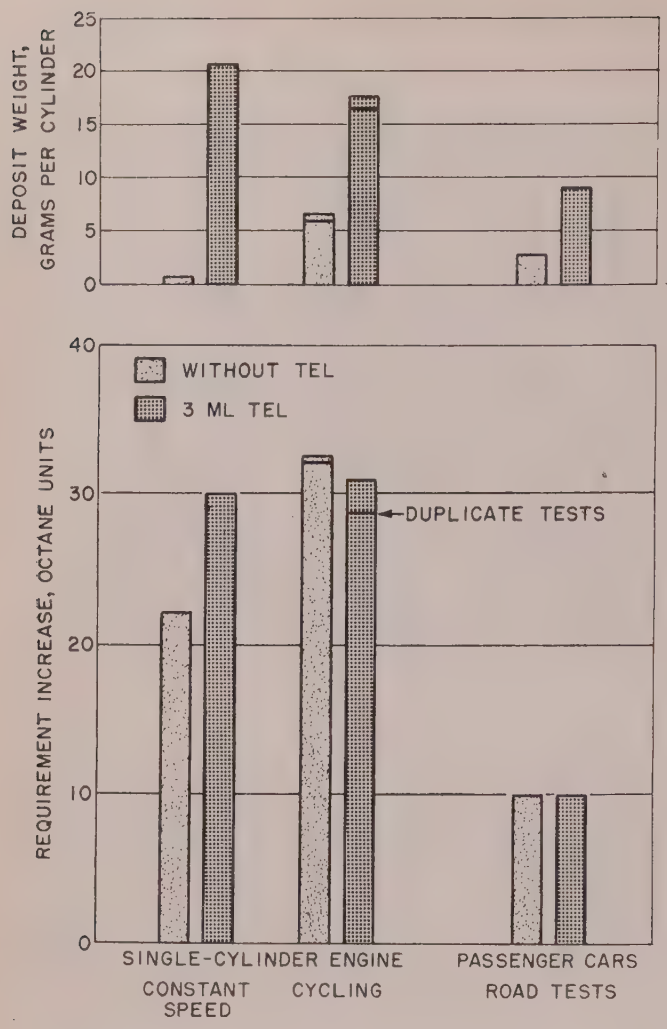


Fig. 3 – Effect of tel on requirement increase and deposit weight under various operating conditions.

Fuel: Catalytically cracked gasoline, 3-A
Oils: Nonadditive, high V.I.

be established, and the accuracy of requirement observations can be improved materially by their use.

The ignition timing at which deposit effects are observed is an important variable. A set of data taken in a single-cylinder test engine will serve to illustrate this point, which has been confirmed by many observations in both single and multicylinder engines. Fig. 2 presents requirements obtained at a number of ignition timing settings after requirement stability was reached and again after deposit removal. It will be noted that the effect of deposits became progressively less as the ignition timing was advanced to the maximum power setting. While part of this effect is related to the relative size of octane numbers at various levels, it is not entirely explainable on that basis.

Engine speed is a factor in evaluating deposit effects. Data obtained in vehicles on the road generally show less effect of deposits at high than at low speeds. Since these results are usually obtained at full throttle with standard distributors, ignition timing is a factor contributing to the apparent effect of speed. In general, ignition timing is less than that required for maximum power at the lower engine speeds, and approaches that required for maximum power at higher speeds. This tends to make deposit effects less at the higher engine speeds.

The effect of the fuel/air ratio at which observations are made of deposit effects has been investigated quite thoroughly. If the same fuel/air ratio is used for observations with and without deposits, there appears to be little effect.

The reference fuels used are of importance; in general, the change in requirement in terms of sensitive commercial gasolines is about two-thirds of that observed with primary reference fuels.

The many factors which can affect the evaluation of a given set of deposits make it desirable to minimize these effects insofar as is possible. For this reason, all data in this paper are based on observations before and after deposit removal. Such practice, which is recommended for research in this field, minimizes the effects of uncontrolled atmospheric variables and unavoidable mechanical changes which occur over a period of time. Likewise, all data are based on tests run to requirement equilibrium. Primary reference fuels were used for all requirement observations.

Laboratory Engine Test Procedures

For an investigation involving numerous engine tests on many variables, road testing involves high costs, the expenditure of much time and the difficulties of close control. As a result, the development of a reliable laboratory engine test procedure was one of the first problems to be considered. The engine selected was a single-cylinder CFR knock-test engine with the standard overhead-valve cylinder replaced by an L-head cylinder. The L-head cylinder was selected primarily because the combustion chamber could be opened easily for access to the deposits. A single-cylinder engine was desirable because it was susceptible to close control of operating conditions, readily knockrated and permitted testing of comparatively small samples of fuels and oils.

Cycling versus Constant-Speed Operation

A test procedure was desired for the single-cylinder engine which would respond to the deposit effects of fuel, oil, and tel in the same relative manner as passenger cars in light-duty service, in which requirement increase effects are the most troublesome. Available data on which to base development of a laboratory test procedure included road tests run in a fleet of 66 passenger cars. These road tests, which have been reported previously[1] were run with and without tel as Motor Antiknock Compound (hereafter tel) added to a full-boiling

range, catalytically cracked naphtha.

The effect of constant speed and load operation on requirement increase was investigated in the single-cylinder engine using the same gasoline, with and without tel, as in the fleet test. Tests of approximately 170 hr duration, under the conditions shown in Table 2, were run to permit requirement to reach equilibrium values. Results were different from those observed in the fleet test. As shown in Fig. 3, the clear gasoline produced substantially less requirement increase than the same fuel containing 3.0 ml tel, whereas the 66-passenger-car fleet test showed no effect of tel. Similarly, the weight of deposit produced by the leaded fuel was 22 times greater than that of the clear fuel, rather than three times greater as had been observed in the road tests. This indicated that deposit formation and requirement increase on the road involved factors not present in the constant-speed tests.

To approximate road operation more closely, a cycling test schedule for the single-cylinder engine was then developed using the engine conditions also shown in Table 2.

Moderate engine temperatures, low engine speeds and considerable idling operation were used to simulate light-duty type of service. The antiknock value of the fuels used during deposit accumulation was always sufficient to prevent knock. Fig. 3 shows that with this test cycle, requirement increase was essentially unaffected by tel. Moreover, the ratio of leaded to unleaded deposit weights was 2.9/1, similar to the 3.0/1 ratio observed in the fleet tests. In this figure and in the following figures, right-angle lines through columns or bars indicate the results of duplicate tests. It will be noted that repeatability was reasonably good.

Requirement increase in the single-cylinder engine cycling procedure was considerably higher than was obtained on the road. This is not believed to be a serious drawback. Other work to be discussed later has shown that the retarded ignition timing, such as used in the laboratory tests, tends to increase both the amount of deposit formed and the effects of the deposits on knock. As a result, an expanded scale is obtained which makes small effects more readily observed. This is felt to be as useful for the laboratory test program as would be a direct numerical correlation with road results.

In addition to influencing the effects of tel, the type of engine operation was found also to influence the relative effects of some leaded fuels. Fig. 4 shows that under constant-speed conditions, leaded diisobutylene gave less requirement increase than leaded toluene, while under cycling conditions the relative results with the two fuels were reversed. This figure also shows that the unleaded catalytically-cracked road-test fuel was affected most by change in type of operation, while the leaded blend gave the same requirement increase in both types of tests. In addition to responding to tel concentration in the same manner as passenger-car operation, the single-cylinder engine cycling procedure has been found also to evaluate fuel and oil effects satisfactorily, as will be shown in later sections of the paper.

Table 2 – Operating Conditions Used in Single-Cylinder, L-Head Engine

	Constant-Speed Test	Cycling Test	
Load	Full Throttle	Idle	Full Throttle
Cycle Duration, sec		50	150
Speed, rpm	900	600	900
Fuel/Air Ratio	0.077	0.087	0.077
Ignition Timing	Tdc	Tdc	
Jacket Coolant Temperature, F	150	150	
Oil Temperature, F	160	160	
Intake Air:			
Temperature, F	112	112	
Moisture, grains/lb	34	34	
Pressure	Atmospheric	Atmospheric	
Compression Ratio	7.0/1	7.0/1	
Test Duration, hr	170	165	
Fuel Required, gal	70	60	

Fig. 5 illustrates how the cycling laboratory test differentiates among several hydrocarbons and gasolines. In this comparison, requirement increase of

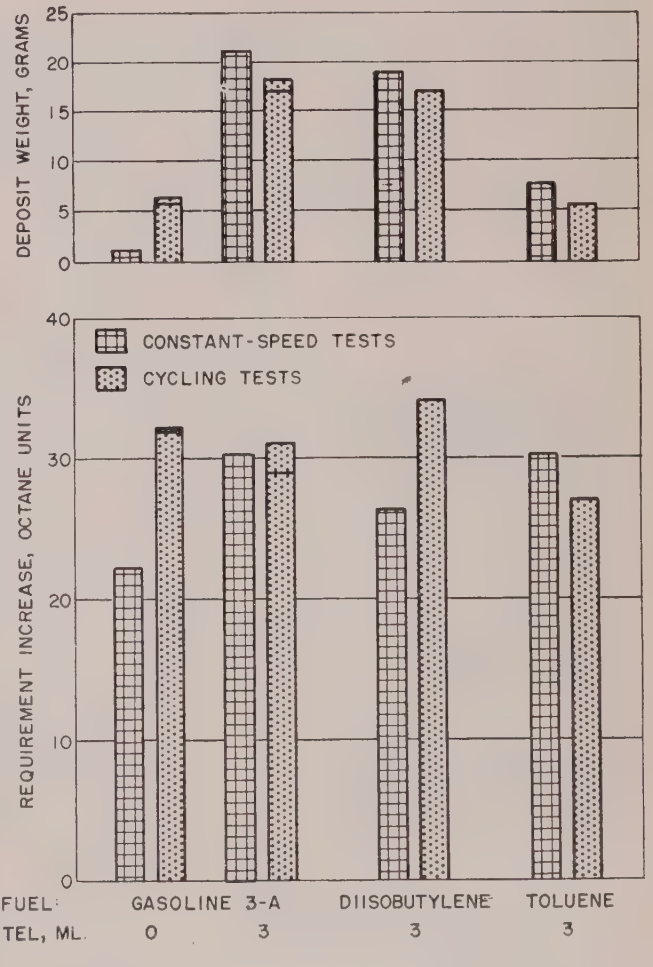


Fig. 4 – Influence of operating conditions on the relative deposit effects of several fuels.

Engine: Single-cylinder
Oil: Nonadditive, high V.I., 1-A

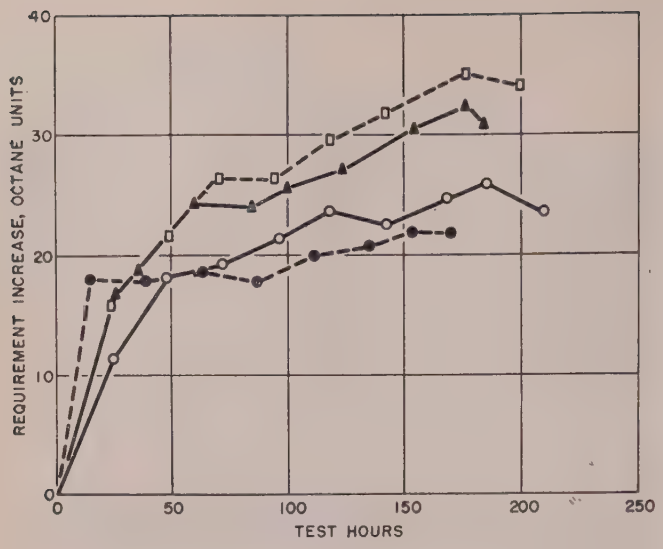


Fig. 5 – Relation of requirement increase to test duration for several fuels.
Procedure: Single-cylinder engine, cycling
Fuels: Contain 3 ml tel
Oil: Nonadditive, high V.I., 1-A

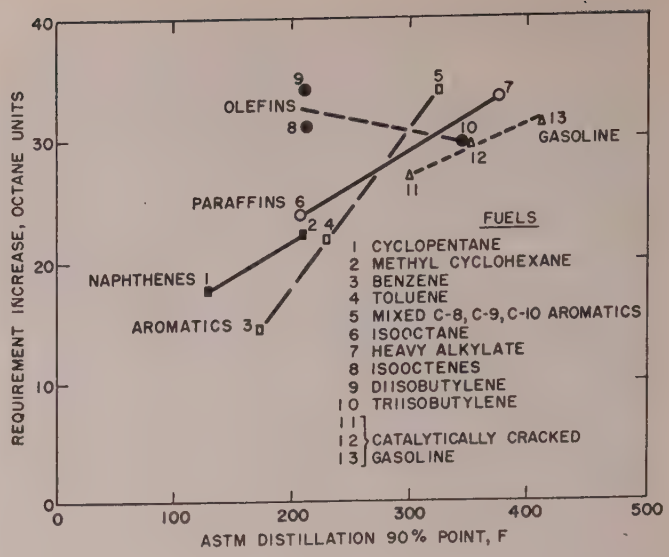


Fig. 6 – Effect of hydrocarbon type and volatility on requirement increase.
Procedure: Single-cylinder engine, cycling, low oil consumption
Fuels: Contain 3 ml tel
Oil: Nonadditive, high V.I., 1-A

the poorest test was approximately 50% greater than that of the best. This figure also shows the value of running tests sufficiently long to reach maximum requirement. The earlier stages of requirement build-up do not necessarily relate to end results, and the differences among the fuels become more pronounced in the later stages of the tests.

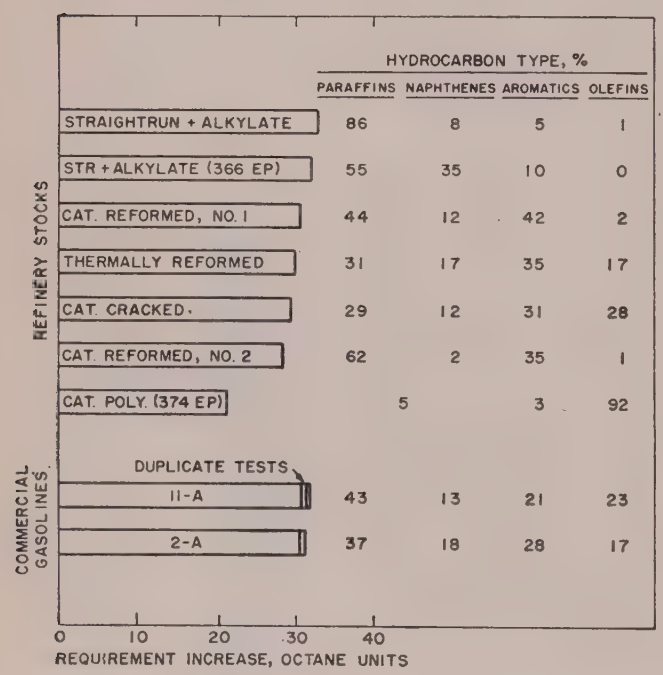


Fig. 7 – Requirement increase with various fuels.
Procedure: Single-cylinder engine, cycling
Fuels: Approximately 400 F endpoint, 3 ml tel
Oil: Nonadditive, high V.I., 1-A

Fuel Factors

Fuel is one of the factors which can influence requirement increase. As a step toward determining the relative importance of several fuel characteristics, tests were made to investigate the effects of volatility and composition, tel and sulfur content.

Fuel Volatility and Composition

Several fuels were tested which were representative of different individual hydrocarbon types and which had different volatilities. These fuels contained 3.0 ml tel per gal and were tested with a nonadditive, high viscosity-index crankcase oil. Fig. 6 shows that there was a marked tendency for requirement increase to become greater within the paraffinic, naphthenic, and aromatic fuel types as the boiling point of the fuel was increased. However, the low-boiling olefins produced deposit effects as great as those in the high-boiling range. No one hydrocarbon type containing tel appears superior throughout the boiling range investigated, but in the low-boiling range, aromatics appear best.

The significance of fuel volatility was investigated also by distilling a catalytically-cracked naphtha to three different end points: 360, 398, and 454 F. Requirement increase was found to become less as the heavier portions of the fuel were removed, and the relationship between requirement increase and volatility, Fig. 6, was about the same with this gasoline as that found with the paraffinic fuels.

Seven different full-boiling blending stocks, produced by a variety of refinery processes, were tested to determine their effects on requirement increase. Five of these blending stocks had similar distilla-

tion curves with end points of approximately 400 F. Two of the fuels were somewhat more volatile. These data are presented in Fig. 7. As might be expected from the results of the tests of individual hydrocarbons at about 350 F 90% point (Fig. 6), there were only minor differences in requirement increase among these refinery stocks in spite of their wide variation in hydrocarbon composition. The two fuels blended from straightrun and alkylate produced the greatest requirement increase, which was slightly above the values obtained with the cracked stocks. The one fuel most distinct from the rest was the highly olefinic catalytic polymer. The exceptionally low requirement increase obtained with this fuel, which was somewhat more volatile than the others, was lower than would be expected on the basis of volatility alone, and is in contrast with the high requirement increase obtained with the olefinic hydrocarbons. This shows that no generalizations can yet be made as to the behavior of the olefins, and that additional work on this fuel type would be of considerable interest.

From these findings it would be expected that the relative tendencies of commercial gasolines to contribute to requirement increase would be governed more by the amount of high-boiling components than by the hydrocarbon composition of the whole fuel. As will be shown later in this paper, fuel effects are dependent on the contribution of the oil to the deposits. It is likely, therefore, that effects different from the above might be found if an oil of deposit-forming tendency different from that of the reference oil were used with the same fuels.

Tel Concentration and Deposit Composition

Tel is another factor to be considered in a discussion of fuel effects. As previously pointed out, tel concentration has been found to have no influence on requirement increase with typical commercial gasolines. This absence of effect, however, is believed to be the resultant of the diverse effects which tel has in the different hydrocarbon types which make up gasoline. For example, two chemically different hydrocarbons of similar volatility, isooctane and toluene, responded oppositely to addition of tel. Results of tests on these two fuels with tel concentrations of 0, 1.0, and 3.0 ml per gal, shown in Fig. 8, indicate that without tel isooctane produced a low requirement increase and toluene a high value. When tel was added to isooctane, requirement increase became much greater and the weight of deposit was also considerably increased. The converse was true with toluene as regarding requirement increase. Addition of tel to toluene caused requirement increase to become smaller while deposit weight was increased by a comparatively small amount. Passenger-car road tests on the same fuels and oil, which will be discussed later, have confirmed these effects.

The tests on isooctane and toluene showed that fuel type was a major factor influencing deposit weight, as well as requirement increase. Fig. 9 shows the weights of lead compounds, carbon, and total deposits produced by isooctane and toluene with two oils, one of high viscosity index and the other of low viscosity index. On the road and in the laboratory the leaded aromatic produced about half as much deposit on a weight basis as the leaded paraffin, with the high V.I. oil. This appears to have been caused mainly by a reduction in the amount of lead compounds, since the amount of carbon remained about the same for both fuels. The low V.I. oil produced less weight of deposit than the other oil with both leaded fuels, but the proportions of deposit constituents were not affected appreciably by oil type.

In passing, it should be mentioned that discussion in this paper of deposit quantity or composi-

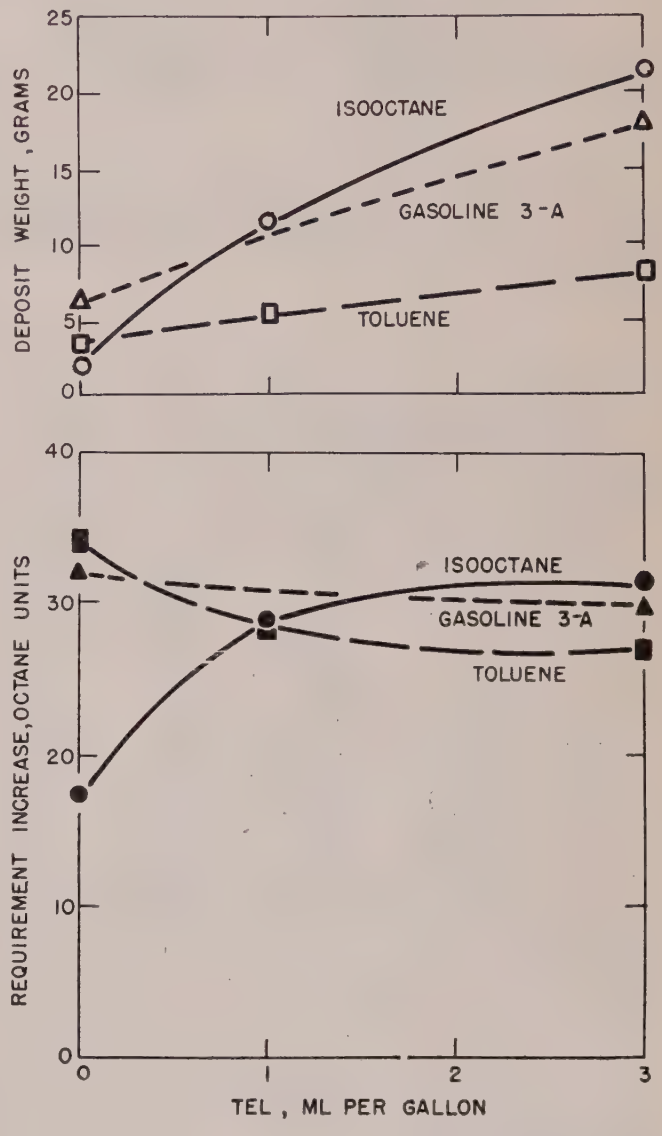


Fig. 8 – Effect of tel content and fuel type on requirement increase.
Procedure: Single-cylinder engine, cycling
Oil: Nonadditive, high V.I., 1-A

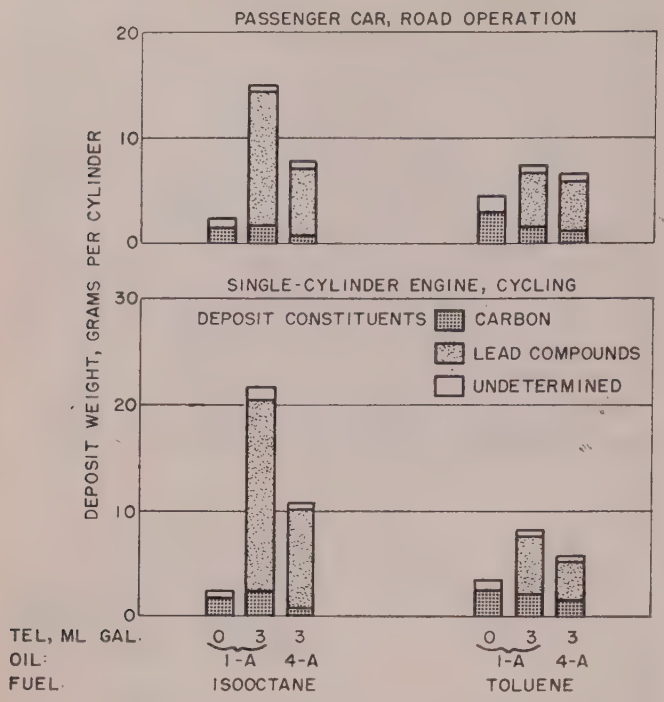


Fig. 9 – Effect of fuel type, oil type, and tel on deposit weight and composition.
Oils: 1-A, high V.I.; 4-A, low V.I.

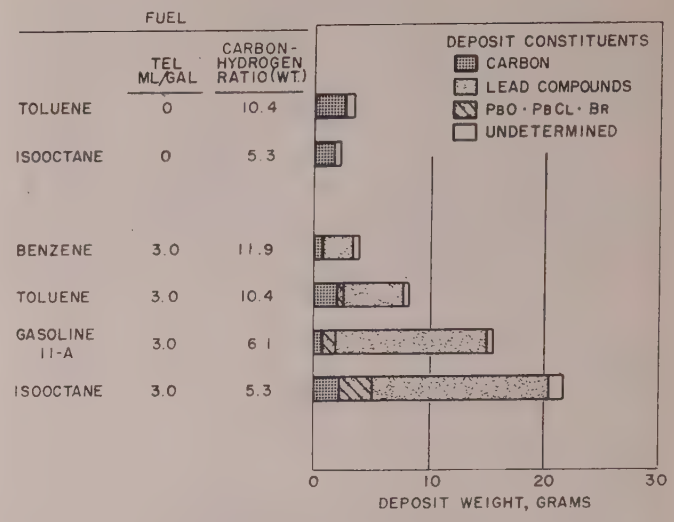


Fig. 10 – Effect of fuel type and tel on deposit weight and composition.
Procedure: Single-cylinder engine, cycling
Oil: Nonadditive, high V.I., 1-A

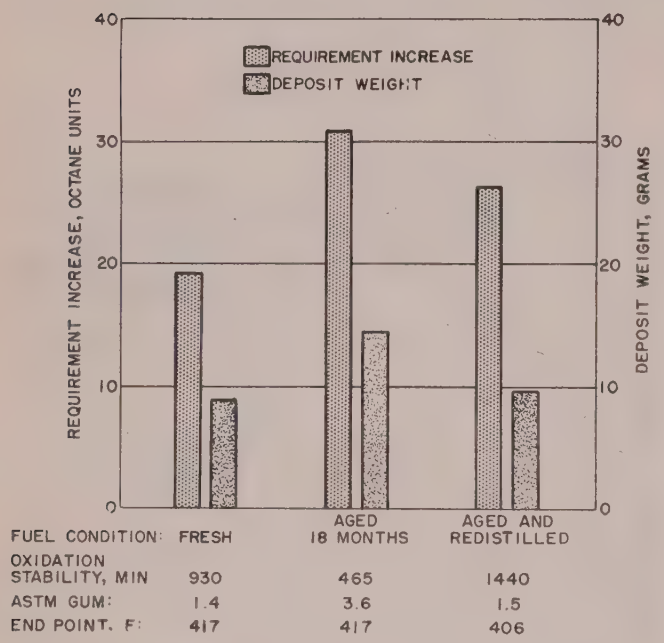


Fig. 11 – Effect of fuel deterioration on requirement increase.
Procedure: Single-cylinder engine, cycling
Fuel: Commercial gasoline, 11-A, 3 ml tel
Oil: Regular grade, low V.I., 4-A

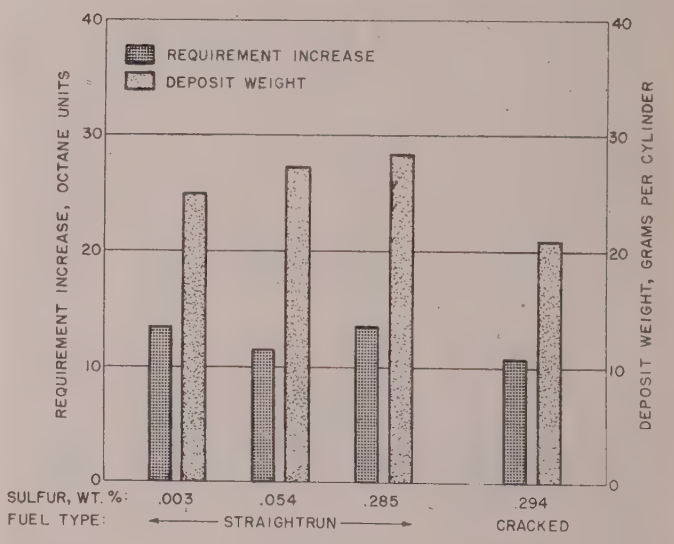


Fig. 12 – Effect of natural sulfur on requirement increase.
Fuels: Contain 3 ml tel
Oil: Premium, high V.I., 18-A
Engine: Passenger car, make D
Operating Conditions: 10,000 miles city-surburban operation

tion is based on weight data, although it appears that requirement increase is related more directly to deposit volume than to deposit weight. It is impractical to express deposit analyses on a volume basis since the form of combination of the elements making up much of the deposit volume is unknown.

Fig. 10 shows the effects on deposit composition of four fuels having widely different carbon/hydrogen ratios. Carbon, total lead compounds, and one of the lead compounds, the oxygenated basic lead halide (PbO•PbCl•Br) are shown as deposit constituents. The deposits from unleaded isooctane and toluene were largely carbon. The small undetermined portions probably consisted mainly of oxygen and hydrogen. Unleaded toluene produced more deposit than unleaded isooctane, but the reverse was true when tel was present. In the presence of tel, fuels highest in carbon/hydrogen ratio produced less deposit and less lead compounds. The oxygenated lead halide appears to have been minimized by fuels of high carbon/hydrogen ratio. Since fuels of high carbon/hydrogen ratio also tend to produce the greatest amount of carbon in the absence of tel, the data suggest that an interaction takes place between fuel-formed carbon and lead compounds which tends to remove both types of materials from the combustion chamber. This effect may result partially from chemical reduction of the high-melting oxygenated lead compounds to other materials of lower melting point. Such an effect would increase the removal of carbon as well as removal of lead compounds from the engine, since the carbon would be oxidized while reducing the lead compounds.

A similar line of reasoning may also explain the relative influence of cycling and constant-speed operation on the effects of tel. Under constant-speed operation, somewhat more complete burning of the fuel would be expected than under cycling conditions, which involve rich-mixture idling and periods of acceleration when more imperfectly vaporized fuel may enter the cylinder. With the resulting formation of additional soot and carbon under cycling conditions, this type of operation would be expected to form more deposits with unleaded fuel and less deposits from leaded fuels than would form in constant-speed operation. Consequently, requirement increase from clear and from leaded fuel would be expected to be more nearly the same under cycling conditions than in constant-speed operation. This reasoning is in agreement with the data shown in Figs. 3 and 4.

The effects of fuel volatility which have been discussed might appear contradictory to the viewpoint that carbon formation in the presence of lead compounds results in mutual scavenging. It must be recognized, however, that the partial burning of fuel produces oxygenated carbonaceous resins in addition to producing sooty carbon. The fact that resinous products can originate from fuel has been well established by many published

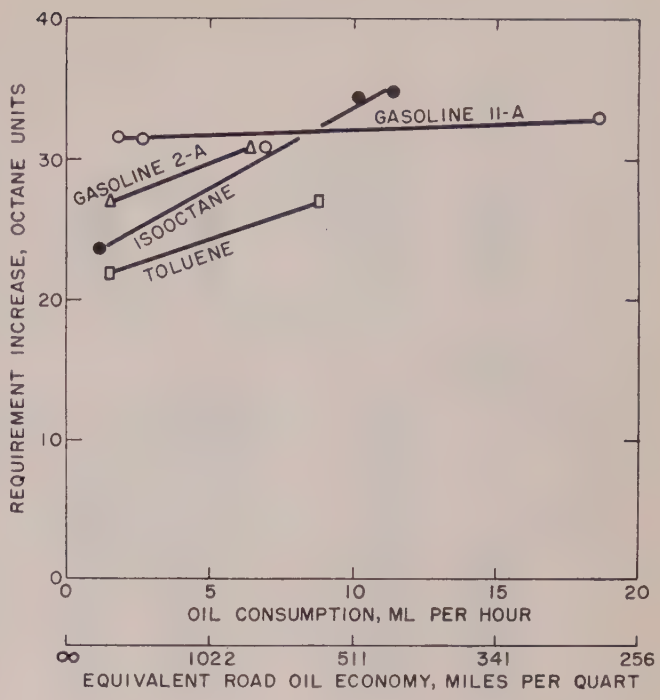


Fig. 13 –Effect of oil consumption rate on requirement increase.
Procedure: Single-cylinder engine, cycling
Fuels: Various, containing 3 ml tel
Oil: Nonadditive, high V.I., 1-A

studies of engine cleanliness, in which engine varnish and sludge were the object. It is believed that the resins contribute to deposit accumulation by providing an adhesive binder and that a balance exists between beneficial effects of sooty materials and harmful effects of the accompanying resins. Such a balance would be influenced by fuel type and volatility as well as by engine operating conditions.

Fuel Deterioration

There are many aspects of fuel behavior which are poorly understood and are fields for additional research. One of these which has been encountered is a change in deposit-forming tendency which accompanied aging of a commercial gasoline in storage. As shown by Fig. 11, the fresh gasoline produced a requirement increase of 19 octane units and 8.9 g deposit. These values increased to 31 octane units and 14.5 g of deposit for the same gasoline after 18 months storage. During this period, the only usual gasoline inspection test showing an appreciable change was oxidation stability, which dropped from 930 to 465 min. Gum content increased only from 1.4 to 3.6 mg. Redistillation of the aged fuel reduced the gum to its original amount, and replacement of inhibitors restored oxidation stability. Requirement increase was reduced to 26 octane units and deposit weight to 9.6 g.

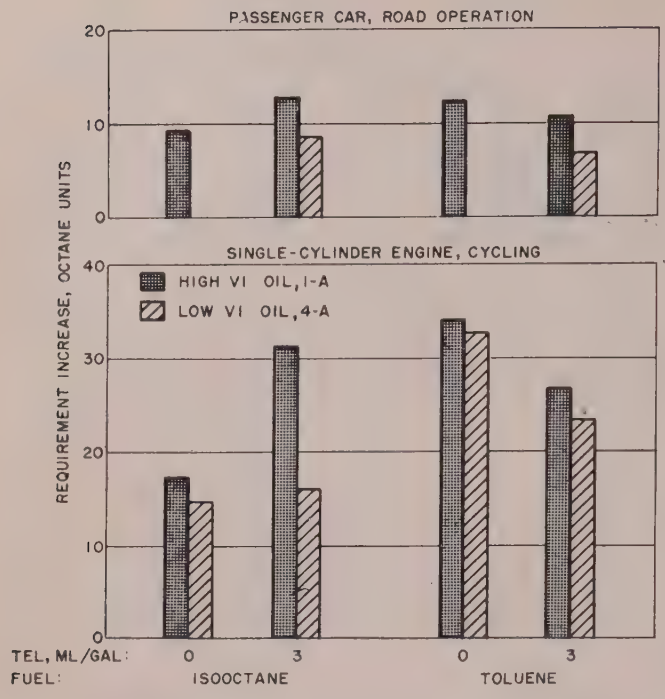


Fig. 14 – Effects of oil type, fuel type, and tel on requirement increase.

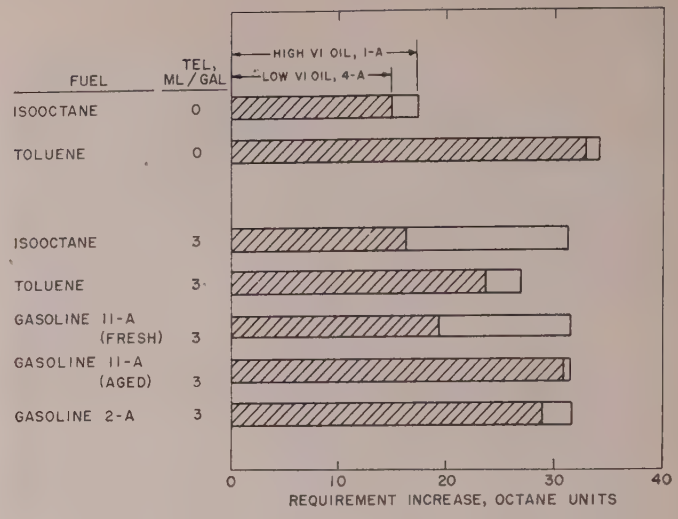


Fig. 15 – Effect of oil type on requirement increase with several fuels.
Procedure: Single-cylinder engine, cycling

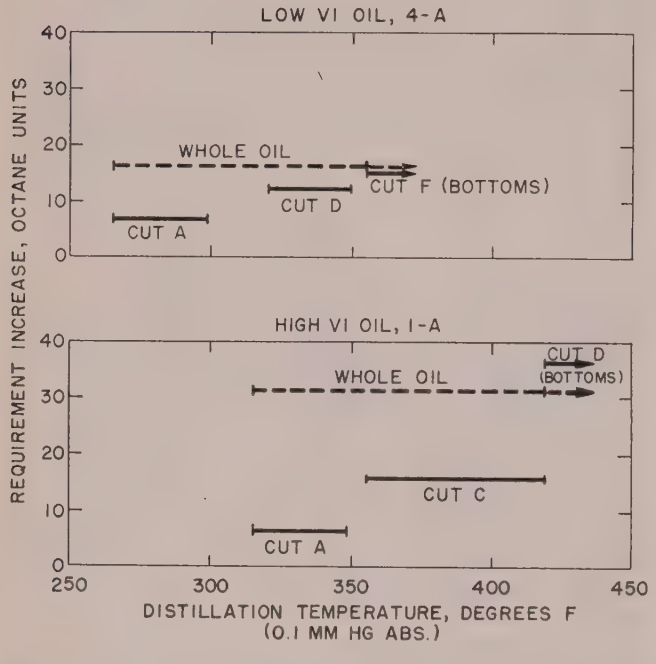


Fig. 16 – Effect of oil volatility and type on requirement increase.
Procedure: Single-cylinder engine, cycling
Fuel: Isooctane, 3 ml tel

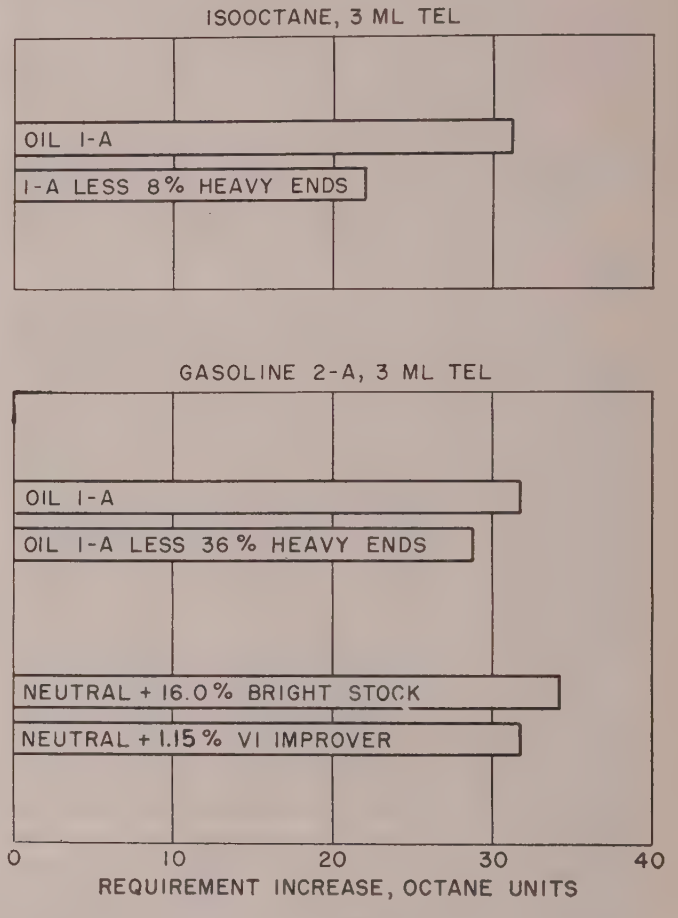


Fig. 17 – Effect of high-boiling constituents of oil on requirement increase.
Procedure: Single-cylinder engine, cycling

Table 3 – Analyses of Reference Oils and Distilled Fractions

	Oil 1-A	Cut A	Cut B	Cut C	Cut D (Bottoms)
Per Cent (vol) of Whole Oil	100	25.0	12.5	25.0	35.5
Distillation Range[a], F	316 – 419+	316 – 344	344 – 358	358 – 419	419+
Gravity, deg API	29.5	30.4	30.5	29.7	27.9
Viscosity, SUS at 100 F	307.0	108.5	138.4	204.6	620.4
SUS at 210 F	53.4	41.1	44.4	48.1	66.4
Viscosity Index	103				
Sulfur, wt %	0.28	0.26	0.22	0.24	0.33
Phosphorus, wt %	0	0	0	0	0
Zinc, wt %	0	0	0	0	0
Per Cent (wt) of Total Carbon as:					
Paraffinic Carbon	65	61	61	63	67
Naphthenic Carbon	27	31	32	30	25
Aromatic Carbon	8	8	7	7	8

	Oil 4-A	Cut A	Cut B	Cut C	Cut D	Cut E	Cut F (Bottoms)
Per Cent (vol) of Whole Oil	100	25.0	12.5	12.5	25.0	4.4	17.2
Distillation Range[a], F	266 – 356+	266 – 293	293 – 307	307 – 324	324 – 351	351 – 356	356+
Gravity, deg API	20.0	21.4	20.7	20.1	19.5	18.5	19.2
Viscosity, SUS at 100 F	475.0	93.7	171.9	268.2	632.3	1180.6	1467.0
SUS at 210 F	52.2	39.0	42.2	49.0	60.0	78.5	102.7
Viscosity Index	10.6						
Sulfur, wt %	0.26	0.21	0.22	0.24	0.25	0.24	0.40
Phosphorus, wt %	0.025	0.012	0.011	0.012	0.016	0.019	0.059
Zinc, wt %	0.014	0	0	0	0	0	0.027
Per Cent (wt) of Total Carbon as:							
Paraffinic Carbon	40	38	39	39	40	40	44
Naphthenic Carbon	41	42	42	43	43	42	38
Aromatic Carbon	19	20	19	18	17	18	18

[a]At 0.080 to 0.140 mm Hg absolute pressure.

Sulfur

Sulfur content is another variable of interest in conjunction with fuel composition. The following data are in general agreement with reports of other investigators that sulfur in gasoline is not a significant factor in passenger-car requirement increase.[2] Four fuels of low and high natural sulfur contents were tested with a high V.I. oil in closely-controlled passenger-car road operation, with maximum speed limited to 50 mph. Three of these fuels were straightrun gasolines. The fourth was a thermally-cracked fuel. Fig. 12 shows that requirement increase was essentially the same for the four gasolines and not related to sulfur content, although the straightrun gasoline lowest in sulfur produced a slightly lower weight of deposit than did the higher sulfur straightruns.

Lubricating Oil Factors

There are a number of characteristics of lubricating oils to be considered in evaluating their overall performance. Included are those affecting oil consumption, engine cleanliness, engine wear, and combustion-chamber deposition. Most of the published information on the combustion-chamber deposition characteristics of oils has been concerned primarily with deposit weight rather than requirement increase. The work reported herein represents an attempt to determine, from the standpoint of requirement increase:

1. The effect of oil-consumption rate.
2. The interrelationship of fuel and oil factors.
3. The characteristics of an oil which are desirable.

Oil-consumption rate has long been recognized as an important factor in the formation of combustion-chamber deposits. Its effects on requirement increase is illustrated in Fig. 13. While these data are by no means as extensive as desired, they do show that oil consumption is an important variable and that its effect varies with the fuel used in the engine. All of these fuels contained 3.0 ml tel per gal, and oil-consumption rate was varied by changing piston ring combinations. Maximum effect was observed with isooctane as a fuel. Less effect was shown for toluene and fuel 2-A while there was substantially no effect with fuel 11-A. These results illustrate the importance of maintaining fairly constant oil consumption from test to test when evaluating various factors in this type of work.

Some variations in oil-consumption rate were encountered in the single-cylinder engine while effects of other oil factors were being investigated. However, the data which are to be presented have been corrected to a common oil-consumption rate, and are believed to be comparable.

Oil-Fuel Relationship

Two typical oils of SAE 20 classification and widely different in characteristics were selected for reference purposes for this research program. Inspection data on these two oils are presented in Table 3.

The effect of tel in different types of hydrocarbons on requirement increase in the single-cylinder L-head engine cycling test procedure using the high V.I. Oil 1-A has already been discussed. Similar tests were made using the low V.I. Oil 4-A and the results are presented in the lower portion of Fig. 14. (The deposit weights corresponding to these values were presented in Fig. 9.) Oil 4-A reduced requirement increase only slightly below

that obtained with the oil 1-A when the fuels were clear isooctane and clear toluene. Oil 4-A had a small beneficial effect when used with the leaded toluene and a large effect with the leaded isooctane. Most of these oil-fuel combinations also were run in a 1949 V-8 passenger car operated under a carefully controlled, light-duty city-suburban driving schedule with a maximum speed of 50 mph. The data are presented in the upper portion of Fig. 14, and show reasonable qualitative agreement with the single-cylinder L-head engine test results. These results, with those shown in Fig. 3, are evidence of the significance and reliability of the single-cylinder cycling test procedure as a laboratory tool for this type of work.

The laboratory engine data on isooctane and toluene are included with data on two commercial fuel blends in Fig. 15 in order to emphasize further the point that the gains to be obtained by going from oil 1-A to oil 4-A were dependent to a large extent on the fuel used. Organic materials originating from both fuel and oil act as binders for lead compounds. If the fuel is the source of a large amount of organic materials, the contribution of the oil will be proportionally less important. The reverse is equally true. Further supporting this statement are the two sets of data for fuel 11-A, fresh and after aging. As was indicated in the discussion of Fig. 11, progressive aging of fuel 11-A greatly increased the deposit effects when tested with oil 4-A. Not shown in Fig. 11 were the data based on tests with oil 1-A, which gave constantly high requirement increase regardless of the aging of the fuel. The data presented in these two figures (11 and 14) emphasize the vital part

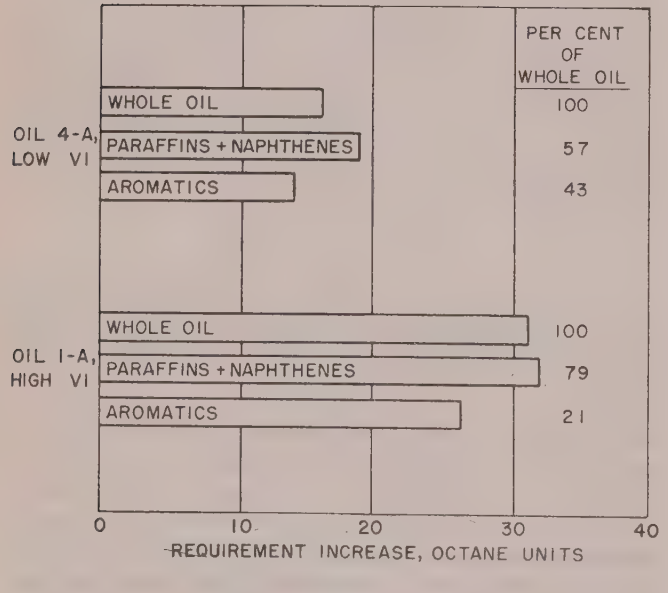


Fig. 18 – Effect of chemical components of oils on requirement increase.
Procedure: Single-cylinder engine, cycling
Fuel: Isooctane, 3 ml tel

that the fuel has in the evaluation of a lubricating oil. It is possible to emphasize or de-emphasize the deposit-forming tendencies of certain lubricating oils by the characteristics of the fuel used.

The foregoing data have shown marked differences in requirement increase, between oils 1-A and 4-A, both in the laboratory and on the road, particularly when a clean-burning fuel such as isooctane was used. The main differences between these two oils are that oil 4-A, in common with most oils of low V.I., is more volatile and contains a higher percentage of aromatics and naphthenes than oil 1-A, which is a typical high V.I. oil. The question then arises as to whether the beneficial effect of oil 4-A was due mainly to its higher volatility or to its different hydrocarbon composition. The investigation of these factors is described in the following sections.

Oil Volatility

The first step in the investigation of oil volatility was to vacuum-distill each of the reference oils, 1-A and 4-A, into fractions. Laboratory engine tests then were run on several of the fractions using leaded isooctane as the fuel. The requirement-increase results are plotted against distillation temperature in Fig. 16. Not all of the fractions were used in this investigation, hence there are gaps shown in the distillation temperature, and the fraction letters are not consecutive. As previously noted, requirement increase with whole oil 1-A was greater than with the more volatile oil 4-A. The distilled fractions from both oils gave progressively higher requirement increases as the distillation temperatures increased. Cut D (bottoms) of oil 1-A was the least volatile, boiling above 419 F at 0.1 mm Hg absolute pressure, and gave the highest requirement increase (36 octane units), which was 5 units above that obtained with the whole oil. In contrast, cut F (bottoms) of 4-A, boiled above 356 F and gave a requirement increase of 15 units, approximately the same as the whole oil. It is believed that the variation in requirement increase with the different distilled cuts was due primarily to changes in oil volatility, since the hydrocarbon composition of the cuts, as measured by ring analysis and shown in Table 3, was fairly consistent throughout the distillation range of each whole oil. These data emphasize the fact that requirement increase can be reduced to a very low level by use of a clean-burning leaded fuel with a volatile lubricating oil.

The effect of oil volatility on requirement increase has been investigated in other tests. The majority of the commercial lubricating oils on the market today contain bright stock, used as blending material for the control of physical properties such as the viscosity of the finished oil. Oil 1-A was a blend of bright stocks and neutrals, and since cut D (bottoms), comprising 36% by volume of the whole oil gave such a large requirement increase, it appeared desirable to determine the

effect of removing only the least volatile portion of the heavy ends of the whole oil. Accordingly a heavy fraction, amounting to 8% of the whole oil, was removed by means of a molecular still. This reduced requirement increase by 30%. These results are presented in the upper portion of Fig. 17.

The effects of oil volatility so far discussed were observed in conjunction with leaded isooctane which is believed to contribute little to the resinous portion of deposits. Commercial gasolines, however, generally form an appreciable amount of resinous materials in the combustion chamber. These materials in conjunction with resinous material from the lubricating oil, act as a binder for other combustion products including lead compounds. In order to determine the effect of oil volatility on requirement increase when using a commercial gasoline, two pairs of tests were run with fuel 2-A, the results of which are shown in the lower portion of Fig. 17. The first pair of tests showed that even though 36% of the heavy ends of oil 1-A were removed, the reduction in octane requirement increase amounted to only 3 units with the commercial fuel in contrast to the 9 units obtained with isooctane when only 8% of the heavy ends of the same oil were removed.

The second pair of tests were run with oils similar to those described in a recent paper[7] which pointed out that deposit weight was reduced appreciably by eliminating the bright stock from oils and substituting certain V.I. improvers for control of viscosity. One of these oils was a blend of light solvent-refined neutral stock having a viscosity of 43 Saybolt sec at 210 F plus 16% of a distillate bright stock having a viscosity of 78 Saybolt sec at 210 F. The other oil was a blend of the same neutral stock plus 1.15% of a commercial V.I. improver. Both oil blends had a viscosity of 45 Saybolt sec at 210 F and contained 1% of a commercial oxidation inhibitor. The viscosity indexes of the two blends were 101 and 120, respectively. The substitution of the V.I. improver for the distillate bright stock lowered requirement increase from 35 to 32 units. This improvement was thus of the same order as that obtained by removing the less volatile portions of oil 1-A.

The foregoing test data, obtained in the laboratory engine, indicate that the bulk of the detrimental materials originating from the lubricating oil is formed from the heavy ends. When leaded isooctane was used as a fuel, substantial reduction in requirement increase were obtained by making the oil more volatile. The reduction was much less, however, with a leaded commercial gasoline.

Oil Composition

In the preceding section, the comparatively large effects of oil volatility were discussed and there

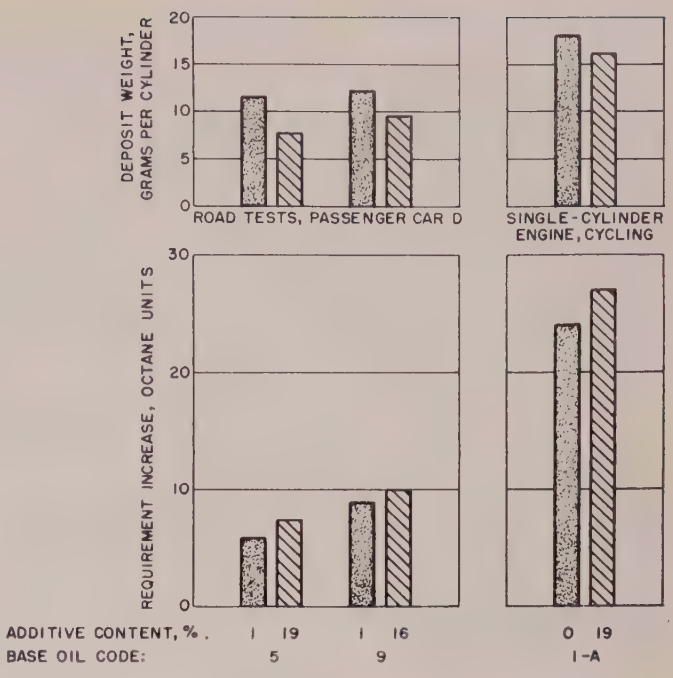


Fig. 19 – Effect of oil additive concentration on requirement increase. Fuel: Commercial base stock with 3 ml tel

was some indication that it was a more important factor than hydrocarbon composition. Consequently, the effect of composition was investigated by tests on fractions of oils separated according to hydrocarbon type by solvent extraction, thermal diffusion, and by silica gel chromatography. Tests of paraffin-naphthene fractions and aromatic fractions separated from oils 4-A and 1-A indicate, as shown in Fig. 18, that with leaded isooctane the paraffin-naphthene fractions gave slightly greater requirement increase and the aromatic fractions gave slightly less than the respective whole oils. It is expected that these effects would be even smaller with commercial gasolines. Various blends of the different hydrocarbon type fractions were tested and, in general, it was found that the requirement increases were not greatly different from the weighted average of the effects obtained with the individual constituents.

These results, in conjunction with those in the previous section, indicate that oil volatility is a more important factor than hydrocarbon composition insofar as the contribution of lubricating oil to requirement increase is concerned.

Effect of High Additive Content

The effects of high additive content, as in Series 2 oils, were investigated in two pairs of tests in passenger-car operation on the road and in one pair in the laboratory cycling test procedure. The road tests were conducted in a 1949 Car D under controlled light-duty driving conditions with a top

[7] See General Papers, presented before the Division of Petroleum Chemistry of the American Chemical Society, Vol. 28, 1952, pp. 25-47: "Combustion Chamber Deposits as Related to the Carbon-Forming Properties of Motor Oils," by J. D. Bartleson and E. C. Hughes.

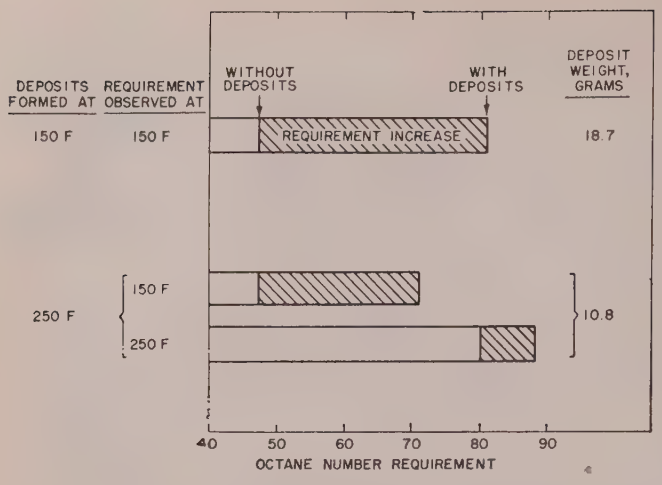


Fig. 20 – Effect of coolant temperature on requirement and requirement increase.

Procedure: Single-cylinder engine, cycling
Fuel: Gasoline 2-A, 3 ml tel
Oil: Nonadditive, high V.I., 1-A

speed of 50 mph, and were run for 10,000 miles. Each pair of tests compared a given base oil containing 1% of an antioxidant type additive with the same base oil containing 16 or 19% of a Series 2 additive. The data are presented in Fig. 19 and show that in the paired tests the high additive blends produced slightly higher requirement increase. In all cases, the high additive blends resulted in less deposits by weight than were obtained with the low additive-content blends although deposit volumes were about constant. From these and similar results which have been reported, it appears that commercial oil additives as used in present-day oils are not a major factor in the requirement increase problem.

Engine and Operating Factors

Engine design and operating conditions are other factors which influence the magnitude of the deposit problem. While some changes in operating conditions can be shown to have a major effect in lowering requirement increase, they do not offer a practical means of alleviating the problem in service. One example of this is the effect of the spark advance used during deposit accumulation. A pair of tests was run with leaded isooctane and oil 1-A in the single-cylinder L-head engine cycling test procedure, at spark timings of top dead center (standard) and 15 deg before top center (maximum power). The test run at the more advanced timing resulted in the formation of less deposits by weight, and less requirement increase over the entire range of ignition timings. It is believed that the deposit modification and reduced deposition at advanced timing results from higher combustion-chamber temperatures and pressures which tend to minimize deposit build-up. Since the major portion of most multicylinder-engine operation is under part-throttle conditions with the ignition timing fairly well advanced, this effect is probably not of much significance from a practical standpoint.

Table 4 – Effect of Operating Conditions on Requirement Increase and Deposit Weight in Engine D

Fuel Tel Content, ml per gal	Requirement Increase, Due to Deposits, octane units		Deposit Weight, g per cyl	
	Passenger Cars[a]	Trucks[b]	Passenger Cars[a]	Trucks[b]
0	8.1	3.5	3.5	7.0
1.0	9.7	...	8.7	....
1.5	....	4.5	....	17.0
3.0	9.5	4.0	11.0	14.8

[a] Average for 7 cars at each tel concentration.
[b] Average for 4 trucks at each tel concentration.

Jacket-Coolant Temperature

Use of higher temperatures is frequently proposed as a means of reducing combustion-chamber deposition. This factor has been investigated in single-cylinder engines using a leaded commercial gasoline and the high V.I. reference oil. Fig. 20 compares the effects of deposits accumulated at 150 F and 250 F. Requirement observations on the deposits accumulated at 250 F were made at both 250 F and 150 F. It can be observed that use of the higher jacket temperature reduced both the weight of deposits and the effects on requirement increase, when measured at both temperatures. However, the requirement level was raised by the increased temperature more than the deposit effects were reduced. As a result, the increased temperature caused a substantial increase in fuel requirement both with and without deposits. Further increase in jacket-coolant temperature to 350 F resulted in slightly higher fuel requirements and in slightly smaller deposit effects than were observed at 250 F.

Light-Duty versus Heavy-Duty Engine Operation

It is generally recognized that light-duty city-type driving conditions promote the formation of deposits which result in relatively high requirement increases. Data were presented in Table 1 which showed the effect of full-throttle and high-speed operation in reducing the effects of deposits previously formed under mild conditions. Table 4 presents a comparison of typical deposit effects in normal passenger-car operation and heavy-duty truck service, using the same make of engine. The heavy-duty truck operation resulted in approximately twice as much deposit by weight as obtained in the passenger-car fleet test, and yet the effect of these heavier deposits on requirement increase was less than half as much. This is still another indication that requirement increase and deposit weight have no direct relation. Deposit analyses indicated that the heavier deposits from truck operation contained little organic matter while

those from passenger-car operation contained several per cent. Half of the requirement increase in the trucks was chargeable to increase in compression ratio while less than a quarter of that in the passenger cars would be accounted for on that basis.

Oil Consumption

The data which were presented in Fig. 13 indicated that maximum requirement increase with several fuels is a direct function of oil consumption under mild operating conditions, and illustrate the importance of maintaining low oil consumption rates if deposit effects are to be minimized. High oil consumption also accelerate the rate of requirement increase, particularly in the early stages. As a result, combinations of piston ring and cylinder bore, which require long break-in periods before effective oil control is reached, aggravate the deposit problem. Engine designs which require a short break-in and then provide the minimum oil consumption rate consistent with good lubrication are, of course, highly desirable. Likewise, oil additives which reduce wear and maintain a high level of cleanliness in the ring zone are also desirable.

Combustion-Chamber Design

Combustion-chamber design is one of the factors that affect the magnitude of requirement increase due to deposits. Early in this paper it was indicated that requirement increase was believed due to the combined effects of increase in compression ratio and heating of the charge. On this basis, it would appear that the amount of combustion-chamber surface area to which deposits could adhere might be an important factor in the requirement increase problem. If surface area is an indication of the amount of deposition to be expected and cylinder displacement is an indication of the amount of intake charge, a general relationship should exist between requirement increase and the ratio of surface area to displacement. Since the surface-to-displacement ratio is minimized by making a more compact combustion chamber, and also by increasing the displacement per cylinder, it might be expected that those engines with a minimum value for this ratio would suffer minimum requirement increase. Fig. 21 shows such a general relationship for a number of engines. The requirement increase is plotted against the ratio of combustion-chamber surface area in sq in. to cylinder displacement in cu in. The points connected by straight lines represent test results obtained on different engines with the same fuel and lubricant and similar operating conditions. The single-cylinder CFR engine was run with both the standard valve-in-head chamber and an L-head chamber; the latter has been used for most of the test work reported in this paper. The passenger cars were run in normal city-suburban service, and the tractor engines were run on the dynamometer for 100 hr at governed speed and 75% load. There appears to be a general trend of higher requirement increase as the ratio of surface area to cylinder displacement is increased. The L-head engines, as compared with valve-in-head engines, have a higher surface-to-displacement ratio and usually have higher requirement increases. The foregoing suggests that combustion chambers be designed to provide the minimum ratio of surface area to cylinder displacement consistent with other design considerations.

Factors other than surface-to-displacement ratio also are involved. One of these is the relative amount of deposits located in the end-gas region of the combustion chamber. Deposits were selectively removed from different portions of the combustion chamber of the single-cylinder L-head engine and changes in requirement were observed. The average results of five tests showed that deposits in the end-gas half of the combustion chamber, comprising 45% of the total weight, caused 60% of the requirement increase. These results confirm published data on multicylinder engines,[4] wherein it was reported that deposits removed from the end-gas region constituted only 25 to 35%

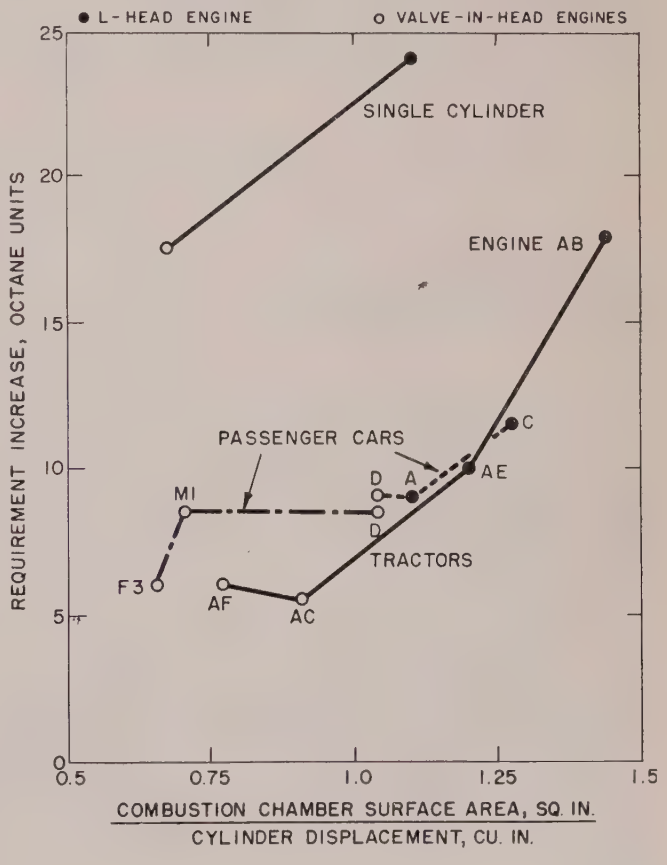


Fig. 21 – Effect of ratio of combustion chamber surface area to cylinder displacement on requirement increase. Connected points represent tests with same fuel and oil under similar operating conditions.

of the total weight and yet were responsible for approximately 50% of the requirement increase.

Summary

An investigation of the effects of fuels, oils, tel, and engine variables on requirement increase has been made. This problem warrants extensive research. Elimination of the effects of combustion-chamber deposits, which increase the requirement of passenger cars by an average of 10 octane units, would permit engine compression ratios to be increased by approximately one ratio.

1. A cycling test procedure for a single-cylinder laboratory engine has been developed, which evaluates the effects of various fuel and oil factors on requirement increase. This procedure gives reasonable qualitative correlation with results obtained in passenger cars operated in light-duty, city-suburban service, in which deposit effects are most severe.

2. Organic materials originating from both fuel and oil are of major importance in deposit formation. Consequently, fuel and oil effects are so interrelated that either variable should be studied only with careful consideration of the effects of the other.

3. Tests with individual leaded hydrocarbons as fuels show some major differences, but no one type is superior over a wide boiling range. In full-boiling, leaded gasolines covering a wide range in hydrocarbon type proportions, no distinct relationships between hydrocarbon composition and requirement increase have been found. However, with both hydrocarbons and gasolines there is a trend toward smaller requirement increase as fuel end-point is reduced.

4. Addition of tel to pure hydrocarbons has been found to make requirement increase either larger or smaller depending on the type of individual hydrocarbon to which it is added. Lubricating oil can influence the magnitude of these effects.

5. The carbon-hydrogen ratio of leaded hydrocarbons can influence the amount and composition of the deposit formed. High carbon-hydrogen ratios appear to be favorable.

6. Removal of heavy ends from lubricating oils tends to lessen requirement increase. The magnitude of this effect is dependent on the contribution of the fuel to the organic portion of the deposit. Variations in oil hydrocarbon composition appear to be less important than changes in oil volatility.

7. Engine operating conditions have a major influence on deposit effects. Requirement increase can be reduced by high-output operation during deposit accumulation or following light-duty service. Increasing the jacket-coolant temperature reduces deposit effects, although the net effect is to raise the requirement level.

8. Combustion-chamber design and deposit location appear to influence requirement increase. A small ratio of combustion-chamber surface area to displacement volume appears to be desirable.

DISCUSSION

Autoignition and Knock Seen as Combined Problem

– J. G. Moxey, Jr.
Sun Oil Co.

I HAVE just two points that I would like to discuss in connection with this paper.

First of all, I think it is an excellent example of the progress that has been made with this problem of combustion-chamber deposits. There has been a tremendous amount of work done on the subject over the last three or four years, and it seems to me that we have gone through two distinct stages.

Back in the beginning, we were all, quite naturally, interested in our own commercial products. Most of the early work was aimed at finding out just how much of a problem deposits were in actual service with commercially available fuels and lubricants. It was pretty well proved, I think, that they were a serious problem, although at the same time, it was pretty well proved that the bulk of the commercially available fuels and lubricants didn't differ greatly in their tendency to produce deposits. I think the 10 octane number figure quoted in the paper is a pretty good rough measure of the order of magnitude of the problem.

More recently, we have graduated to a second stage in our progress. We have got away from simply trying to evaluate our commercial products, and we're trying to take things apart and find out if there isn't something we can do about the situation. This paper is an excellent example of the work that's being done now, attempting to isolate the different variables that are contributing to deposits. It points up very nicely the interrelation between the various fuel and lubricant factors, and the difficulty involved in trying to isolate the effect of any one factor.

It seems to me that it is only through detailed studies, such as those described in this paper, of the individual effects of the various contributors to requirement increase that we're going to make any real progress toward solving the problem.

The second point I would like to make is on the subject of autoignition which was not discussed in the paper except to point out that it was an additional problem superimposed on top of the problem of knock.

Most of the work that has been done so far on combustion-chamber deposits has been concerned with ordinary knock that results from compression of the end gas by a normal, spark-initiated, flame front. However, I think that most of the knock that we're getting in actual service these days is knock that is complicated by the occurrence of autoignition. And I also think that combustion chamber deposits are, in most cases, serving as the source of the autoignition. Any flame front that is set off in the combustion chamber, other than that initiated by the normal spark, results in more rapid compression of the end gas which means, of course, more knock.

It seems to me that, as we progress with our studies of

combustion-chamber deposits, it's going to be more and more important that we try to look at this combined problem of auto ignition and knock rather than at the problem of knock alone.

Laboratory Tests Clarify Lubricating Oil Data

– R. L. Overcash
Kendall Refining Co.

WE believe that the authors have presented an excellent paper covering a great many of the phases concerned with this very complex problem. Experimental work in our laboratory verifies many of the conclusions reached as a result of the work on lubricating oil factors. We have some comments to make that might stimulate further thought and might serve to clarify further certain phases of the work done.

In order to better understand the relative importance of volume and heat effects of deposits as well as catalytic effects, could the authors establish the magnitude of the octane requirement increase due to heating effects of the deposits. The average value for the increase due to deposits in passenger cars has been quoted as about 10 octane units. The increase in compression ratio of 0.3 units has been reported as equivalent to an increase in requirement of about 2 octane numbers. If an average octane number increase for the heating effects can be given, then the difference between the sum of the volume and heating effects and the overall increase could possibly be ascribed to what is called a catalytic effect.

Under the heading engine and operation factors the net results of higher combustion-chamber temperatures and pressures is stated by the authors to be the minimization of deposit build-up. If higher combustion-chamber temperatures accompany deposit build-up, and equilibrium is reached after these temperatures become high enough to burn away deposits as rapidly as they are formed, then engine design for higher combustion-chamber temperatures with clean engines might somewhat alleviate the problem of deposit build-up. If equilibrium is reached with less deposits, due to higher combustion-chamber temperatures, then there would certainly be less octane requirement increase due to change in compression ratio. Is there any way that the heat transfer from the combustion chamber with new engines could be reduced in order to speed up the burning away of deposits which are formed in the first few thousand miles in any engine? Could this be accomplished by use of some insulating material on the cylinder head surface and top of the piston that would greatly reduce the transfer of heat? This would tend to cause the octane requirement of the engine to reach equilibrium much earlier in its useful life and perhaps it would increase in octane requirement to a smaller degree.

Under the deposit evaluation phase the authors stated that all data in this paper are based on observations before and after deposit removal. The practice in some laboratories is to use an average of the clean requirement at the beginning and at the end of the test to compare with the requirement with deposits. This practice can be used where significant mechanical changes do not occur. Perhaps tests should not be relied upon where large changes do occur. The averaging of two clean requirement results adds to the precision of the overall test results.

It is noted that all of the data presented have been corrected to a common oil consumption rate. It would be interesting to learn how this correction has been made and how large the correction was in comparison to the overall octane requirement increase. If the correction has been based on the data presented in Fig. 13 then we believe that erroneous conclusions might have been reached by applying such corrections to the consumption rates found for all tests. For instance, an increase in oil consumption due to increased volatility of the lubricating oil might not have the same detrimental effect as when oil consumption was increased by different ring arrangements used to obtain the data in Fig. 13.

In the discussion of the effects of oil volatility, a pair of tests were reported on two oils. One oil contained a neutral stock having a viscosity of 43 Saybolt sec at 210F and 16% of distillate bright stock. The other oil contained the same neutral stock plus 2.3% of a commercial V.I. improver. Is it possible that the small difference found upon the substitution of the V.I. improver for the distillate bright stock might be due to the contribution of the 43 Saybolt sec neutral. Is such a neutral volatile enough to minimize the effect of lubricating oil on combustion-chamber deposits?

Authors' Closure To Discussion

MR. OVERCASH has raised a number of interesting questions regarding the deposit problem. One of these concerned the relative magnitude of the heating and volume effects of deposits. It was pointed out in the paper that deposits increase requirement for ordinary knock largely because of these two effects. Catalytic effects, if they do exist, appear to be relatively minor. Our work has indicated that the volume effects in raising the compression ratio of the engine account for approximately 20% of the requirement increase, and heating effects account for the remainder.

The question has been raised regarding the possibility of minimizing deposit buildup by use of higher combustion-chamber temperatures and insulating coatings intended to raise surface temperatures. In connection with this, the paper pointed out that the use of higher cooling jacket temperature was found to reduce deposition but also to increase the requirement level of the engine, both "clean" and with equilibrium deposits. Our experience with insulating coatings has been similar, in that deposit buildup was reduced but equilibrium requirement was not lowered.

With regard to the comments on oil consumption, we have found that the rate of oil consumption affects requirement increase with many oil-fuel combinations. Moreover, it is quite difficult to maintain constant oil consumption rates from test to test even in the same engine. Because of this, it was found desirable to establish curves of requirement increase versus oil consumption for the several reference oil-fuel combinations. The data obtained with a test oil-fuel combination could then be compared with the reference oil-fuel combination at the same oil consumption. Likewise, a series of test oil-fuel combinations could be compared at the same oil consumption rate if the relationships of requirement increase to oil consumption rate were determined. In most of the tests reported in the paper, the corrections for variations in oil consumption rates were relatively minor. This method of handling the oil consumption variable is a refinement of test technique which is applicable mainly to single-cylinder engines.

The question was raised as to whether the neutral oil stock of 43 Saybolt sec viscosity at 210 F, used in the tests shown in Fig. 17, was sufficiently volatile to minimize the effect of lubricating oil on combustion-chamber deposits. Although some beneficial effect was observed, the work on distilled oil fractions, presented in Fig. 16 and Table 3, indicates that even larger gains could have been made had the neutral oil been even more volatile. It should be mentioned, however, that different base-oil types of the same viscosity can have different volatilities, as indicated in Table 3. Because of this, the deposit-forming tendencies of different base-oil types are more directly related to their volatilities than to their viscosities.

LUBRICANTS —

THE lubricants used to protect metal surfaces against wear are engineering materials in the same sense as the metals constituting the surfaces. If the wear characteristics of automotive parts can be influenced by varying the metals used, they can also be influenced by varying the lubricants used. In this paper the present state of the art regarding the formulation of engine, transmission,

THE present state of the art regarding the formulation of automotive and aviation lubricants from the standpoint of protecting metal parts against mechanical and chemical wear is outlined in this paper. The discussion is limited to four classes of lubricants, namely, (a) engine oils, (b) transmission oils, (c) rear-axle oils, and (d) greases.

By virtue of the wide variety of service conditions to which they are subjected, engine parts can be worn as the result of either mechanical or chemical action. Engine oils must be designed, therefore, to cope with both these types of wear. Oil viscosity and viscosity index are both important factors in engine wear. Various types of additives are being used to an increasing degree as "alloying" materials in engine oils to control their wear characteristics. A trend toward the use of specially synthesized lubricants for both aviation and automotive service is in evidence.

In gear lubrication the pressure-viscosity characteristics of the lubricating oils may be important in preventing surface failure. In low-temperature operations low-viscosity, high-viscosity-index gear lubricants perform satisfactorily and, in addition, show increased efficiency at normal operating temperatures. Fluids for automatic transmissions must permit good friction coefficients between steel and the friction-band materials to achieve smooth operation of the unit.

For hypoid-gear service mineral oil must be fortified with additives which will react chemically with the gear surfaces to form solid film lubricants. In grease-lubricated automotive mechanisms fretting and means of alleviating this condition constitute the major problem.

THE SURFACE SAVERS

J. F. Kunc, Jr., and J. P. Hamer,
Esso Laboratories, Standard Oil Development Co.

This paper was presented at the SAE Annual Meeting, Detroit, Jan. 13, 1953.

and rear-axle lubricants, as well as greases, will be reviewed from the standpoint of wear prevention.

Small changes in the manner in which metals are alloyed can have rather profound effects on their wear properties. Similar effects can be obtained by "alloying" lubricants through the inclusion of relatively small amounts of the right kinds of additives. The types of additives that might be good for one application, however, will not necessarily be satisfactory for another. For example, the types of additives needed to make a satisfactory lubricant for hypoid axles are entirely unsuitable for use in crankcase oils. Each case must be considered on its own merits.

Lubricants cannot be expected to work miracles in wear prevention. Thus, lubricants cannot overcome deficiencies in the design or fabrication of operating parts. Neither should they be expected to protect these parts against the ravages of poor maintenance practices.

Much research has been done and much more is being done in an effort to obtain a better understanding of the mechanisms by which wear takes place and the role played by lubricants in preventing wear. Much has been learned, but much confusion and divergence of opinion still exists.

This paper will not review the theories which have been put forth to explain how lubricants function in preventing wear. However, for purposes of orientation, it is necessary to classify the types of wear which are encountered in dealing with automotive parts.

Broadly, wear falls into two categories, namely, mechanical or chemical. Mechanical wear is defined as loss of metal resulting from the physical rubbing of two mating surfaces or that resulting from the interjection of solid foreign particles between the two mating surfaces. Mechanical wear as herein defined is also commonly designated as frictional or abrasive wear. Chemical wear is defined as loss of metal resulting from chemical attack of the mating surfaces by water, acids, and so forth, which may contact the surfaces. Chemical wear might also be defined as corrosive wear or simply corrosion. In the case of mechanical wear, metal is removed in its original chemical state by tearing or grinding high spots or asperities off the contacting surfaces. In chemical wear the metal is first converted to a new chemical form (for example, iron is converted to iron oxide) which is readily rubbed off, thus reexposing the surface for further attack. In practice, mechanical and chemical wear are often encountered simultaneously. In those cases where both types do occur, it is usually difficult (if not impossible) to make a clear-cut distinction between them or to determine which occurred first. A good example of this is the controversy which exists over the phenomenon of fretting or fretting corrosion. When fretting involving ferrous metal parts occurs, rust is usually in evidence. One school maintains that the metal is removed in its original form by the physical rubbing of the parts, and the tiny particles of metal removed then rust very rapidly. The other school, however, insists that the metal surfaces rust first and the rust is then removed by physical rubbing.

Engine Lubricants

The oils used for the lubrication of internal-combustion engines, regardless of whether they be

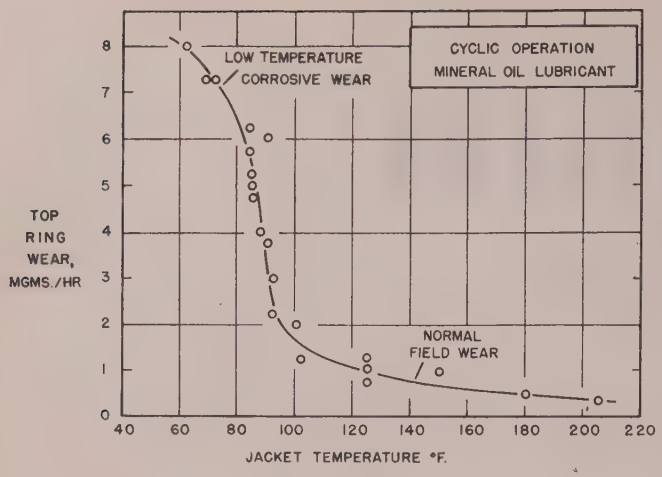


Fig. 1 – Effect of jacket temperature on Lauson engine wear

otto or diesel cycle and designed for aviation, automotive, or stationary service, are subjected to a very wide and diversified range of temperatures and service conditions.

The internal-combustion engine is the only prime mover in which combustion takes place in the same mechanism that converts a pressure and volume change into power. Here, then, is a unique case in which a lubricant is called upon to do a number of important jobs simultaneously. To wit, the lubricant must act as a heat transfer medium, it must provide a film of adequate thickness to prevent physical contact of metal parts, it must be capable of protecting all of the engine parts against chemical attack by the potentially corrosive products formed either by combustion of the fuel or oxidation of the lubricant itself, and it must act as a repository or catch basin for all of the solid and liquid contaminants finding their way to the lubricated areas.

Factors Influencing Wear – Although all of the rubbing, sliding, or rotating parts of an internal-combustion engine are subject to wear, the most critical parts in reciprocating engines are the piston rings and cylinder walls. The rate of piston-ring and cylinder-liner wear occurring in an engine depends on a large number of factors other than the kind of lubricant supplied to it. Some of these are: (1) the materials of construction used; (2) the design of the piston rings and cylinder liners; (3) the conditions under which the engine is operated; (4) the general design of the engine's cooling system and the rapidity with which the engine is warmed up; (5) the type and quantity of abrasive materials finding their way into the crankcase oil; (6) the frequency with which the lubricating oil is replaced; and (7) the quality of the fuel used. In view of the multiplicity of factors involved, it is readily apparent that any sweeping generalizations made regarding engine wear and the influence of lubricants on engine wear must be viewed with considerable caution.

Not all of these factors will be discussed here. It is important, however, to recognize the tremendous influence of operating temperatures, not only on the rate but also in connection with the type of wear which takes place on the cylinder walls and piston rings of an engine. The data presented in Fig. 1 illustrate this effect. In this figure the rate of piston-ring wear observed in a single-cylinder gasoline engine is plotted as a function of the jacket temperature at which the engine was operated. It will be noted that the wear rate jumps sharply as the jacket temperature is reduced below 120 F. For example, the rate at 60 F is 4 times that at 90 F and 8 times as high as that at 120 F. The temperature at the inflection point of the curve closely approximates the dewpoint of the combustion gases which is in the range of 130 to 135 F. When the jacket temperature of an engine is below the dewpoint of the gases, water and other potentially corrosive materials are condensed on the cylinder walls and piston rings, leading to chemical or corrosive wear. When the jacket temperature is above the dewpoint, this does not occur, and the wear appears to be primarily mechanical in nature.

Microscopic examination of cylinder walls and piston rings taken from engines operated at various jacket temperatures, both in the field and in the laboratory, support this viewpoint. Parts from engines operated at low jacket temperatures show the pitting and pockmarking typical of corrosive attack, while those from engines operated at jacket temperatures above the dewpoint show only the scratches and surface damage typical of mechanical rubbing.

How Engines Wear in the Field – It is thus evident that it is possible to have both mechanical and chemical wear occurring in engines. There is considerable difference of opinion as to which is the more important factor, and the role which lubricants play in controlling mechanical and chemical wear in engines. Of well over 100 gasoline engines brought into garages for overhaul and carefully examined by the Esso laboratories, evidence of chemical wear (corrosion) was observed in only 10%, whereas the remaining 90% appeared to have been mechanically worn by either friction or abrasion. This indicates rather strongly that mechanical wear is the more important factor controlling the wear rate of internal-combustion engines.

In designing lubricating oils, however, provisions must be made for protecting the engine parts against both chemical and mechanical wear. During recent years considerable improvement has been made in the wear characteristics of engine oils. This improvement has been brought about almost entirely by the inclusion of relatively small amounts of additives rather than through changes

in the refining techniques applied to the petroleum base stocks used in the formulation of the oils. The role of additives in controlling engine wear will be discussed more fully later.

Role of Viscosity and Viscosity Index – The best lubricant in the world put into the crankcase of an engine cannot prevent wear of the piston rings and cylinder liners of this engine unless it is present on these parts in sufficient quantity the instant the pistons begin to move up and down. Only a few strokes of a dry piston ring on a dry cylinder liner can do a lot of surface damage which will take a lot of subsequent running to heal up. In order that adequate lubrication may be established promptly when an engine is started, the crankcase oil should have as low a viscosity as possible under the prevailing operating conditions. The limiting viscosity is dictated to a large extent by oil consumption characteristics, although ring belt temperatures and bearing design are also important factors. It follows that the lubricant should change as little as possible in viscosity with temperature. In other words, it should have a high viscosity index.

The question of how low in viscosity an oil can be before scuffing and excessive engine wear are encountered has been debated for a great many years, and the debate is still going on. No attempt will be made to answer this question here, because the science of lubrication has not progressed far enough to permit giving an answer which can be applied to any engine under all operating conditions. The only satisfactory way of answering this question is to "ask" the engine. The fact remains, however, that the trend in crankcase oils over the years has been to lower viscosity levels, and this trend is still continuing. Many automotive engines are being lubricated satisfactorily the year around with oils in the SAE 5W category. Furthermore, successful operation is being obtained in both laboratory and field tests with oils of even lower viscosity than those permitted under the 5W classification.

Data obtained by the Esso laboratories using the radioactive piston-ring technique developed by the Atlantic Refining Co. show that at least in some engines and under certain operating conditions low-viscosity oils give substantially less wear than those of higher viscosity. The results of tests made on a group of four uncompounded mineral oils varying in viscosity from 36 to 59 Saybolt Universal sec at 210 F, but all made from the same general crude source and all refined in the same manner, are presented in Fig. 2. In these tests piston-ring wear with the oil having a viscosity of 36 sec was less than two-thirds that obtained with the oil of 59-sec viscosity.

Effect of Detergent-Inhibitor Additives – During the past dozen years or so the use of detergent additives and oxidation inhibitors in crankcase oils has increased very rapidly. The detergents were first introduced in crankcase oils to cope with the excessive deposits of fuel soot and carbon accumulating in the ring zone and piston skirt areas of diesel engines. Detergent-compounded oils have been eminently successful in this application for not only reducing the amount of deposits formed but also in reducing the amount of piston-ring and cylinder-liner wear obtained. Oxidation inhibitors are being used in oils to improve their resistance to oxidation in high-temperature service, since oxidation of the oils can result in the formation of acidic products which are corrosive to certain of the alloy bearings such as copper-lead now in use, as well as undesirable varnish and sludge deposits.

The degree to which detergent-type additives can reduce piston-ring wear in diesel engines is illustrated in Fig. 3. The data, which were obtained by the California Research Corp. in a Caterpillar engine using a fuel containing 1.0% of sulfur, show that wear is very significantly reduced as the detergent additive concentration is increased. It will be noted that the Series II oil, made by adding roughly 16% of additive to the base oil, showed only about 15% as much wear as the base oil. Correspondingly less reduction in wear was obtained with the lower additive concentrations required to make 2-104B and Supplement 1 oils.

Inclusion of detergent-inhibitor additives in motor oils intended for gasoline engine service has also proved beneficial from the standpoint of reducing wear, as well as in preventing the accumulation of undesirable ring zone, piston skirt, and sludge deposits. However, the magnitude of the wear reduction obtained in gasoline engine crankcase oils through the use of detergent additives will vary considerably, depending on the type of service to which the oils are subjected. This, in turn, is related to the kind of engines being used,

The Authors

J. F. KUNC, JR., (M '38) is in charge of automotive fuel and lubricant research at the Esso Laboratories – Research Division of the Standard Oil Development Co. He has been with Standard Oil for 25 years and has been intimately associated with fuel and lubricant research since 1934. Mr. Kunc is a graduate of the Polytechnic Institute of Brooklyn.

J. P. HAMER has been affiliated with the Esso Laboratories – Research Division of the Standard Oil Development Co. since 1947, doing research on industrial and process oils and greases. He received his B.S. in chemistry from the University of Maryland, and received his master's degree in chemistry and his Ph.D. from Pennsylvania State College.

Table 1 – Effect of Detergent Additives on Gasoline-Engine Wear

(Light-Duty Truck Fleet – Stop-and-Go Operation)

Detergent-Inhibitor Additive in Base Oil, %	Piston-Ring Weight Loss in 12,000 Miles, g	
	High-Deposit Fuel	Low-Deposit Fuel
None	0.81	0.45
4.4	0.56	0.42
16	0.40	....

the quality of the fuel used, and a host of other factors. An example of the variation in wear improvement which is obtained through the use of detergent additives is shown in Table 1. This table presents the results of 12,000-mile tests run on a mineral oil and on the same oil with a detergent-inhibitor added, in a fleet of small trucks making house-to-house deliveries. The tests covered roughly a year of operation. When a fuel of relatively poor quality from a sludge and varnish-forming standpoint was used, an appreciable reduction in wear was obtained by incorporating a detergent-inhibitor additive in the oil. When a fuel of better cleanliness quality was used, however, wear with the straight mineral oil was very much reduced, and the detergent-inhibitor additive now had a much less significant effect on wear.

Detergent additives are not being used very extensively as yet in aviation engine oils. This stems, at least in part, from specification limitations coupled with the difficulties involved in accumulating sufficient actual flight test data on experimental products. However, experimental work in this field is continuing.

The exact mechanism by which the detergents now being used in crankcase oils function in reducing piston-ring and cylinder wear is not thoroughly understood. The detergents now most commonly used are oil-soluble barium or calcium phenates or sulfonates. These materials are quite polar in nature and are quite highly adsorbed on metal surfaces. They are also relatively nonvolatile. Their beneficial action is probably due to one or more of the following effects:

1. Resistance to being volatilized off metal surfaces by the extremely high temperature flashes developed by interlocking asperities on mating metallic surfaces.
2. Resistance to being rubbed off metal surfaces because of their highly polar nature.
3. Protection of metal surfaces against corrosive combustion products (water, acids, and so forth).
4. Prevention of the formation of carbonaceous deposits of an abrasive nature.
5. Reaction with metals to form more wear resistant surfaces.

Oiliness Agents – A rather wide variety of materials which might be termed oiliness agents have been suggested for use in crankcase oils, and additives of this type are being used in some oils. The term oiliness agent is used rather loosely, but it usually is applied to compounds which are capable of reducing the coefficient of friction or increasing the load-carrying ability of crankcase oils. The kinds of materials which have been proposed for inclusion in crankcase oils range from fatty acids, fatty oils, oxidized petroleum fractions, esters of various organic as well as inorganic acids, to graphite.

There is little doubt that under conditions of boundary lubrication beneficial effects can be demonstrated for at least certain of the oiliness agents which have been proposed. However, an increasing amount of evidence is being accumulated which shows that a reduction in the coefficient of friction of an oil or an improvement in its load-carrying ability does not necessarily lead to a reduction in engine wear. As a matter of fact, some oiliness agents have proved to be harmful rather than beneficial when used in engines because of undesirable corrosive effects. By and large, it may be that mineral oils of petroleum origin have sufficient oiliness and load-carrying ability to operate satisfactorily in present-day reciprocating engines and that further knowledge of the mechanism of wear is necessary before the magnitude of the improvements in wear obtainable through further enhancement of these properties can be assessed accurately.

Synthetic Oils – Synthetic oils, as contrasted to oils of petroleum origin, are now being used commercially to some extent for the lubrication of aircraft turbine as well as automotive reciprocating engines.

Synthetic ester-type lubricants have proved much more satisfactory than petroleum mineral oils for use in aircraft turbine engine units geared down for propeller operation. The synthetic oils devised

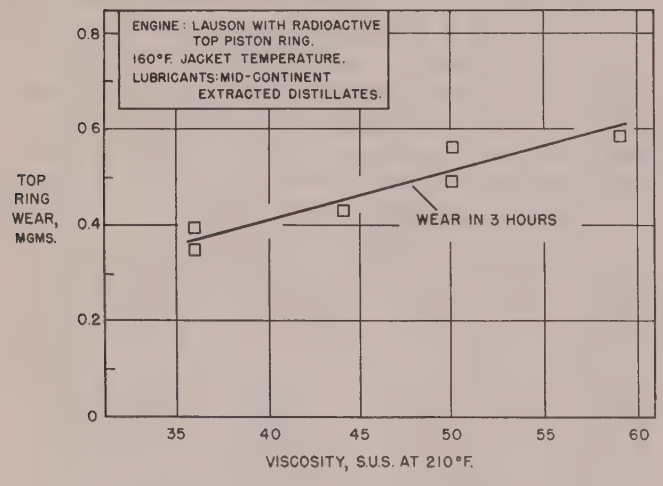


Fig. 2 – Effect of lubricant viscosity on piston-ring wear

for this service have much more desirable volatility characteristics than mineral oils of comparable viscosity. As a result, they are less prone to be flashed off the bearing surfaces of the engine at the high temperature developed both during operation and by heat soakback while the engine is cooling off after operation. In addition, the synthetic oils are capable of satisfactorily lubricating the heavily loaded gears in the speed-reduction units, whereas mineral oils are not. Synthetic oils of the ester type being used in aircraft turbine service are, at present, entirely too costly for use as crankcase lubricants.

A considerable amount of research work is being done, both within and outside the petroleum industry, on the development of synthetic oils of the ether and ester types which might find application as crankcase lubricants. At least one wholly synthetic oil is now being marketed for this purpose. It is too early to predict where this research will ultimately lead, but up to the present time there is no evidence to show that synthetics offer any advantages over petroleum base oils from the standpoint of engine wear. It is quite probable, however, that oils produced synthetically and designed to have specific performance properties may find increasing application in the crankcase-oil field during the next decade or so. One of the very important considerations in this field will reside in the economics of making synthetic oils.

Future Crankcase Oils – More efficient, economical, and compact automotive engines than those now available will undoubtedly be built in the future. To the extent that the temperatures and pressures on operating parts are increased, new lubrication problems may be introduced in these future engines. It is impossible to predict whether any radical changes in the wear characteristics of crankcase oils will be needed to satisfactorily meet the lubrication demands imposed by new or improved design engines. It is quite certain, however, that future improvements in the wear characteristics of crankcase oils will be effected through the use of chemical additives or synthetic base oils rather than through further refinement of mineral oil base stocks. The general problem of engine wear is currently receiving a good deal of attention by the petroleum industry as a whole, and it is quite likely that further improvements in the wear characteristics of lubricating oils will result from this effort.

Transmission Oils

The two widely different types of automotive transmission, namely, conventional manual shift and automatic (gear shift or torque converter), require basically different lubricants for optimum

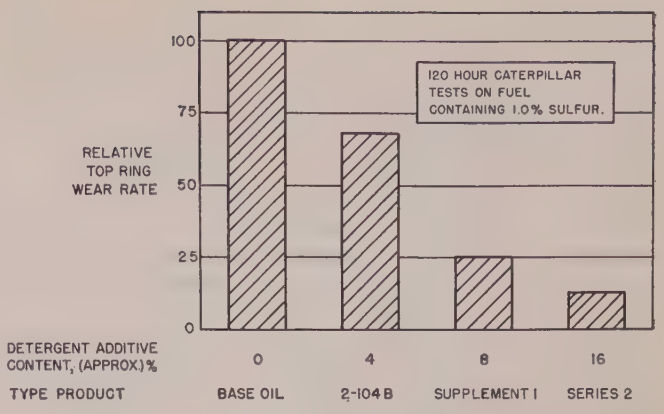


Fig. 3 – Effect of lubricant-detergent content on diesel engine wear. (Data from California Research Corp.)

performance. In the conventional transmission, the gears are designed to be lightly loaded and generally can be lubricated satisfactorily with conventional mineral oils.

The primary requirement of a gear lubricant is to prevent wear, scoring, scuffing, galling, or any other surface failure of the gear teeth. The viscosity of the lubricant under the high pressures existing between meshing gear teeth may be an important factor in this situation. Generally, viscosity increases rapidly with an increase in pressure. The resulting high viscosity between gear teeth may explain the ability of conventional mineral oils to prevent metal-to-metal contact. Work sponsored by the ASME[1] has indicated that oils vary widely in viscosity increase under pressure, as shown in Table 2. The significance of these differences in terms of gear lubrication is not known at present, since viscosity is only one of the factors involved. Pressures of the order of 100,000 psi are believed to be frequently encountered in gear lubrication.

In this same work it has also been observed that paraffinic lubricants or lubricants with high melting or freezing points partially solidify under high pressures. Under any significant rate of shear, the solid material is broken up, and the viscosity of the lubricant approaches normality. However, the small

Table 2 – Viscosities of Lubricants at High Pressures

Lubricant	Viscosity at 210 F, centipoises	
	At Atmospheric Pressure	At 100,000 Psi
Naphthenic Oil	4.7	280,000
Paraffinic Oil	4.5	13,000
Synthetic Oil	2.8	400

[1] See ASME Transactions, Vol. 73, July 5, 1951, pp. 667-676: "Viscosity and Density of Lubricating Oils from 0 to 150,000 psig and 32 to 425 F," by F. D. Bradbury, M. Mark, and K. V. Kleinschmidt.

particles of solid material in the lubricant may be considered to serve as a solid film lubricant between meshing gears, thus further preventing metal-to-metal contact and wear. This effect may prove to be fully as important as viscosity under pressure.

Corrosion of transmission-unit components can also be a problem if the lubricant is not stable against the effects of oxygen and water. This is particularly true when extreme-pressure differential lubricants are used in the transmissions, as is common practice in many cases. However, most commercial products now on the market are recognized to be sufficiently stable under the moderate temperature conditions encountered so that long corrosion-free lives are expected.

Some work has been done on the use of low-viscosity transmission and differential lubricants of petroleum or synthetic origin, in order to improve the flow of lubricant to the gear surfaces when very low temperatures prevail. The primary objective of such studies is to develop lubricants which have a low viscosity at low temperatures without decreasing lubricating ability at normal operating temperatures. This generally requires high-viscosity index oils. Lower viscosity oils also show a decrease in viscous or frictional drag at low starting temperatures, thus permitting faster cranking. At normal temperatures, a gain of about 3% in overall efficiency would be anticipated when lubricants having about half the viscosity of conventional SAE 90 gear oils are employed. Interest in these low-viscosity gear oils is expected to increase.

Automatic transmissions have greatly different requirements from conventional manual-shift units. Operating temperatures in these units are considerably higher than in conventional transmissions, and the fluid is subjected to very high rates of shear. A low viscosity is desirable in order to minimize viscous drag of the oil, obtain maximum efficiency, and permit easy starts at low temperatures. At the same time, a normal viscosity level is required at operating temperatures for satisfactory lubrication of associated gears and bearings. Fluids of high-viscosity index are therefore desirable for this service. Oxidation inhibition and detergency are required in order to provide long oil life and to prevent the deposition of sludge deposits in vital portions of the system.

The critical problem in these units is generally the lubrication of friction-band material rubbing on steel. The satisfactory lubrication of the friction-band material is important to prolong its life and to obtain sufficient smoothness for the various gear shifts of automatic transmissions. It represents a system which must have exact friction characteristics.

Currently available transmission fluids appear to be giving generally satisfactory performance, and no marked changes are expected in the near future.

Rear-Axle Oils

The same considerations of viscosity applying to oils for manual transmissions also apply to rear-axle or differential oils. In the hypoid-gear unit, however, the magnitude and type of load are such that conventional mineral oils are unsatisfactory, and extreme-pressure properties are required. An extreme-pressure lubricant may be defined as one incorporating some chemical compound which can more effectively separate meshing gear teeth and protect them against surface failure than mineral oil itself. This generally requires that the chemical compound or additive be capable of forming by chemical reaction a solid film lubricant on the surface of the steel, which is removed only with difficulty by the rubbing or sliding action of the gears.

Phosphorus, sulfur, chlorine, and lead compounds, individually or in combination, are used as additives to give extreme-pressure gear lubricants. The function of phosphorus has been generally recognized as being one of antiwear or polishing. Its use in this manner may be considered similar to the well-known treatment of steel surfaces to form various phosphorus-containing surfaces or coatings. Under high-friction conditions, the phosphorus generally is reduced and forms an iron alloy having an unknown and variable composition. This alloy has a lower flowpoint than steel, and under the pressures present on the surfaces, moves to fill the spaces between asperities in the steel surface. The load is thereby distributed over a larger contact area with a resultant decrease in unit loading. It is recognized, of course, that the actual contact area may amount to only a very small fraction of the total area of the friction surfaces. By increasing the contact area, lower friction results with a decrease in temperature and a greater resistance of the surface to loss of metal by wear. Since this alloying reaction occurs at relatively low temperatures, phosphorus-containing additives are effective in normal operating ranges. The iron-phosphorus alloy is not capable of supporting extremely heavy loads.

For heavily loaded surfaces, sulfur and chlorine compounds become important wear-reducing additives. These materials react with steel to form solid film lubricants between the friction surfaces. Sulfur compounds, for instance, react at moderately high temperatures to form iron sulfides of an amorphous nature which act as load-carrying agents by adhering closely to the surface of the steel from which they are formed. Sulfur compounds, depending on their type, may react with steel at room temperature. One measure of the reactivity of sulfur compounds is by the amount and type of stain on copper developed at different temperature levels. Chlorine compounds react with steel in a similar manner to form iron chloride. Such materials are usually more effective than sulfur compounds at higher temperatures. Chlorine-containing additives must be properly selected, since corrosion may oc-

cur in some cases if water enters the system. Lead soaps are generally used in conjunction with sulfur to form films of lead sulfide on the steel surface. This material is an effective solid-film lubricant in preventing metal-to-metal contact at moderate temperatures at relatively high loads.

The main considerations in selecting the type of additive to be used in an extreme-pressure lubricant are: first, its ability to maintain adequate lubrication; second, its effect on the oxidation stability of the base oil; and third, its ability to resist the action of water which may enter the mechanism. Effective gear oils are widely available for present equipment. Future automotive rear axles are expected to continue to use gears as small as possible, with consequently higher unit-loading. This trend will probably be accompanied by the development of even more effective extreme-pressure lubricants.

Greases

Greases are used widely in automotive applications to prevent wear in those mechanisms to which an adequate volume of oil cannot be supplied. Springs, shackles, kingpins, water pumps, cams, and wheel bearings are usually lubricated with grease. Most greases consist of a mineral lubricating oil thickened with a soap. For general chassis lubrication, a calcium (lime) soap grease is employed satisfactorily. In such applications the primary requirement is that the grease remain in position and not run out of the mechanism, so that adequate lubrication and an effective dirt-seal are maintained. Loads are generally light, and the mineral oil base used in the grease is usually capable of completely satisfactory lubrication. For wheel bearings, a grease of better structural stability is necessary, and sodium soap greases are widely employed. Here again, with proper application, the lubrication properties of sodium base greases have been found to be entirely adequate. Aluminum, barium, and various mixed soap greases have also been used to some extent in automotive applications. Currently available automotive greases are suitable for all temperatures encountered in the great majority of automotive applications.

In recent years, multipurpose greases containing lithium soaps have appeared on the market and have been used for all automotive lubrication requiring grease. Other multipurpose greases have been formulated with silica gel or bentone thickeners. The use of multipurpose products eliminates the necessity for handling more than one grease in service stations. It also minimizes the possibility of misapplication. However, the cost of such multipurpose greases is generally considerably higher than common chassis greases. In many cases it has been found that the service station operator will not pay the higher price for the multipurpose grease when his major volume is for calcium base chassis grease. In addition, the use of multipurpose greases does not insure against malpractices in application or entrance of dirt during station storage or application. It is expected that interest in the multipurpose types of grease will continue.

Table 3 – Effectiveness of Molybdenum Disulfide as a Fretting Inhibitor

Test Lubricant	Number of Cycles for Fretting
Clean	1-30
Mineral Oil	1500
MoS_2 Dusted	72,000
MoS_2 Bonded	28,000,000

Fretting of wheel bearings has occasionally given bearing surface failure. Fretting occurs either on the races of the bearings themselves or on the mating surfaces between the inner bore and the spindle. It generally results in the formation of iron oxide on the steel surfaces. This accelerates the oxidation of the bulk of the grease and increases the wear of the bearing. In the field it has been found that the use of a soft grease with a high bleeding rate is a very effective means for minimizing fretting. It is believed that the escaping oil constitutes a continuous supply to those areas undergoing fretting, both to prevent the corrosion and to carry away any corrosion debris which is formed.

The NACA has found that molybdenum disulfide is an effective material for preventing fretting under some circumstances. They have found[2] that the maximum benefits from the molybdenum disulfide are realized only when a closely adhering film is formed on the steel surface. This is best done by applying a suspension of molybdenum disulfide in some resinous vehicle and then baking and removing the excess film. In the NACA experiments, summarized in Table 3, the effectiveness of the molybdenum disulfide was demonstrated by rubbing a steel ball against a glass flat in a rapid reciprocating motion.

In some cases molybdenum disulfide does not completely eliminate fretting, even when properly applied. Furthermore, control of fretting with molybdenum disulfide is fairly expensive because of the high cost of the basic material and the difficulty of correctly applying it.

No marked changes are visualized in the lubrication requirements of grease-lubricated automotive mechanisms or in present types of greases. Although greases using synthetic oil bases are used under extreme-temperature conditions such as in aircraft, it is doubtful that products of this type will be widely used, especially in view of their high cost.

[2] See *Lubrication Engineering*, Vol. 8, October, 1952, pp. 241-243, 262: "NACA Studies of Mechanism of Fretting (Fretting Corrosion) and Principles of Mitigation," by D. Godfrey and E. E. Brisson.

Some Factors Affecting

THE tendency of a fuel-air mixture to knock is determined by the extent and type of precombustion reactions occurring in the unburned charge ahead of the flame front. A convenient method for studying these precombustion reactions involves the use of a motored engine. Since this technique substitutes compression of the unburned charge by the piston for compression of the end gas by the flame front, the problem of isolating the pro-knock reactions is greatly simplified. Preliminary investigation of the use of this technique in the du Pont laboratories was described in an earlier paper.[1] Work of this type has been continued along paths leading to a better understanding of the chemical reactions preceding knock.

The build-up of knock-producing materials depends to a considerable extent on the chemical composition of the fuel and is influenced by the ratio of hydrocarbon to oxygen and the time allowed for the oxidation reaction to take place. These factors are controlled in engine operation by fuel/air ratio and engine speed and in this paper are considered under the headings:

1. Mixture composition studies.
2. Time factor studies.

It is the purpose of this discussion to describe some of the observed results in terms of the physical manifestations of the precombustion reactions and to relate these observations to certain problems concerning the adaptation of fuels to engines, such as the temperature sensitivity of hydrocarbons.

Mixture Composition Studies

Fuel/air ratio is an important factor affecting the knock-limited power output of an engine. The basic reasons for the marked effect of mixture ratio on the chemical reactions leading to knock

USE of a motored engine as a convenient method for studying precombustion reactions is described in this paper. Preliminary investigations of this technique were made in 1950, and this paper covers the work accomplished since then in establishing a better understanding of chemical reactions preceding knock.

Extent and type of precombustion reactions which occur in the unburned charge ahead of the flame front determine the tendency of a fuel-air mixture to knock. Use of a motored engine substitutes compression of the unburned charge by the piston for compression of the end gas by the flame front.

The buildup of knock-producing materials is considered in studies of mixture compositions and time factors. Observed precombustion reactions are related to problems concerning the adaptation of fuels to engines.

The Authors

M. W. CORZILIUS, is a chemical engineer in the Combustion Research Group at the Petroleum Laboratory of E. I. du Pont de Nemours and Co., Inc. He joined du Pont in 1951 after obtaining his B.S. degree from Grove City College and M.Ch.E. degree from the University of Delaware.

D. R. DIGGS (J '46) is a mechanical engineer in the Combustion Research Group at the Petroleum Laboratory of E. I. du Pont de Nemours & Co., Inc. He joined du Pont in 1950. Mr. Diggs received his B.S. degree in mechanical engineering from Northwestern University in 1944 and thereafter was engaged in turbine and compressor research at the Cleveland Laboratory of the National Advisory Committee for Aeronautics. He returned to Northwestern in 1946 and obtained his M.S. degree in 1947, remaining as instructor in mechanical engineering until joining du Pont.

D. L. PASTELL (J '49) is leader of the Combustion Research Group at the Petroleum Laboratory of E. I. du Pont de Nemours and Co., Inc. He joined du Pont in 1947 after obtaining his B.S. and M.S. degrees in chemical engineering and his B.S. degree in electrical engineering from the University of Washington.

Precombustion Reactions in Engines

M. W. Corzilius, D. R. Diggs, and D. L. Pastell,
E. I. du Pont de Nemours & Co., Inc.

This paper was presented at the SAE Fuels & Lubricants Meeting, Tulsa, Nov. 6, 1952.

are not well understood, but some clarification should be obtained by a study of the way in which precombustion reactions are influenced by the amount of fuel and oxygen entering an engine. The explanations will, of necessity, be largely chemical in nature and will not be treated in this paper. However, some of the effects of mixture composition changes on cool flame and autoignition limits will be described and related to fired engine operation.

Experimental Techniques

The ASTM Supercharge Method engine (Fig. 1) used in these investigations was operated under the motoring conditions shown in Table 1.

The engine cylinder was equipped with a quartz window; a multiplier-type phototube, sensitive to radiation in both the visible and ultraviolet spectrum (2000 to 7000 angstroms), was used for detecting cool flames. The phototube signal, amplified to give an overall radiation amplification of 40,000,000, was applied to the vertical input of a cathode ray oscilloscope. Cool-flame initiation in the engine combustion chamber was indicated by a small pip on the oscilloscope trace.

The occurrence of autoignition was detected either by the audible sound or, when benzene was used as the fuel, by a rapid rise in the temperature of the exhaust gases. A shielded chromel-alumel thermocouple measured the exhaust gas temperature.

A detailed explanation of the experimental procedure employed in determining cool flame and autoignition limits has been given in a previous publication.[1]

Fuel/Air Ratio Changes

When a hydrocarbon-air mixture is compressed in an engine no detectable chemical reaction occurs until certain temperature and pressure conditions are reached. At this point, which varies with the particular hydrocarbon being used, oxidation and decomposition products of the fuel start to form. These products rapidly increase in amount as the temperature and pressure conditions become more severe until at a later point in the cycle they usually initiate a second stage of the oxidation reaction accompanied by the formation of a cool flame. This is detected by the appearance of faint visible radiation. The cool-flame reaction continues until the temperature and pressure conditions within the engine become severe enough to cause the hydrocarbon to ignite spontaneously. Most hydrocarbons are said, therefore, to ignite by a 2-stage process, the first stage being represented by the precool-flame reactions, and the second stage by the cool-flame reactions. This is illustrated in Fig. 2.

In general, changes in the mixture ratio entering the motored engine affect both stage A and stage B of the 2-stage fuel oxidation reaction. This is illustrated in Fig. 3 in which are shown the cool-flame and autoignition limits of n-heptane at two different fuel/air ratios. In this one figure the experimental points are shown to indicate the consistency

[1] See SAE *Quarterly* Transactions, Vol. 4, October, 1950, pp. 571-587: "Precombustion Reactions in a Motored Engine," by D. L. Pastell.

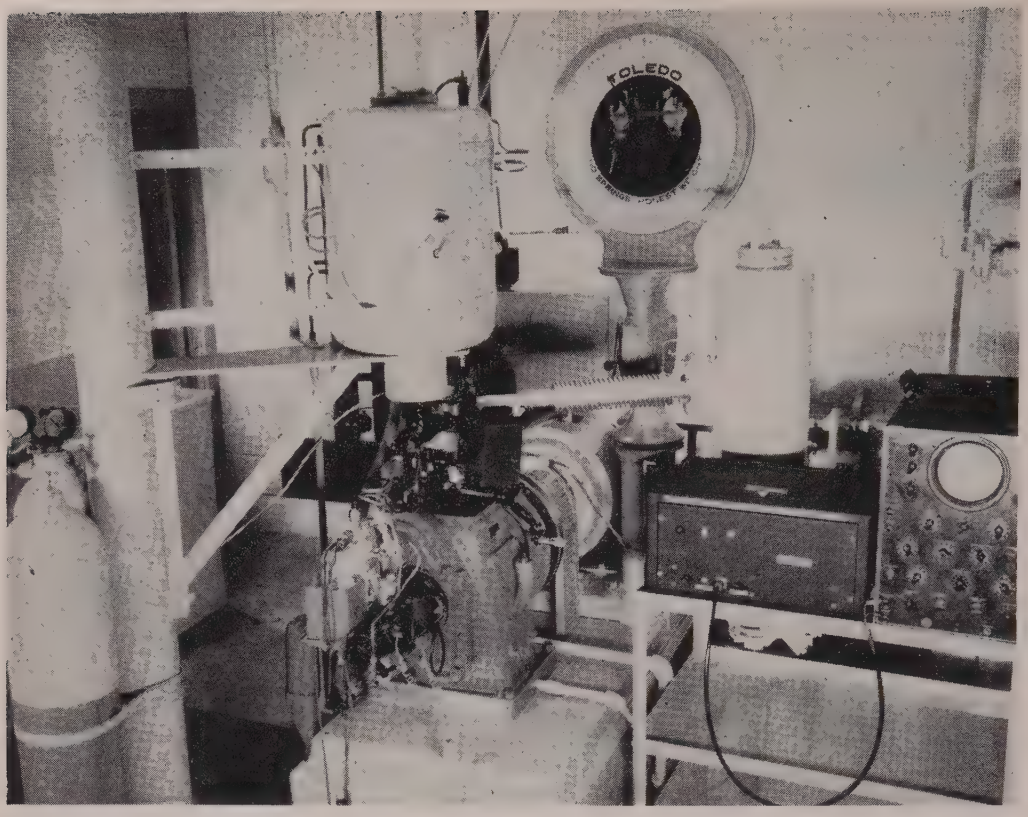


Fig. 1 – ASTM Supercharge Method engine used for studying effect of mixture composition on precombustion reactions

of the data. The lower branch of each curve defines the cool-flame limit and the upper branch defines the autoignition limit. The shaded area between the branches of each curve represents the cool-flame region; that is, the range of engine conditions under which cool flames are observed. It is apparent that an increase in the fuel/air ratio inhibits markedly the early hydrocarbon oxidation reactions since the second stage of the reaction appears at higher temperatures and pressures as indicated by the higher compression ratios.

Although the cool-flame limit curves of Fig. 3 are nearly parallel, the autoignition limit curves tend to approach each other at low manifold pressure, indicating that with n-heptane, at least, fuel/air ratio would have a lesser effect on its knocking tendency at low manifold pressures than at high pressures. This condition results from the fact that at the higher fuel/air ratio the cool-flame reaction is accelerated, so that autoignition is reached more quickly. Since the cool-flame reaction at 0.12 fuel/air ratio is initiated at much higher compression ratios than the cool-flame reaction at 0.051 fuel/air ratio, it occurs at a considerably higher temperature level. This increase in temperature level would accelerate the chemical reactions taking place. It is difficult, therefore, to assess the sole effect of increased fuel/air ratio on the rate of the second stage of the oxidation reactions in these experiments. It is quite probable that an increase in fuel/air ratio also tends to slow the cool-flame reaction as it does the precool-flame reaction; however, the large increase in reaction rate caused by increased temperature may overshadow this effect.

The effect of fuel/air ratio on both the cool-flame and autoignition limits of n-heptane is illustrated more clearly in Fig. 4. For a family of such curves the autoignition limits exhibit a minimum point which occurs at leaner fuel/air ratios as the manifold pressure is increased. The data also show that as the inlet pressure is increased cool flames and autoignition take place at lower compression ratios.

Previous work has established a correlation between autoignition in the motored engine and trace knock in a fired engine.[1, 2] A comparison of the effect of fuel/air ratio changes on motored engine autoignition limits and fired engine trace

Table 1 – Motored Engine Operating Conditions

Fuel/Air Ratio	0.029 to 0.12
Compression Ratio	4/1 to 24/1
Engine Speed, rpm	1800
Jacket Temperature, F	212
Inlet Air Temperature, F	250
Absolute Manifold Air Pressure, in. Hg	20 to 60
Fuel System Type	Manifold injection

[2] "The Oxidation Resistance of Hydrocarbons in a Motored Engine," by A. O. Melby. Presented at Delaware Chemical Society Symposium, Jan. 13, 1951.

knock limits reveals the marked similarity between these two fuel testing techniques. Fig. 5 shows the trace-knock limited compression ratio of a primary reference fuel blend as a function of fuel/air ratio at 30 in. Hg manifold air pressure. Knock is observed to occur most readily in the region of stoichiometric fuel/air mixtures, as was the case for autoignition in the motored engine. In fact, the motored and fired engine curves are very similar in shape and appearance.

The beneficial effects of adding tel to n-heptane are shown in Fig. 6. The important reactions leading to knock are believed to take place during the cool-flame stage of hydrocarbon oxidation. Any method by which these reactions can be slowed, therefore, will delay the occurrence of knock. It is only at very low fuel/air ratios, below those used in fired engines, that pure n-heptane is seen to have a wide cool-flame zone, indicating a rather slow cool-flame reaction. At more normal fuel/air ratios the cool-flame reaction is accelerated greatly so that the transition from cool flames to autoignition is quite rapid. Tel so slows the cool-flame reaction that the broad part of the cool-flame zone is shifted to normal fuel/air ratios. Also of considerable importance is the fact that tel flattens the autoignition curve of n-heptane so that autoignition, or knock, is much less dependent on fuel/air ratio. Tel, therefore, should lessen the variation of knocking tendency with mixture strength for this hydrocarbon. Limited data indicate that some other hydrocarbons, such as isooctane, do not follow this same behavior.

The effect of fuel/air ratio changes on the autoignition limits of five different hydrocarbons is shown in Fig. 7. The data are shown for a motored engine manifold air pressure of 60 in. Hg. It appears that the autoignition limits for n-heptane, isooctane, n-pentane, and cyclohexane pass through minimum points while those for benzene rise steadily as the fuel/air ratio is increased. The slope of the cyclohexane curve is much steeper than the slope for isooctane at rich mixture conditions, and at a fuel/air ratio of about 0.12 the autoignition curves for the two fuels intersect. The autoignition resistance of cyclohexane, relative to isooctane, is appreciated, therefore, as the fuel/air ratio is increased. Benzene shows a similar behavior when compared with isooctane. The data of Fig. 7 are helpful in understanding the significance of the knock-limited power output curves for these fuels and their relative blending values at different mixture ratio conditions. The knock-limited output curves are shown in Fig. 8 for isooctane, cyclohexane, and a blend of isooctane and benzene, each fuel containing 4 ml tel per gal.[3, 4]

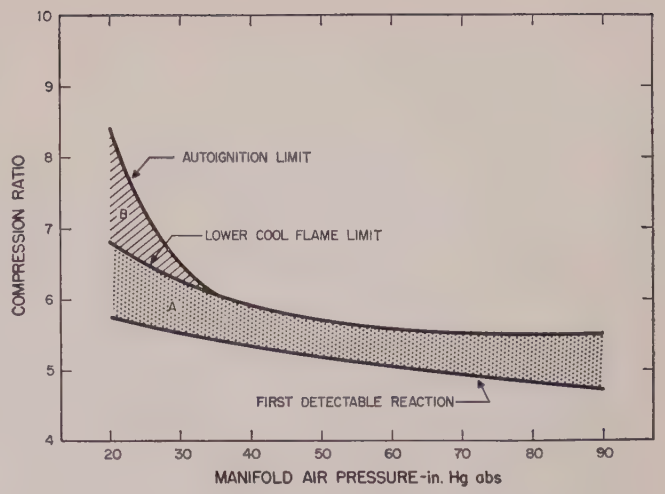


Fig. 2 – Precombustion reaction limits of n-heptane illustrating the two stages of oxidation

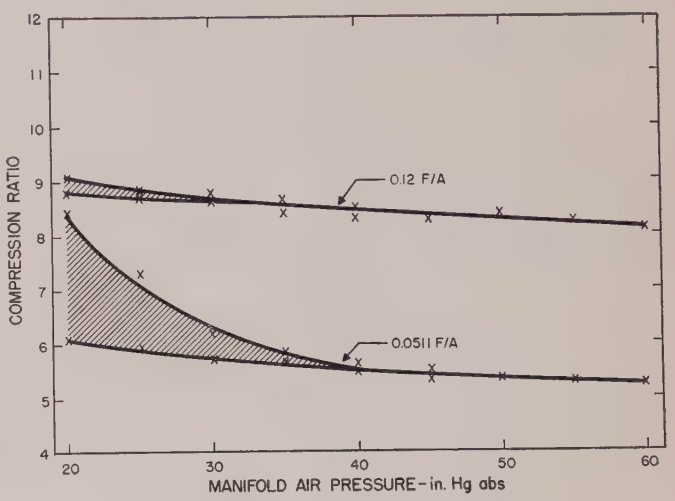


Fig. 3 – Ignition limits of n-heptane at two different fuel/air ratios – 1800 rpm

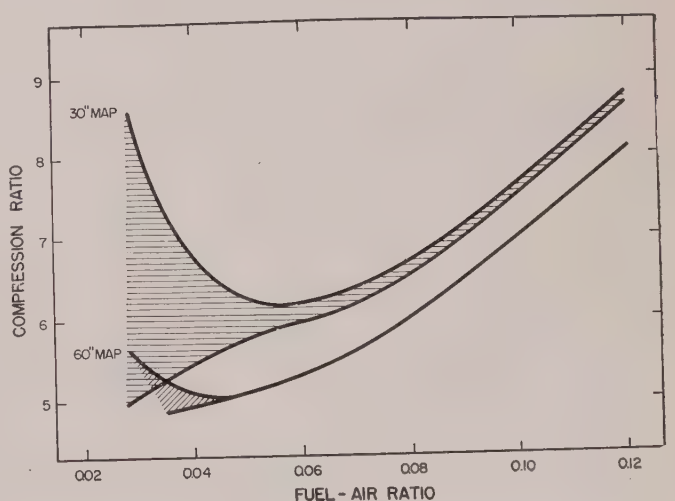


Fig. 4 – Ignition limits of n-heptane as a function of fuel/air ratio at two manifold air pressures – 1800 rpm

[3] See NACA Wartime Report E-229: "The Knock-Limited Performance of Fuel Blends Containing Aromatics," by J. R. Branstetter and C. L. Meyer. Originally published as ARR Report E5A20, January, 1945.

[4] See NACA Wartime Report E-240: "A Correlation of the Effects of Compression Ratio and Inlet Air Temperatures on the Knock Limits of Aviation Fuels in a CFR Engine," by H. E. Alquist, L. O'Dell, and J. C. Evvard. Originally published as ARR Report E6E13, June, 1946.

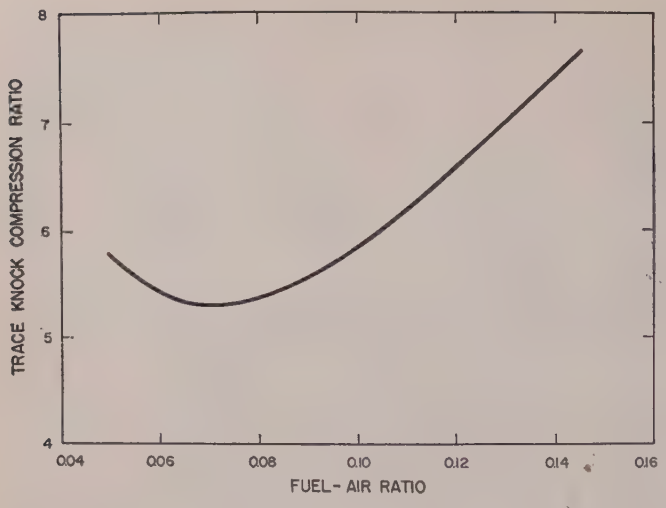


Fig. 5 – Trace-knock limited compression ratio of a reference fuel blend as a function of fuel/air ratio

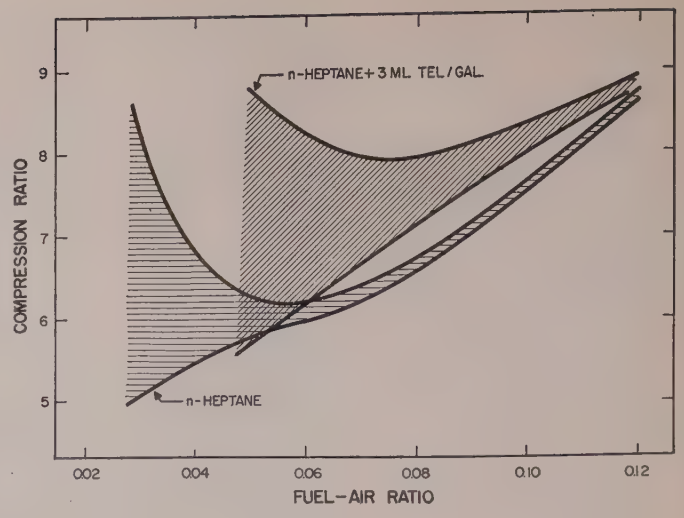


Fig. 6 – Effect of tetraethyl lead on the ignition of n-heptane – 30 in. Hg manifold air pressure, 1800 rpm

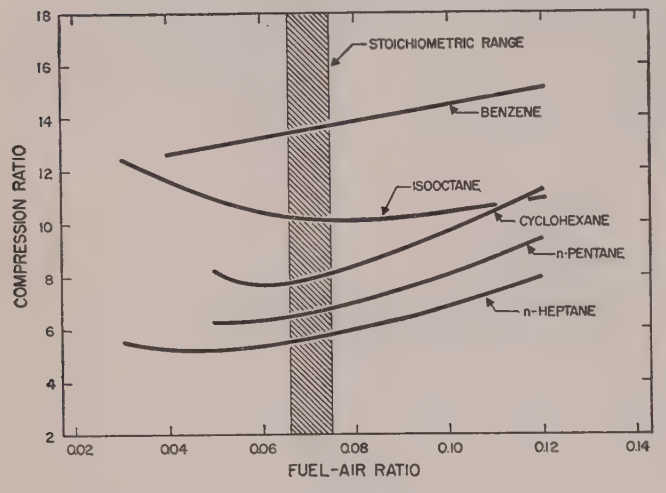


Fig. 7 – Autoignition limits of five hydrocarbons – 60 in. Hg manifold air pressure, 1800 rpm

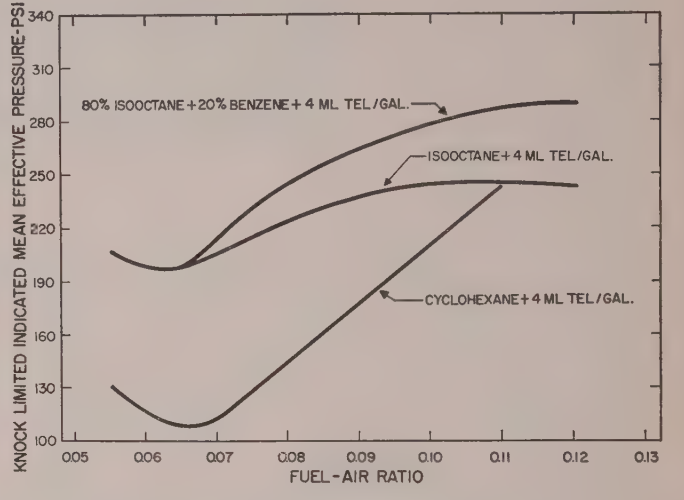


Fig. 8 – Variation of knock-limited power output with fuel/air ratio for three fuels

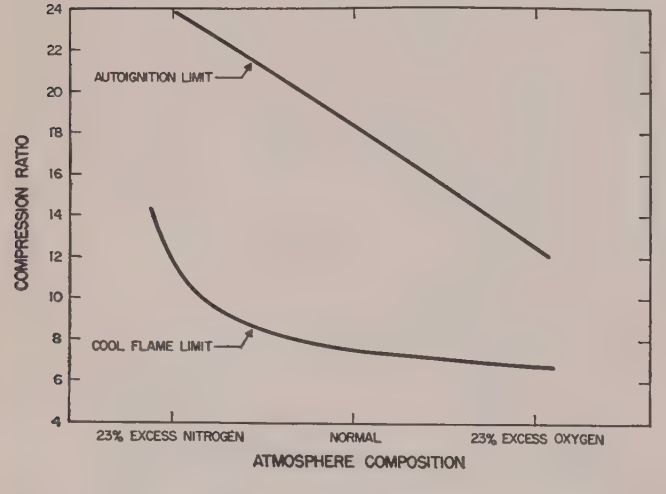


Fig. 9 – Effect of nitrogen and oxygen dilution on ignition limits of isooctane – 30 in. Hg manifold air pressure, 1800 rpm

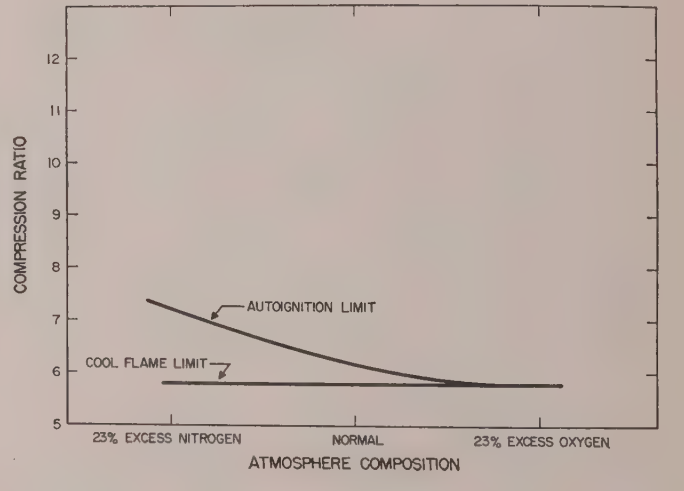


Fig. 10 – Effect of nitrogen and oxygen dilution on ignition limits of n-heptane – 30 in. Hg manifold air pressure, 1800 rpm

The knock-limited curves for these fuels show the same trends as would have been predicted from the autoignition limits.

Altered Atmosphere Studies

The mechanisms of gas-phase oxidation reactions often are investigated by first proposing a reaction scheme and then determining the effect caused by certain changes in the reaction constituents on the course of the reactions. When the proposed mechanism successfully predicts the change in the reaction course, a new kinetic interpretation of the oxidation phenomena has been evolved. The kinetic interpretations of combustion phenomena in bombs and tubes sometimes are based on experiments of this type in which the nitrogen/oxygen ratio of the atmosphere is altered. In similar experiments conducted in a motored engine the altered atmospheres were obtained by replacing a portion of the normal air flow to the engine with the desired diluent. A constant mass flow to the engine at a given manifold pressure was maintained regardless of the composition of the atmospheres, neglecting slight differences in the molecular weights of oxygen, nitrogen, and air. Data were obtained at diluent concentrations of 12, 23, and 35%; however, for simplicity in presentation, only the data for 23% dilution are shown. The compositions by weight of the atmospheres used are shown in Table 2.

The same fuel flow was used at any particular manifold pressure and was such that the fuel/air ratio with the normal atmosphere was 0.051. The same six hydrocarbons used in the study of the fuel/air ratio effects also were used in this investigation.

The effect of adding 23% excess oxygen or nitrogen to an isooctane-air mixture is shown in Fig. 9. Excess oxygen lowered the autoignition limit while excess nitrogen raised it. These effects were observed with all six hydrocarbons and at all excess concentrations of either diluent. This behavior might have been predicted since an inert gas such as nitrogen retards the speed of gas phase chain reactions. The presence of excess oxygen should increase the probability of collisions between the fuel and oxygen molecules and hence accelerate the chain reactions leading to autoignition.

Since the lower cool-flame limit represents the end of the first stage of a 2-stage oxidation reaction, any change in the cool-flame limits is a reflection of differences, either in degree or kind, in the first stage or precool-flame reactions. The altered atmospheres were observed to have peculiar effects on the cool-flame limits. The expected shift with excess nitrogen to higher than normal compression ratios, and to lower compression ratios with excess oxygen, was observed with isooctane, cyclohexane, and n-pentane. Unexpectedly, n-heptane had the same cool-flame limits with all atmospheres, as illustrated in Fig. 10.

Table 2 – Compositions of Altered Atmospheres

Weight Per Cent Replacement by Diluent	Weight Per Cent Nitrogen	Weight Per Cent Oxygen
Normal Atmosphere	76.7	23.3
23% Nitrogen	82.0	18.0
23% Oxygen	59.0	41.0

The effect of either excess oxygen or nitrogen on the ignition limits of leaded n-heptane is shown in Fig. 11. It has been noted previously that with a normal atmosphere the addition of 3 ml tel per gal to n-heptane gave no appreciable shift in the lower cool-flame limit. The cool-flame limit is leaded n-heptane also was not changed appreciably by the presence of excess oxygen or excess nitrogen. The autoignition limits, on the other hand, were displaced to more severe engine conditions whether in a normal atmosphere or in the altered atmospheres.

Because of the unexpected effects of the diluents on the lower cool-flame limit of n-heptane, the ignition limits of this fuel were defined for the additional nitrogen concentrations given in Table 3.

Since the data were obtained at a constant manifold air pressure of 30 in. Hg absolute and a stoichiometric fuel/oxygen ratio, the amount of oxygen and fuel admitted to the engine decreased as the nitrogen concentration increased. The results of these experiments are plotted in Fig. 12 where the compression ratios for cool flames and autoignition are shown as a function of fuel concentration in terms of the molfraction of n-heptane in the charge. The physical conditions defining the cool flame and autoignition limits of n-heptane are obviously a function of the fuel concentration in the combustion chamber. A more generalized picture is presented in Fig. 13 in which the effects of varying fuel/oxygen ratios have been included. The additional data shown were obtained using a normal air atmosphere and determining the cool-flame and autoignition limits at various fuel/air ratios. Referring to Fig. 13, it is apparent that, over the range of conditions tested, the lower cool-flame limit was the same whether a nitrogen diluted atmosphere was used with a constant

Table 3 – Composition of Nitrogen-Diluted Atmospheres

Weight Per Cent Air Replacement by Nitrogen	Weight Per Cent Nitrogen	Weight Per Cent Oxygen	Fuel Flow, lb per hr	Fuel/ Atmosphere Ratio, mols per mol
0	76.7	23.3	3.98	0.0191
12	79.5	20.5	3.51	0.0168
23	82.0	18.0	3.07	0.0149
35	84.8	15.2	2.59	0.0125
46	87.4	12.6	2.14	0.0100
60	90.7	9.3	1.59	0.0073
68	92.6	7.4	1.28	0.0058
75	94.2	5.8	1.01	0.0047
86	96.7	3.3	0.54	0.0028

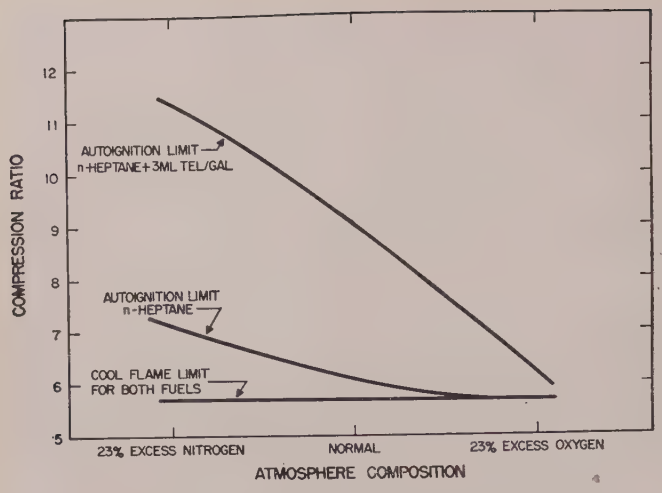


Fig. 11 – Effect of nitrogen and oxygen dilution on ignition limits of n-heptane and n-heptane containing 3 ml tel per gal – 30 in. Hg manifold air pressure, 1800 rpm

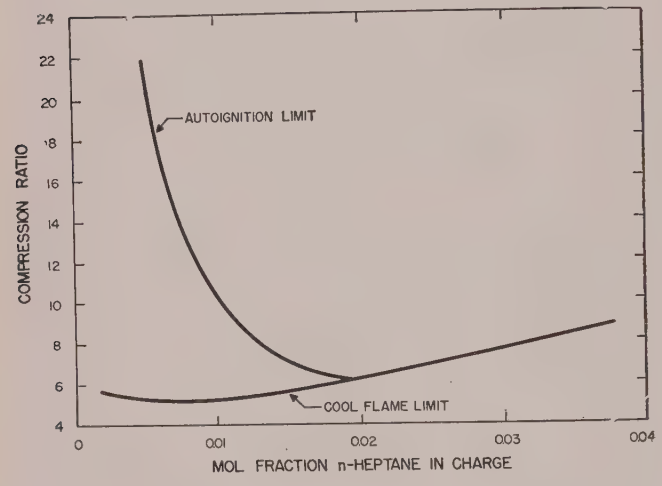


Fig. 12 – Ignition limits of n-heptane as a function of absolute fuel concentration in altered atmospheres – stoichiometric fuel/oxygen ratio, 30 in. Hg manifold air pressure, 1800 rpm

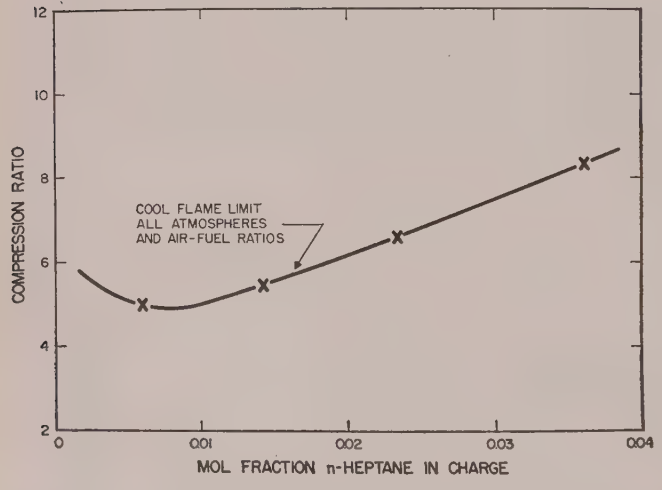


Fig. 13 – Cool flame limit of n-heptane as a function of absolute fuel concentration in altered atmospheres and at various fuel/air ratios – 30 in. Hg manifold air pressure, 1800 rpm

Table 4 – Motored Engine Operating Conditions

Fuel/Air Ratio	Stoichiometric
Compression Ratio	4/1 to 24/1
Engine Speed, rpm	600, 1200, 1800, 2400, 3000
Inlet Air Temperature, F	250
Jacket Temperature, F	212
Absolute Manifold Air Pressure, in. Hg	20 to 80
Fuel System Type	Manifold injection

fuel/oxygen ratio or a normal atmosphere was employed with different fuel/oxygen ratios. These data lead to the conclusion that the compression ratio required to establish cool flames in n-heptane is fundamentally independent of (a) fuel/air ratio, (b) fuel/oxygen ratio, and (c) fuel/nitrogen ratio, and is dependent solely on the absolute fuel concentration. This is a rather surprising conclusion which would not be predicted by the current theories concerning the mechanism of precombustion reactions. It appears, however, that other hydrocarbons behave somewhat differently than n-heptane.

The data presented in this section show that, in general, the tendency of a fuel to knock is controlled by the rate of the reactions occurring both before and during cool flames. Since both nitrogen and tel retard the cool flame reactions and only nitrogen retards the earlier reactions, it is evident that different types of chain reactions are involved. Such information is helpful in the interpretation of the chemical mechanisms involved. For full understanding, however, kinetic data derived from these types of experiments must be coupled with data showing the effect of time on reaction rate.

Time Factor Studies

From a physical standpoint, knock results when the end gas is stressed to critical temperatures and pressures for a sufficient time. The study of the time factor and its significance in controlling precombustion reactions is therefore of fundamental importance in the elucidation of the knock phenomenon. The effect of the time factor in terms of engine speed has been investigated by determining the cool-flame and autoignition limits of three hydrocarbons at different engine speeds.

Experimental Techniques

A Waukesha type E-48 crankcase was fitted with a standard 4-hole, overhead-valve cylinder. The motored engine conditions employed are listed in Table 4.

A variable-speed, direct-current dynamometer was used to motor the engine. The engine speed was maintained constant within 3% by an amplidyne dynamometer control system.

A photograph of the engine test setup is shown in Fig. 14.

The cool-flame and autoignition limits were

defined using the instrumentation and technique previously described.

Time Factor Changes

The effect of engine speed on the cool-flame and autoignition limits of n-heptane, n-heptane containing 3 ml tel per gal, and isooctane is shown in Figs. 15, 16, and 17, respectively. For each of these hydrocarbons the lower cool-flame limit is very nearly coincident at all engine speeds and does not appear to be a function of the time factor. Furthermore, the lower cool-flame limits are practically independent of the manifold air pressure. The addition of tel to n-heptane has little or no effect on the lower cool-flame limit, but it does extend the range of temperatures and pressures over which cool flames exist. For example, at a speed of 2400 rpm and 30 in. Hg manifold air pressure, cool flames with normal heptane first appear at approximately 6.5/1 compression ratio and persist until a compression ratio of 7.5/1 is reached, where autoignition occurs. With normal heptane containing tel, cool flames are initiated at about the same compression ratio and persist to a compression ratio of 11.5/1.

The autoignition limits show a distinct change with engine speed. More severe engine conditions are required for the occurrence of autoignition as the engine speed is increased; that is, as the time factor is made shorter. This is true not only for n-heptane and isooctane, Figs. 15 and 17, but also for benzene, as shown in Fig. 18. It might be noted that benzene does not exhibit cool flames in engines. At any particular manifold pressure the compression ratios required for autoignition vary differently with speed for the different hydrocarbons. It can be seen, therefore, that in comparing knock test methods which are conducted at different speeds, the time factor alone will change the absolute rating of a particular reference fuel system.

The effect of pressure, temperature, and time on the action of antiknocks adds further to the complex behavior of fuels. Information concerning the effect of the time factor can be obtained from the data of Figs. 15 and 16 by measuring the difference in autoignition compression ratios for the leaded and unleaded n-heptane at each of the engine speeds. The measured compression ratio differences are plotted as a function of engine speed at a constant manifold air pressure of 30 in. Hg in Fig. 19. In this case the engine speed range covered was extended to 3600 rpm and data were taken at engine speed increments of 300 rpm. The effectiveness of tel in suppressing autoignition in n-heptane is seen to increase from the lowest engine speed up to a maximum value at 3300 rpm, and then fall off rapidly with further increases in engine speed. Isooctane, unlike n-heptane, shows a constant response to the addition of tel over the entire speed range. Fig. 19 also shows a plot of the compression ratio differences for n-heptane as

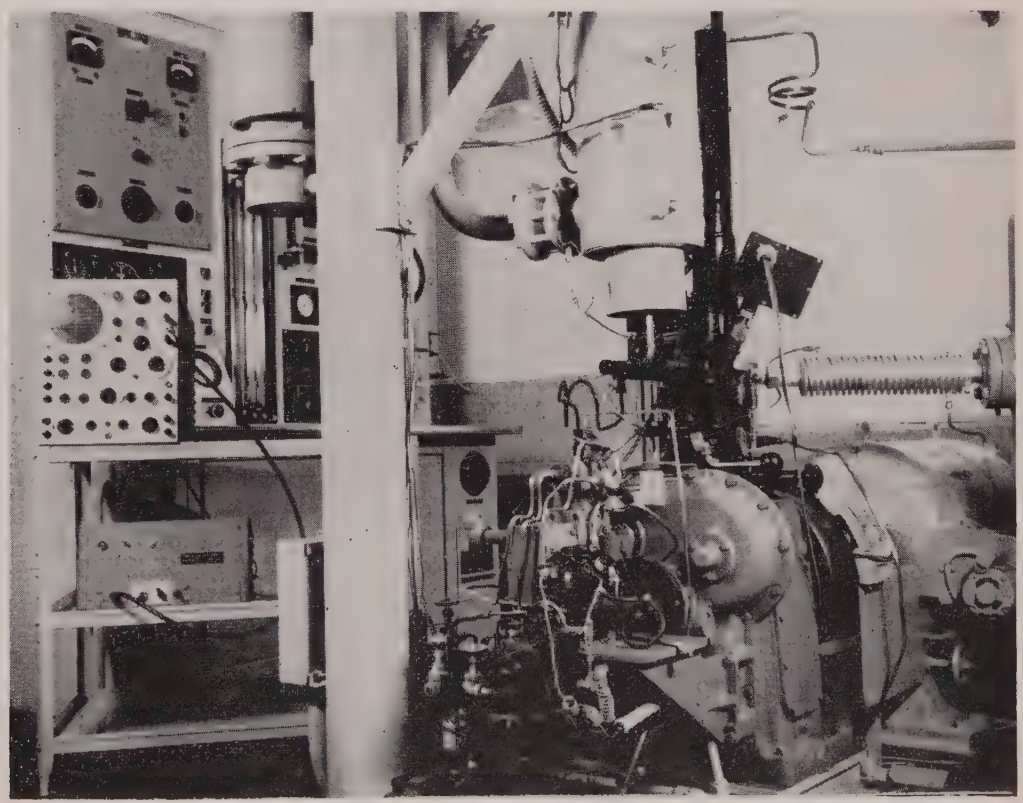


Fig. 14 – Waukesha E-48 engine used for studying effect of time factor on precombustion reactions

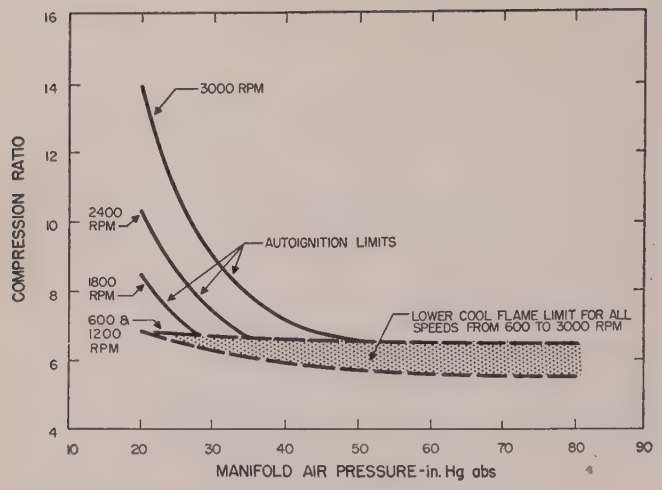


Fig. 15 – Ignition limits of n-heptane at different engine speeds – stoichiometric fuel/air ratio

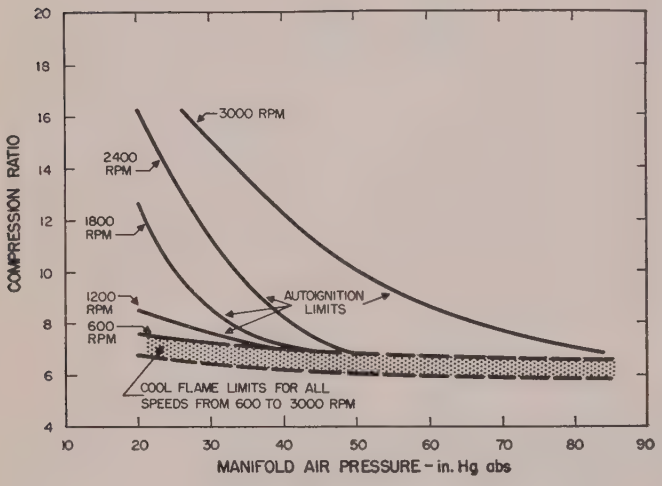


Fig. 16 – Ignition limits of n-heptane containing 3 ml tel per gal at different engine speeds – stoichiometric fuel/air ratio

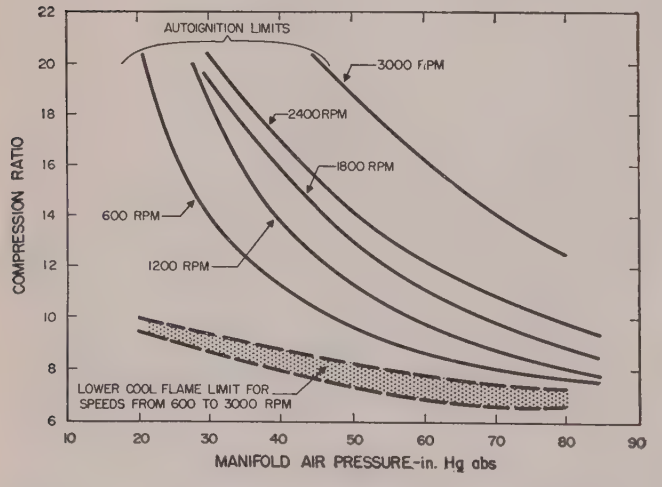


Fig. 17 – Ignition limits of isooctane at different engine speeds – stoichiometric fuel/air ratio

Table 5 – Knock-Test Data for Leaded n-Heptane

Engine Speed, rpm	Octane Number: n-Heptane	Octane Number: n-Heptane + 3 Ml Tel/Gal
600	0	33
2000	0	52

a function of reciprocal engine speed, that is, as a function of time. It is quite evident that tel is many times more effective when shorter times are allowed for reaction and shows very little antiknock action at the longer reaction times. The time function curve has been extrapolated to pass through the origin of the plot since under the theoretical conditions of zero time factor tel would not have a chance to exert any antiknock action. The increased effectiveness of tel in n-heptane at high engine speeds is shown also by the fired engine knock-test data in Table 5. These data were obtained in a General Motors single-cylinder, variable-compression engine.[5]

Temperature Sensitivity

In Fig. 20 the autoignition limits of n-heptane, isooctane, and benzene obtained at 600 rpm have been compared on temperature-pressure coordinates. The reciprocal slopes of these curves, designated by the symbol S, represent the increases in the autoignition-limited pressures permitted by a unit decrease in the maximum cycle temperature. These slopes, therefore, are a measure of the absolute temperature sensitivity of the hydrocarbon. Benzene, for example, which is known to be highly temperature sensitive, has an S-value of —0.5 psi/F. This means that a reduction of 1 deg F in the peak cycle temperature will permit an increase of 0.5 psi in the autoignition-limited pressure. Isooctane and n-heptane are used widely as reference standards for determining the temperature sensitivity of other fuels, and blends of these hydrocarbons are assigned by definition a sensitivity of zero. Under the conditions used to obtain the data of Fig. 20, isooctane actually does have an S-value of about zero near the middle of the curve, although it can be seen that this value varies considerably with changes in pressure and temperature. Also, at the engine operating conditions of Fig. 20, n-heptane has a temperature sensitivity much greater than benzene, while under other engine conditions n-heptane is quite insensitive, having S-values as low as zero psi/F.

Mixture temperature, manifold pressure, and engine speed are three important factors affecting the temperature sensitivity of a fuel, since a change

[5] See Eleventh Annual Report, API Research Project 45: "Tabulated Knock-Test Data to June 30, 1949."

Table 6 – Basic Engine Conditions for Motor and Research Rating Engines

	Motor Method Engine	Research Method Engine
Speed, rpm	900	600
Mixture Temperature, F	300	About 100
Spark Advance, deg	Variable	13 btc

in one of these variables causes a corresponding change in the temperature-pressure-time conditions. The effect of manifold pressure and engine speed on the temperature sensitivity of isooctane, benzene, and n-heptane in a motored engine is illustrated in the following figures.

Fig. 21 shows the autoignition limits of isooctane as a function of temperature and pressure for five engine speeds. The upper end of each of the limits corresponds to a low manifold pressure, and the lower end to a high manifold pressure. The effect of manifold pressure can be observed, therefore, by traversing each of the curves from end to end. Thus at 600 rpm, the S-value changes from a positive value at low manifold pressures to a negative value at high manifold pressures. The same trend, though less pronounced, is evident at 1200 rpm. At engine speeds of 2400 and 3000 rpm the reverse is true; that is, a negative S-value exists at low manifold pressures while a positive value is evident at high manifold pressures. A speed of 1800 rpm must represent a transition region, since the curve contains characteristics of the limits at both higher and lower speeds.

A similar plot for n-heptane, Fig. 22, shows that this hydrocarbon also exhibits a distinct change in temperature sensitivity with changes in manifold pressure. The dependence of S-values on engine speed is evident also in this figure, but the reversal of the curves with engine speed is absent. Neither engine speed nor manifold pressure exerts a profound effect on the S-values of benzene, as illustrated in Fig. 23.

The sensitivity of a fuel usually is defined as the difference between the Motor Method and the Research Method octane numbers. A summary of the important engine conditions for the two methods is given in Table 6.

In Fig. 24 the motored engine autoignition limits for isooctane and benzene have been plotted on pressure-temperature coordinates for engine speeds of 600 and 900 rpm. The end-gas temperature-pressure paths for fired engine cycles, using a relatively nonreacting fuel such as isooctane and having the inlet mixture temperatures of the Motor and Research Method rating conditions, have been superimposed on the motored engine curves. The fired engine end-gas temperatures were calculated by the method of Goodenough which does not take into account the heat of the preflame chemical

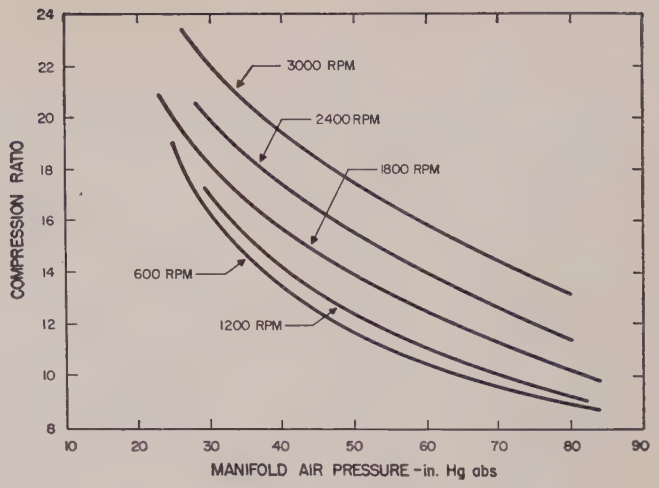


Fig. 18 – Autoignition limits of benzene at different engine speeds – stoichiometric fuel/air ratio

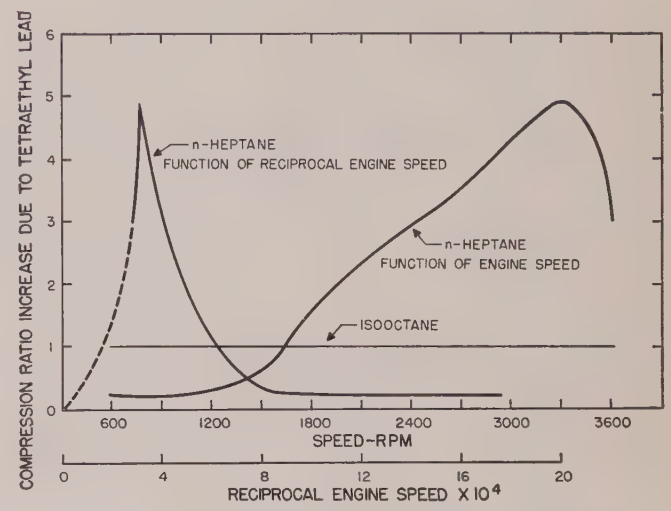


Fig. 19 – Effect of engine speed on tel response of n-heptane and isooctane – 30 in. Hg manifold air pressure, stoichiometric fuel/air ratio

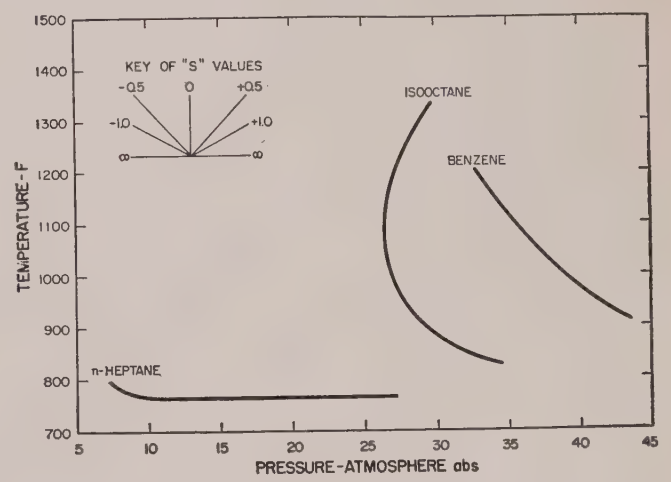


Fig. 20 – Temperature sensitivity curves for n-heptane, isooctane and benzene – stoichiometric fuel/air ratio, 600 rpm

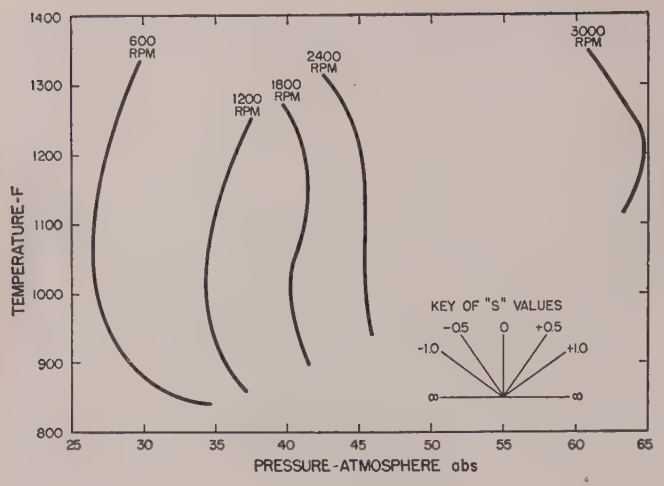


Fig. 21 – Temperature sensitivity curves for isooctane at different engine speeds – stoichiometric fuel/air ratio

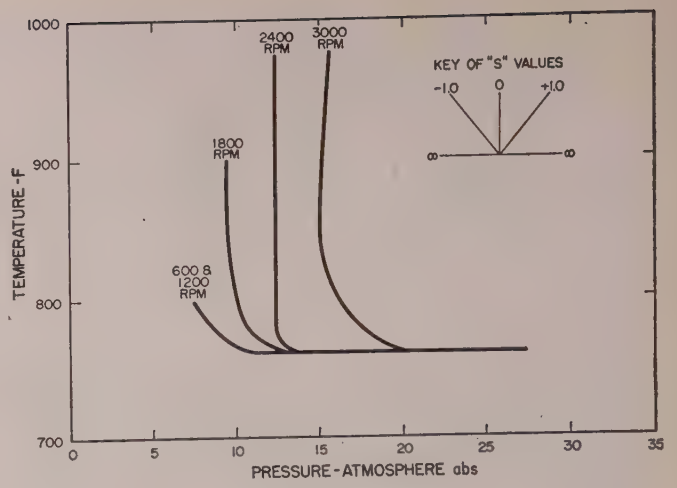


Fig. 22 – Temperature sensitivity curves for n-heptane at different engine speeds – stoichiometric fuel/air ratio

reactions of the fuel.[6] The curve representing Research Method inlet temperature has been extrapolated to higher pressures and temperatures than can be obtained under this rating procedure. Although benzene cannot be rated by the Research Method, the extrapolated temperature-pressure curves defines the end-gas conditions under which benzene would be rated if the compression ratio of the Research engine were raised above 10/1. If this could be done, the physical conditions under which benzene would be rated would be more severe than the conditions for isooctane by the increments ΔP_1 (11.6 atmospheres) and ΔT_1 (97 F). By the Motor Method, however, the conditions at which benzene gives standard knock are more severe than the corresponding conditions for isooctane by the increments ΔP_2 (3.5 atmospheres) and ΔT_2 (37 F). The fact that the pressure and temperature increments are much smaller with the Motor Method procedure explains why the rating of benzene is much closer to isooctane under Motor Method than under Research Method conditions.

Returning to a further consideration of the effect of engine speed on the autoignition limits of benzene, the data of Fig. 18 can be plotted in three dimensions to define an autoignition limit surface in terms of motored engine manifold pressure, compression ratio, and speed. Such a drawing, shown in Fig. 25, includes the three fundamental variables of the autoignition theory of knock – pressure, temperature, and time. Actually the fig-

[6] See page 278, "Principles of Thermodynamics" by G. A. Goodenough. Published by Henry Holt and Co., New York (third revised edition), 1920.

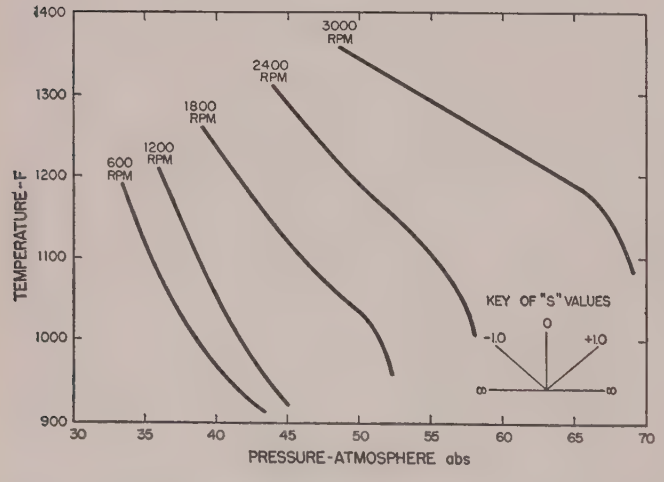


Fig. 23 – Temperature sensitivity curves for benzene at different engine speeds – stoichiometric fuel/air ratio

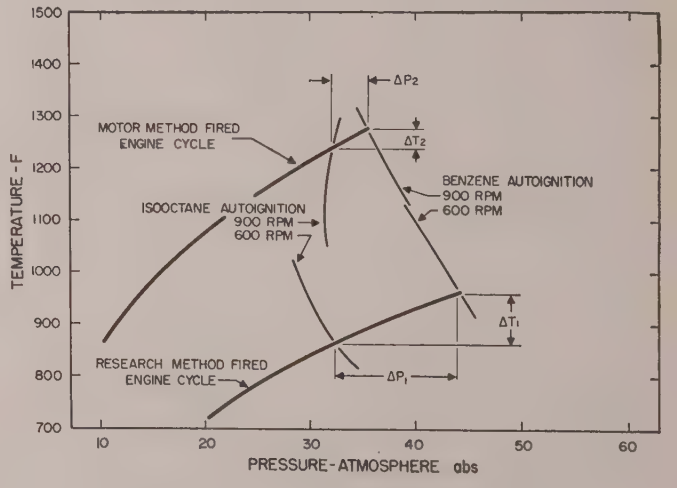


Fig. 24 – Interpretation of knock-test sensitivity of benzene

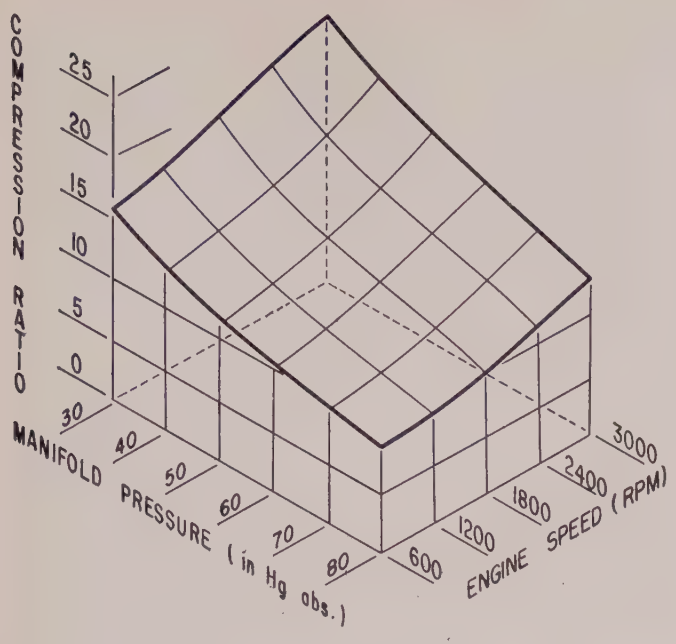


Fig. 25 – Three-dimensional representation of the auto-ignition limits of benzene – stoichiometric fuel/air ratio

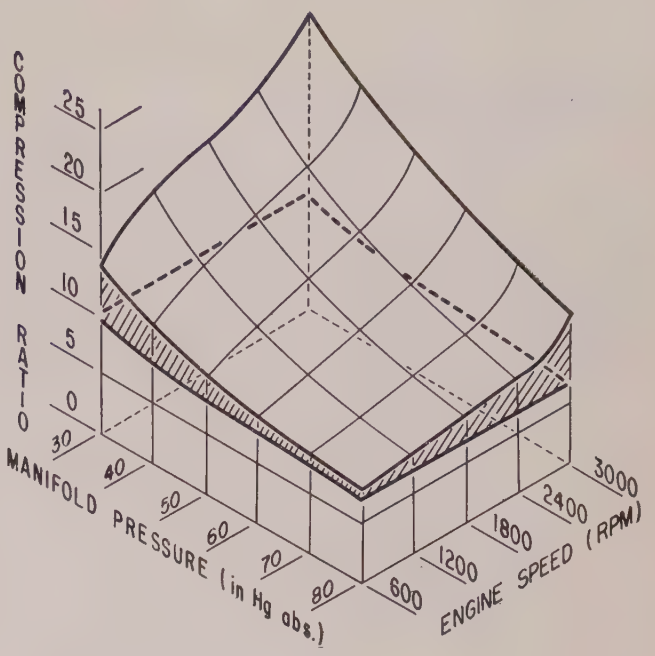


Fig. 26 – Three-dimensional representation of the cool flame and auto-ignition limits of isooctane – stoichiometric fuel/air ratio

ure represents an absolute system of fuel knock rating since all combinations of pressure, temperature, and time available in an operating engine are represented. Such a knock rating scheme, though absolute, is complicated and impractical. The figure does show the autoignition surface of benzene to exhibit a gentle upward slope toward high engine speeds, high compression ratios, and low manifold pressures.

The corresponding 3-dimensional plot for isooctane, shown in Fig. 26, is somewhat more complex since the cool-flame limits must be shown in addition to the autoignition limits. At a constant manifold air pressure the cool-flame limit compression ratios do not show any significant change with engine speed, whereas the autoignition limit compression ratios increase substantially with engine speed, particularly at the lower manifold air pressures. The shaded volume represents the range of conditions under which cool flames can occur.

Conclusions

The following conclusions have been reached as a result of these investigations:

1. Increasing the fuel/air ratio of the charge to an engine suppresses the initiation of cool flames and narrows the range of conditions under which cool flames can exist.

2. Autoignition tendencies of different hydrocarbons do not vary in a consistent manner with changes in fuel/air ratio. Many hydrocarbons show a characteristic minimum in the conditions required for autoignition; this minimum usually occurs close to the stoichiometric fuel/air ratio. The autoignition tendencies of other hydrocarbons which do not exhibit the minimum point are usually suppressed by increases in fuel/air ratio.

3. The physical conditions required for the initiation of cool flames in n-heptane are dependent solely on the absolute concentration of the fuel.

4. The conditions required for the initiation of cool flames are essentially independent of the engine time factor, but the conditions required for autoignition become more severe as the time factor is decreased.

5. Tel exhibits the greatest antiknock effectiveness in n-heptane at high motored engine speeds. In isooctane, tel shows the same effectiveness at all engine speeds.

6. The sensitivity of a fuel, as determined in ASTM knock-test engines, may be interpreted in terms of motored engine autoignition limits.

7. n-Heptane, although assigned by definition a temperature sensitivity of zero, may under some conditions react to its physical environment in such a manner that its temperature sensitivity exceeds that of benzene.

Discussion on following page ⟶

DISCUSSION

Basic Agreement Observed In Precombustion Studies

– L. A. McReynolds
Phillips Petroleum Co.

THE results reported in "Some Factors Affecting Precombustion Reactions in Engines" by Messrs. Corzilius, Diggs, and Pastell are useful in reaching a better understanding of the chemical phenomena leading to engine knock. Studies along similar lines are being conducted in the research laboratories of Phillips Petroleum Co. The procedures vary in a number of details; nevertheless, basic agreement is observed in several areas.

Table A – Engine Conditions

Jacket Temperature, F	212
Mixture Temperature, F	295
Inlet Air Temperature, F	125
Spark Advance, deg atdc	30
Air/Fuel Ratio	11/1
Compression Ratio	4/8

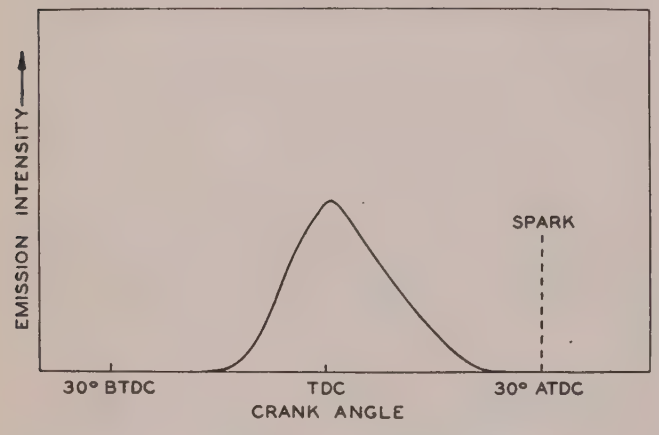


Fig. A – Typical cool flame for a nonautoigniting cycle

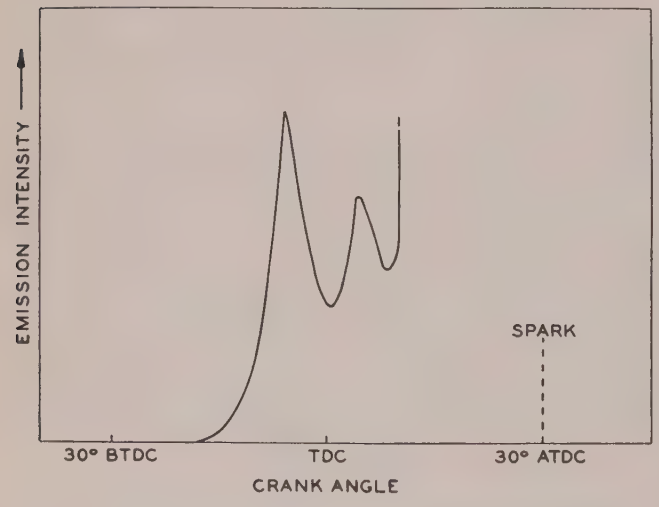


Fig. B – Typical cool flame for an autoigniting cycle

Our studies were carried out in a normally aspirated CFR, variable-compression ratio engine. Engine conditions are given in Table A.

Pressure-time traces for each engine cycle were obtained by means of a catenary strain gage pickup, the output of which was amplified and fed to an oscilloscope. Measurement of light emission from end-gas reactions was made by means of a photomultiplier tube mounted above a fused-quartz window inserted in the pickup hole of the cylinder head.

Mr. Corzilius and co-authors reported the existence of precool-flame reactions. This is in agreement with our work in which we consistently observe the development of appreciable precombustion reaction pressure prior to the onset of cool-flame emission. This appears to be a significant point with regard to reaction mechanism and has been a source of some confusion in the past. Agreement is also indicated between the studies reported by du Pont and our own with respect to the relation between the lower cool-flame limit and engine speed; our data show that for most fuels studied, the start of the cool-flame reaction occurs at essentially the same value of PV, regardless of the manner in which this value is obtained.

Of considerable interest was our observation of not one, but two cool-flame emission maxima under engine conditions where autoignition occurs.

Fig. A is a reproduction of a typical cool-flame oscillogram obtained under conditions not producing autoignition. In thise case only one maximum of emission intensity is observed. This figure was not drawn to represent a specific fuel, but describes the general features of cool-flame emission as we have observed them for a nonautoigniting cycle. Fig. B shows the two emission intensity maxima that we have found to accompany autoignition for the fuels studied to date. These have included various mixtures of pure paraffinic fuels, as well as paraffinic hydrocarbons blended with other selected hydrocarbon types including a naphthenic, an aromatic, and an olefinic material.

In only two cases have we observed the second cool flame in the absence of autoignition. The first case is that of a leaded fuel. If a paraffinic fuel is caused to autoignite and the autoignition is then eliminated by addition of tel, the second cool flame may become virtually undetectable while the first remains very nearly constant. However, if the compression ratio is increased, the second cool flame can be made to reappear without the onset of autoignition. In the second case, elimination of autoignition by leaning out the mixture does not result in the disappearance of the second cool flame. It is interesting to note that Mr. Corzilius and co-workers have observed a significant broadening of the cool-flame region under these same circumstances.

Precool-Flame Reactions Important in Knock

– P. E. Oberdorfer
Ethyl Corp.

THE authors have presented a thorough and extensive investigation of the physical manifestations of the reactions occurring in the unburned portion of the charge during engine operation.

Very recently, evidence has been obtained which indicates that reactions which occur in the precool-flame regime, mentioned by the authors, and which do not result

in detectable heat liberation or radiation, may be of great importance in the mechanism of knock.

Evidence of this type was obtained by inducting the products from a flame tube into an engine. A Pyrex glass tube, of 30-in. length and 17 mm internal diameter, has been found to be a simple and convenient device for maintaining a stabilized cool flame at temperatures up to 350 C. This tube has a transparent coating on its outside surface which permits uniform electrical heating of the tube. A single-cylinder, variable-compression engine of 17.6 cu in. displacement was operated on a chemically correct mixture of n-heptane and air, with about 2.5% of the mixture passing first into the flame tube and then being inducted into the engine. It was necessary to throttle the engine somewhat since the capacity of the tube was limited.

A thermocouple which could be moved along the axis of the tube was used to measure the temperature of the gas and thus to detect the presence of cool flames. The experiment consisted of raising the temperature of the tube and measuring the knock-limited compression ratio of the engine. The data thus obtained are shown in Fig. C. Passage of the mixture through the tube shows no effect on the knock-limited compression ratio as long as the temperature is maintained below about 175 C. However, in the temperature interval above 175 C but below that at which a cool flame is produced, a knock-inducing effect is obtained as evidenced by the decrease in critical compression ratio. Furthermore, this effect is maintained even in the cool-flame region.

The work shows that precool-flame products, possibly peroxides, formed with no detectable evolution of heat, have a proknock effect about equal to that of the cool-flame products. The implication of this is that these proknock materials are produced without exothermic reactions before cool flames appear, and they can pass through the cool flame without being destroyed. The conclusion is drawn that precool flame or pre-precombustion reactions are probably of great importance in the mechanism of engine knock.

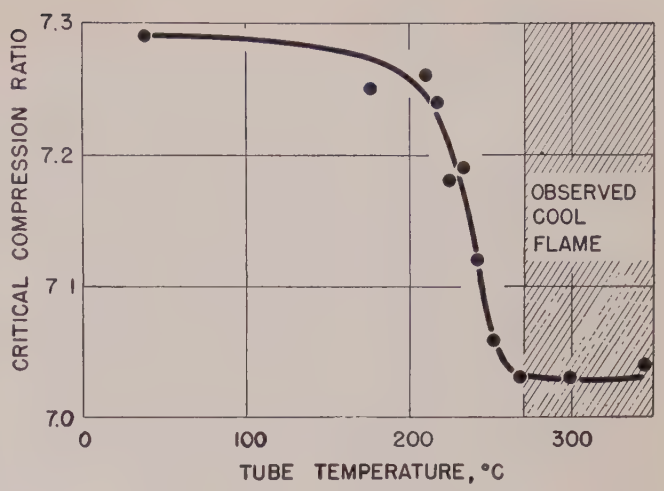


Fig. C – Influence of precool-flame reactions on knock

Precombustion Reactions and Autoignition Types Considered

– C. D. Miller
Battelle Memorial Institute

THE authors have presented a considerable amount of valuable information which should be of use in several ways: (1) to improve our understanding of the quality and the limitations of various fuel-rating methods; (2) to predict fuel antiknock ratings from fundamental measurements; (3) to indicate what characteristics, measurable in fundamental terms, should be expected of a fuel component, for knock resistance; and (4) to assist in an understanding of the nature of the phenomenon of knock.

Some elucidation of the sources and the reasons for the following opinions, as well as some of their implications, seems desirable: "The tendency of a fuel-air mixture to knock is determined by the extent and type of precombustion reactions occurring in the unburned charge ahead of the flame front. . . . The important reactions leading to knock are believed to take place during the cool-flame stage of hydrocarbon oxidation."

Photographic evidence of a 2-stage reaction associated with knock has been published previously.[a,b] Soon afterward, Jost,[c] and Spence and Townend[d] presented results of experiments with 2-stage combustion reactions in apparatus other than an engine, and discussed these results relative to their bearing upon knock in engines. The 2-stage reaction has since gained general acceptance as being closely associated with knock.

The association of 2-stage reactions with knock harmonizes with the observation that benzene does not give rise to the pressure oscillations characteristic of knock when it autoignites[e,f] and the fact that cool flames are not observed with benzene, as well as with much other data. But, a note of caution should be sounded. The first part of a 2-stage reaction was definitely shown by high-speed photographs, for example, in Fig. 12 of a previously published paper.[b] However, Fig. 24 of the same paper[b] shows high-speed photographs of a combustion cycle with violent knock, but with no evidence of a cool-flame stage.

Complication of the physical nature of the phenomenon of knock, to such an extent that it cannot be uniquely defined as "end-gas autoignition," is implicit in the quoted statements concerning the importance of precombustion reactions. If autoignition alone were sufficient, precombustion reactions could not be fitted into the picture; the occurrence of autoignition itself, with or without precombustion reactions, must then be adequate. If both precombustion reactions and autoignition are needed for knock, the precombustion reactions must prepare the end gas for an autoignition of a particular type, different from the autoignition encountered without precombustion reactions. Could this knocking type of autoignition be the detonation wave shown by the high-speed photographs?[b]

[a] See NACA Report No. 785, 1944: "Preknock Vibrations in a Spark-Ignition Engine Cylinder as Revealed by High-Speed Photography," by C. D. Miller and W. O. Logan, Jr.

[b] See SAE *Quarterly* Transactions, Vol. 1, 1947, pp. 98-143: "Roles of Detonation Waves and Autoignition in Spark-Ignition Engine Knock as Shown by Photographs Taken at 40,000 and 200,000 Frames per Second," by C. D. Miller.

[c] See Third Symposium on Combustion Flame and Explosion Phenomena, University of Wisconsin, 1948, published by Williams and Wilkins, Baltimore, pp. 424-434; "Reactions of Adiabatically Compressed Hydrocarbon-Air Mixtures," by W. Jost.

[d] See Third Symposium on Combustion Flame and Explosion Phenomena, University of Wisconsin, 1948, published by Williams and Wilkins, Baltimore, pp. 404-415: "The Two-Stage Process in the Combustion of Higher Hydrocarbons and Their Derivatives," by Kate Spence and D. T. A. Townend.

[e] See NACA Report No. 855: "Relation Between Spark-Ignition Engine Knock, Detonation Waves, and Autoignition as Shown by High-Speed Photography," by C. D. Miller.

[f] See SAE *Quarterly* Transactions, Vol. 4, April, 1950, pp. 232-274: "Ignition of Fuels by Rapid Compression," by C. F. Taylor, E. S. Taylor, J. C. Livengood, W. A. Russell, and W. A. Leary.

Two Exceptions Noted In Research Results

– J. C. Livengood
Massachusetts Institute of Technology

THIS paper offers another contribution in this important field of engine research. The results of these interesting experiments are in general agreement with those obtained in other kinds of apparatus. The reported changes of autoignition limit with variations in the variables explored appear to be in general accord with the results obtained thus far from the M.I.T. Rapid Compression Machine with two exceptions:

First, in the Rapid Compression Machine, benzene exhibits the usual minimum delay value at about the stoichiometric value of fuel/air ratio.[f] This is not in agreement with the trend shown in Fig. 7 of the present paper. However, it is somewhat difficult to make a comparison, since Fig. 7 contains three variables – pressure, temperature and fuel/air ratio, while the M.I.T. experiments were made with pressure and temperature constant and thus the only variables were fuel/air ratio and time effect (delay).

The difficulty associated with the use of compression ratio in the presentation of data has been avoided in Fig. 22 of the present paper. However, these results are again in disagreement with the behavior of the same fuel (n-heptane) in the Rapid Compression Machine, as shown in Fig. D.

In Fig. D it will be seen that in the region above, say 15 atmospheres (225 psia) and 700 F (1160 R), each successive increase in temperature of about 40 deg cuts the delay effectively in half. Fig. 22 of the present paper, while exhibiting somewhat similar "knees" in the constant rpm lines (the reciprocal of rpm can be considered to be analogous to "delay" in so far as the relative time effects are concerned), does not indicate any rational relationship between temperature and the time parameter. For example, in Fig. 22, at 20 atmospheres and 760 F, there appear to exist four simultaneous values of the time parameter, covering a range of 5 to 1. Judging from what is known about the effect of temperature on chemical reactions, this result seems incredible.

A further example of this kind of difficulty (in this case it is in the opposite sense) is shown in Fig. E. Here the autoignition behavior of a stoichiometric mixture of n-heptane and air is shown, this time under conditions of continuous compression. The apparatus is a "one shot" modified CFR engine, and compresses a charge of fresh mixture from known initial conditions until it autoignites.[g] This process is somewhat closer to that used for the data in Fig. 22, than was the case for the data in Fig. D.

As in the case of Fig. D, it will be seen that the various constant rpm curves are well separated by changes in temperature. However, the relation between time effects and temperature is not the simple exponential curve which held for Fig. D at high pressures, and the temperature change required to cut the time effect in half is much greater in any case.

Hence we are here faced with three quite different sets of curves (Figs. 22, D, and E) which purport to describe the autoignition of a mixture n-heptane and air. The question arises – is the fuel/air mixture smart enough to know what kind of machine is compressing it, or are we, the investigators, failing to recognize certain sources of error in measurement or interpretation?

It is felt that there are at least three sources of trouble:

1. The difficulty of estimating the gas temperature with

[g] M.I.T. Thesis by H. C. Johnston, Jr., 1952.

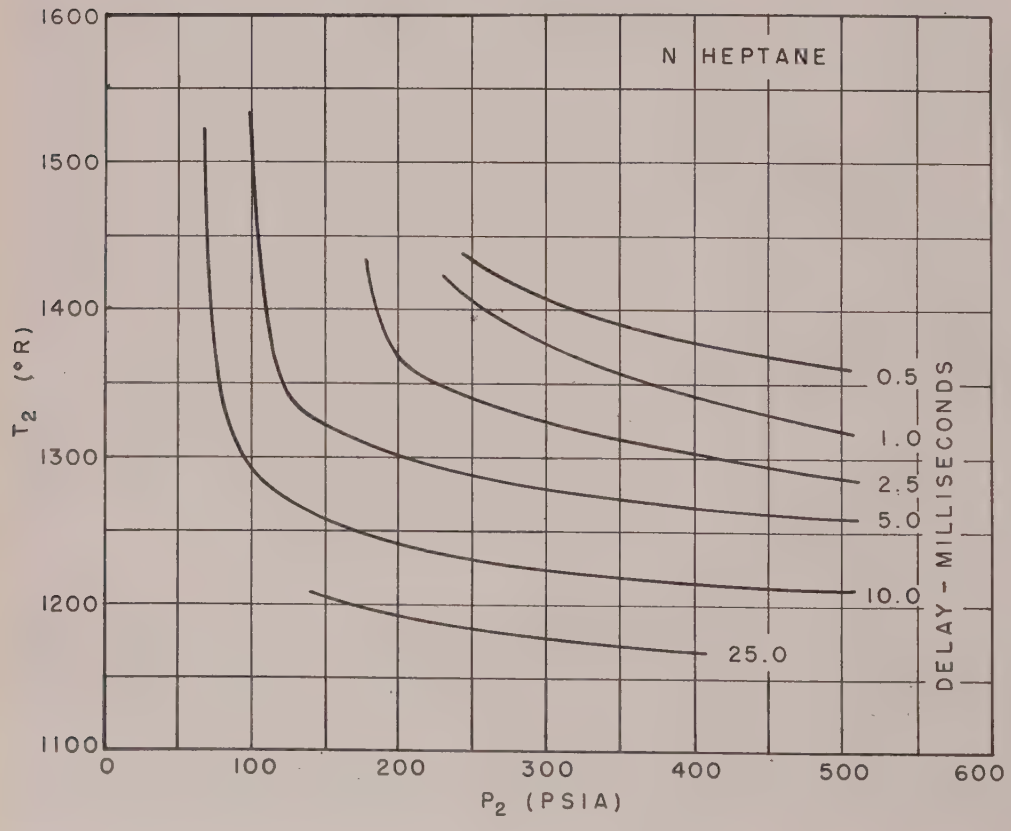


Fig. D – Autoignition map for n-heptane – fuel/air ratio 0.066

sufficient accuracy – especially in motored engines and the "one-shot" continuous compression device.

2. The lack of knowledge of the quantitative relationship between the time effects in a continuous compression process and those in an interrupted, sudden compression process.

3. Possible variations in the degree of contamination of the fresh charge in the different types of apparatus.

If these matters can be straightened out we shall be in a fair position to make rational explanations of "sensitivity," reversal of the "normal" detonation trend in engines with speed, and so forth. However, this achievement will merely describe the fuels' reactions to environment. Much more detailed knowledge is needed concerning the actual environments which exist in engines, especially with respect to temperature and heat transfer.

The permission of the Ethyl Corp., current sponsor of the M.I.T. Rapid Compression Machine project, to publish the data in Fig. D is acknowledged with appreciation.

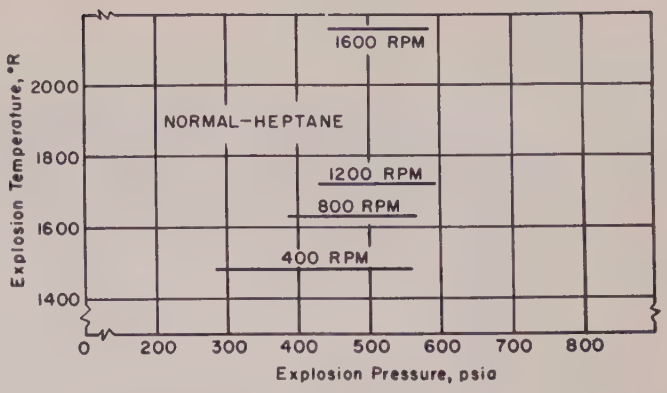


Fig. E – Pressure-temperature ignition map for normal heptane – autoignition by continuous compression

Authors' Closure To Discussion

THE discussion of Mr. Livengood clearly indicates that care must be exercised in extrapolating results obtained in any one type of research apparatus to the knock phenomenon occurring in engines on the road. Mr. Livengood has pointed out that the data of Fig. 7 in the paper do not agree with results obtained with benzene in the rapid compression machine. However, both the shape of the curves in Fig. 7 and the position of the minimum points are dependent on the engine speed and manifold air pressure. Therefore, a direct comparison of these curves with data from any other piece of equipment is probably not justified unless all data are compared in terms of pressure, temperature, and time factor.

The authors agree with Mr. Livengood's analysis of the sources of disagreement between the data of Fig. 22 in the paper and Figs. A and B of the discussion. In addition, consideration must be given to the influence of physical conditions on the cool-flame reactions. Although the cool flame limits are not shown in Fig. 22, the inclusion of these limits would show the apparent simultaneous appearance of cool flames and autoignition at the higher pressures where the autoignition limits are coincident for all engine speeds. Actually, autoignition is preceded by cool-flame formation in this region but the cool-flame reactions proceed at such a high rate that autoignition follows very rapidly after the initial cool-flame formation. At the higher pressures, therefore, the conditions required for autoignition are determined by the conditions necessary for the prior occurrence of cool flames. Until a better understanding is developed concerning the relationships between initial peroxidation, initial cool-flame formation and autoignition, the definition of autoignition limits solely in terms of pressure, temperature and time must be tempered by a consideration of the effect of these physical factors on the precombustion reactions.

The discussions of Mr. McReynolds and Mr. Oberdorfer emphasize the importance of precombustion reactions occurring prior to initial cool-flame formation. The Petroleum Laboratory also has been carrying out a series of investigations of precool-flame reactions. The technique has consisted of analyzing, by chemical and mass spectrometric procedures, samples of the motored-engine combustion-chamber charge obtained over a wide range of engine operating conditions. The results of this work will be presented in some detail in a forthcoming publication. Typical data for a fuel which exhibits cool-flame reactions are shown in Fig. F which shows that peroxides, carbonyl compounds and unsaturated materials are formed before the occurrence of the cool flame. At the time of cool-flame formation, a considerable quantity of reaction products has formed, consisting largely of aldehydic materials. At the conditions of incipient autoignition for the fuel used to obtain the data of Fig. F, over 70% of the original hydrocarbon has been converted to other materials.

Mr. Miller has raised the question of the importance of precombustion reactions to the occurrence of knock. First a clear distinction must be made between fuels that undergo precombustion reactions and those which show no reaction prior to autoignition. Most hydrocarbons fall into the class which exhibits cool-flame reactions and the statements in the paper cited by Mr. Miller refer to this class of hydrocarbons. The Petroleum Laboratory studies of precombustion reaction products previously mentioned indicate a definite correlation between the extent and type of precombustion reactions and the knocking tendencies of hydrocarbons. Moreover, some hydrocarbons which do not exhibit cool-flame radiation do, nevertheless, undergo rather extensive precombustion reactions. Certainly the present conceptions of the chemical processes leading to knock are still rather limited but new information is being developed at an accelerated rate which is bringing us much closer to a clear understanding of and a practical solution to the knock problem.

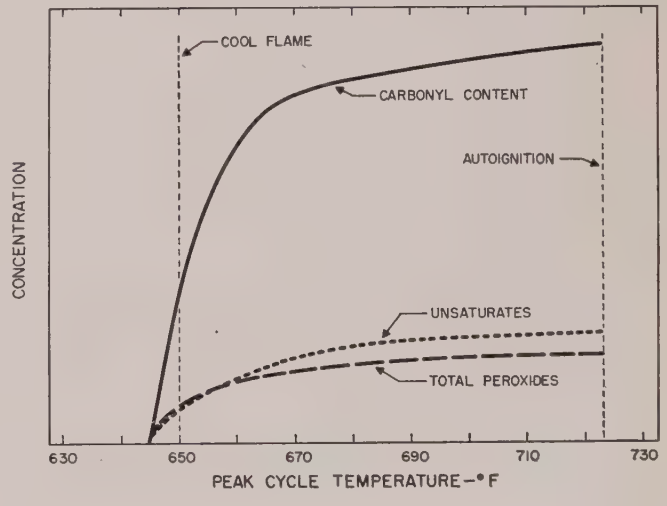


Fig. F. – Typical data for fuel exhibiting cool-flame reaction

Effect of Combustion Time on

KNOCK in spark-ignition engines is known to be the result of chemical reactions which take place in the fuel-air mixture prior to complete combustion. The occurrence or nonoccurrence of knock is determined by the outcome of a race between the flame front and the production of a critical quantity of proknock materials in the unburned portion of the charge. If the flame consumes all of the charge before the proknock reactions are completed, there is no knock. On the other hand, if the reactions progress sufficiently before the flame consumes the charge, knock occurs. Knock in an engine can be controlled by using fuels in which these precombustion reactions proceed at a slow rate. It is apparent that the knocking tendency also should be reduced by increasing the rate of burning of the charge, so that less time is allowed for the formation of proknock materials.

The mass rate of burning can be increased conveniently by increasing the engine speed, and it usually is observed that knock is less likely to occur at high engine speeds than at low speeds. This is not a practical method for controlling knock, however, since engine speed is dictated by

IF combustion of fuel-air mixtures is completed in an engine before chemical reactions have had time to take place, no knock-producing materials are formed – thus solving a chronic, critical problem. Knock can be controlled by using fuels in which reactions proceed at a slow rate. It can be reduced by speeding up the burning of the charge.

The author of this paper has explored the relationship between combustion time and knocking tendency, and has reported the results of a number of new tests.

He used a single-cylinder engine equipped with 17 spark plugs which were especially designed and fabricated for these tests. He shows how far octane requirements can be reduced under certain conditions.

The decrease was found to be significant when combustion time was reduced by firing 17 spark plugs simultaneously instead of firing one plug in the normal location. The decrease was less when the combustion chamber contained deposits than when no deposits were present. When using single ignition the location of the end gas was considered as important as combustion time in determining octane requirement.

The Author

D. R. Diggs (J '46) is a mechanical engineer in the Combustion Research Group at the Petroleum Laboratory of E. I. du Pont de Nemours & Co., Inc. He joined du Pont in 1950. Mr. Diggs received his B.S. degree in mechanical engineering from Northwestern University in 1944 and thereafter was engaged in turbine and compressor research at the Cleveland Laboratory of the National Advisory Committee for Aeronautics. He returned to Northwestern in 1946 and obtained his M.S. degree in 1947, remaining as instructor in mechanical engineering until joining du Pont.

Knock in a Spark-Ignition Engine

D. R. Diggs, E. I. du Pont de Nemours & Co., Inc.

This paper was presented at the SAE Annual Meeting, Detroit, Jan. 16, 1953.

other factors. The mass rate of burning also can be increased by using multiple ignition, but with this method the effect on the knocking tendency is not always predictable. It also is necessary to take into account the effect of increased mass rates of burning on such factors as engine roughness and power.

In order to explore the relationship between combustion time and knocking tendency and to investigate the practicality of high mass rates of burning, a study has been made with a single-cylinder engine in which it was possible to vary combustion time by multiple ignition at otherwise fixed engine operating conditions. The results are reported in this paper. Although a number of investigators[1, 2, 3, 4, 5] have described the effect of multiple ignition on engine performance, little information has been published regarding its effect on combustion time and octane requirement.

Test Equipment and Procedure

A modified ASTM Supercharge Method engine, Fig. 1, was employed in this investigation. The engine was fitted with a cylinder having a special L-head provided with 17 spark plugs, as shown in Fig. 2. The firing of more than one spark plug increased the number of flame fronts and therefore increased the mass of charge burned per unit time. In this manner the combustion time could be decreased without changing other engine conditions.

Miniature spark plugs, normally employed in model engines, were used. Because of modifications in the cooling system necessitated by the use of the special cylinder head, the spark plugs were completely surrounded by the cooling water. It was necessary, therefore, that each plug be encased in a special watertight barrel which projected above the surface of the cooling water. A spark plug, with and without the barrel, is shown in Fig. 3.

A battery ignition system was provided having ten independently variable sets of breaker points. This breaker mechanism, shown in Fig. 4, permitted the firing of any individual plug or any group of plugs over a wide range of spark timings.

The engine operating conditions used during the tests are shown in Table 1.

The tests reported in this paper were conducted using either all 17 spark plugs or one of those marked as 1, 2, or 3 in Fig. 2. The individual plugs were selected after it was found that the use of plug 1 always resulted in the longest combustion time, while the use of plug 3 always resulted in the shortest combustion time. Plug 2 was used since it was in about the normal location for a single plug in this type of cylinder head. The simultaneous firing of all plugs resulted in the minimum combustion time at any operating condition. In each test the spark advance used was that giving maximum power for the particular plug or group of plugs being used. The maximum engine power was essentially the same regardless of which individual plug was used for ignition, although there was a slight increase in maximum output when all 17 plugs were fired together.

At each operating condition a pressure-time diagram was obtained using a balanced-pressure indicator. The rate of pressure rise was considerably greater when all 17 plugs were fired; however, no appreciable increase in engine roughness was

[1] See NACA Report No. 276, 1929, "Combustion Time in the Engine Cylinder and Its Effect on Engine Performance," by C. Marvin, Jr.

[2] See NACA Report No. 399, 1931, "Flame Movement and Pressure Development in an Engine Cylinder," by C. Marvin, Jr., and R. Best.

[3] See NACA Report No. 556, 1936, "Further Studies of Flame Movement and Pressure Development in an Engine Cylinder," by C. Marvin, Jr., A. Wharton, and C. Roeder.

[4] See *Automotive Industries*, Vol. 72, March 3, 1935, pp. 324-329; March 9, 1935, pp. 354-357; and March 16, 1935 pp. 394-397: "Factors Controlling Engine Combustion," by H. Rabezzana and S. Kalmar.

[5] See *Automotive Industries*, Vol. 81, Nov. 15, 1939, pp. 534-542 and Dec. 15, 1939, pp. 632-639: "Gasoline Engine Combustion," by H. Rabezzana, S. Kalmar, and A. Candelise.

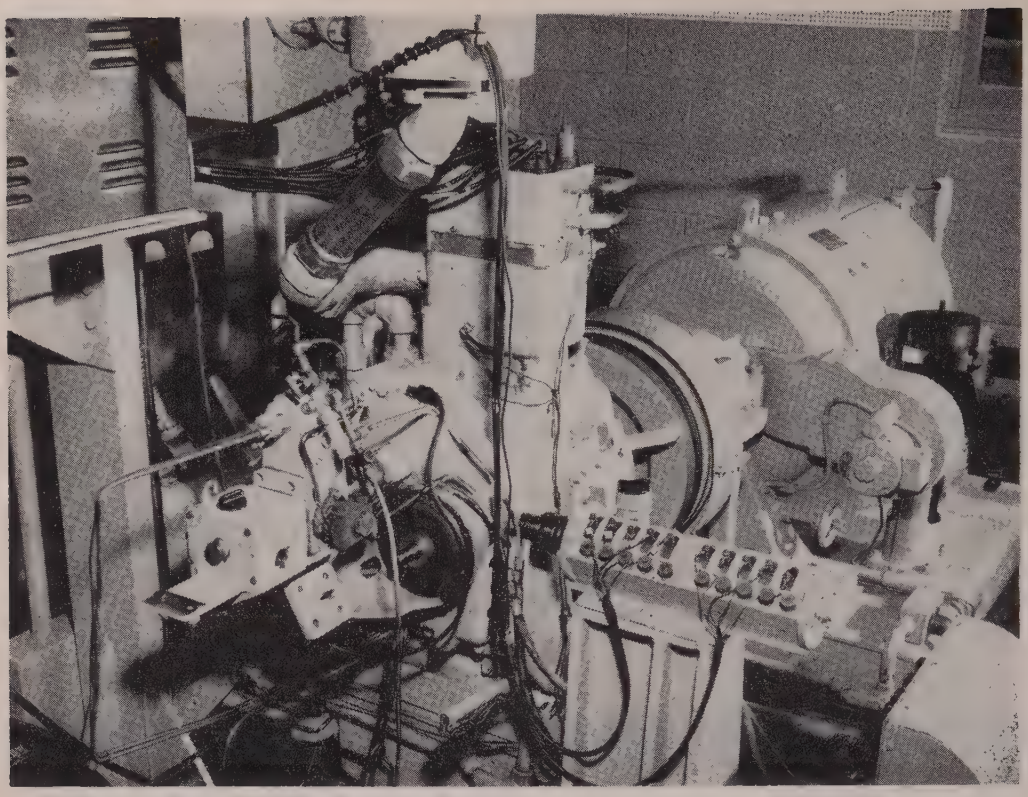


Fig. 1 – ASTM Supercharge Engine with cylinder having special L-head provided with 17 spark plugs

Table 1 – Engine Operating Conditions for Combustion Time Tests

Fuel	Primary reference fuel blends
Fuel/Air Ratio	0.0663 (stoichiometric), 0.10
Compression Ratio	5.83/1
Engine Speed, rpm	900, 1800, 2700
Coolant Temperature, F	180
Inlet Air Temperature, F	250
Absolute Manifold Air Pressure, in. Hg	30
Lubricating Oil	Solvent-extracted Midcontinent base oil
Lubricating Oil Temperature, F	165
Spark Advance	Maximum power setting

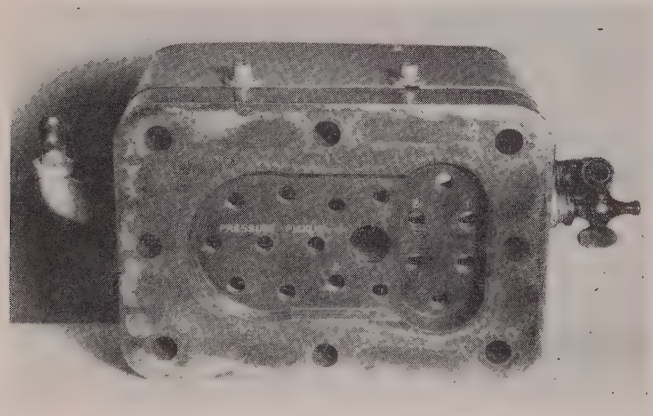


Fig. 2 – Special L-head cylinder showing location of 17 spark plugs and pressure pickup. Numbered plug locations were used in these tests. Bottom view

noticed, probably because of the extreme rigidity of the CFR engine crankcase and crankshaft.

The octane requirement was determined by operating the engine on successively lower octane number blends of primary reference fuels until incipient knock was detected by an external vibration pickup, the signal from which was observed on a cathode-ray oscilloscope. The octane numbers of these blends ranged from 60 to 90 in increments of two octane numbers.

The amount of charge burned at any particular time was calculated using the method of Rassweiler and Withrow[6] which involves separation of the pressure rise due to combustion from the pressure change due to piston motion, and the reduction of the former to constant volume conditions. When computed in this manner, per cent pressure rise due to combustion is assumed equivalent to per cent mass burned. End gas pressures were determined from the pressure-time diagram, and end gas temperatures were calculated by a method originally proposed by Goodenough.[7]

Results and Discussion

Combustion Times – The per cent mass burned as a function of engine crank angle or time is shown in Fig. 5 for tests in which plugs 1, 2, or 3

[6] See *SAE Journal* (Transactions), Vol. 42, May, 1938, pp. 185-204: "Motion Pictures of Engine Planes Correlated with Pressure Cards," by G. Rassweiler, and L. Withrow.

[7] See p. 278, "Principles of Thermodynamics," by G. A. Goodenough. Published by Henry Holt & Co., New York (third revised edition), 1920.

Table 2 – Effect of Spark-Plug Location on Combustion Time

	Combustion Time, millisec				
	900 Rpm		1800 Rpm		2700 Rpm
Plug	0.0663 F/A	0.10 F/A	0.0663 F/A	0.10 F/A	0.10 F/A
1	9.0	8.9	5.3	4.7	3.2
2	7.0	7.4	4.6	4.1	2.9
3	7.0	6.6	4.0	3.3	2.4
All	5.2	4.6	2.8	2.2	1.5

Table 3 – Effect of Engine Speed on Combustion Time – All Plugs Firing Together at 0.10 F/A

Engine Speed, rpm	Combustion Time, millisec	Reciprocal of Combustion Time, per cent of value at 900 rpm
900	4.6	100
1800	2.2	209
2700	1.5	305

were fired and for a test in which all plugs were fired simultaneously. The combustion times at each operating condition, determined from plots similar to Fig. 5, are summarized in Table 2. Data were not obtained at 0.0663 fuel/air ratio and 2700 rpm because under these conditions the use of spark plugs on the far side of the combustion chamber away from the valves resulted in frequent misfiring and generally erratic operation.

These data show that at each operating condition the use of 17 spark plugs resulted in a combustion time about 60% of that obtained when conventionally located plug 2 was fired. Virtually the same decrease in combustion time also was achieved by using plug 3 alone instead of plug 1.

Effect of Spark-Plug Location – The length of time required for combustion of the charge correlated with the length of the flame path as determined by the position of the various spark plugs. Thus, ignition by spark plug 1, which is located so that the flame travel is a maximum, resulted in the longest burning time; the use of plug 3, which is near the center of the combustion chamber and provides the shortest flame path length, resulted in the shortest burning time. Spark plug 2, which is intermediate in respect to location, generally produced intermediate burning times. Under one particular set of conditions, 900 rpm and 0.0663 fuel/air ratio, the use of plugs 2 and 3 produced identical burning times. When all plugs were fired, the flame paths were greatly shortened and the combustion time was much less than when using any single spark plug.

Effect of Engine Speed – No significant variations with engine speed in the general pattern of the combustion time curves were noted. As was expected, increasing engine speed from 900 to 1800 and 2700 rpm resulted in much shorter combustion times because of the greater turbulence of the charge at the higher engine speeds. The nearly linear relationship between the combustion time and engine speed is illustrated in Table 3.

Similar results were obtained at the other operating conditions.

The reduction in combustion time achieved by firing all plugs together as compared with plug 2 decreased substantially as the engine speed was increased. The percentage reduction in combustion

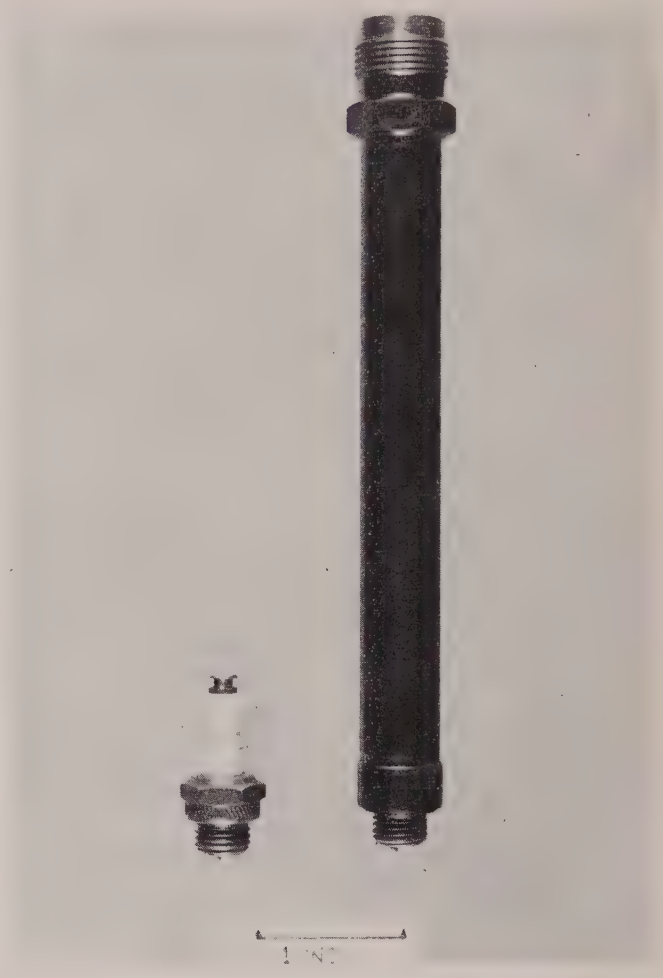


Fig. 3 – Miniature spark plug with and without special barrel

Fig. 4 – Breaker mechanism for 17-spark-plug engine

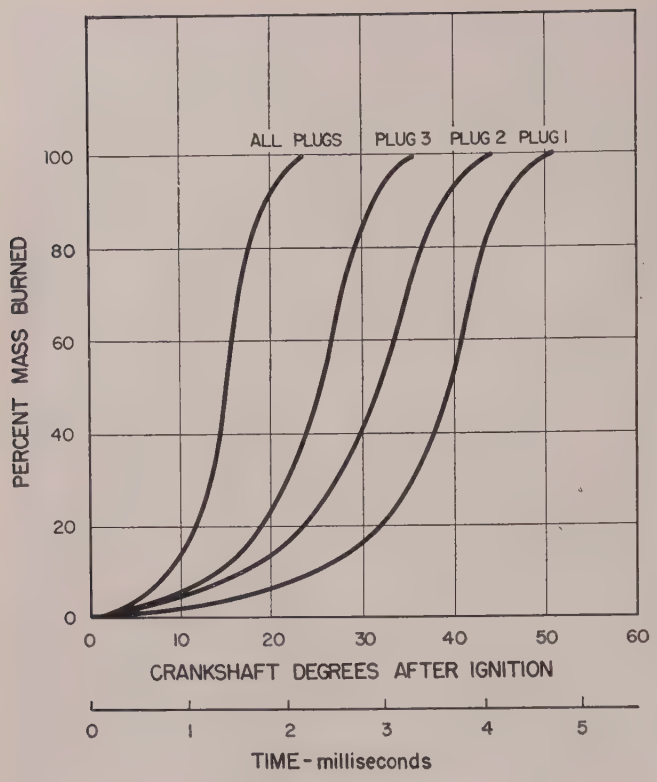


Fig. 5 – Combustion time using various spark plugs – isooctane, 0.10 F/A, 1800 rpm

time, as might be expected, was about the same at each speed. These results are illustrated in Table 4.

End-Gas Pressures and Temperatures

End-gas pressures and temperatures are of great importance in any effort to reduce the octane requirement of an engine since an increase in either of these physical variables tends to promote knock. Therefore, if increased combustion rates act to increase end-gas pressures and temperatures, the benefits expected from reduced end-gas residence time would be offset to some extent. To test this possibility an approximate method for calculating end-gas pressures and temperatures without recourse to experimental data was devised.

Table 4 – Effect of Engine Speed on Obtainable Reduction in Burning Time

(0.10 F/A)

Engine Speed, rpm	Combustion Time, millisec		Decrease in Combustion Time, When Firing All Plugs	
	Plug 2	All Plugs	Millisec	Per Cent
900	7.4	4.6	2.8	38
1800	4.1	2.2	1.9	46
2700	2.9	1.5	1.4	48

In this method, which assumed that the flame velocity was constant, the volume of charge burned as a function of time was computed for ignition with plug 2 and for all plugs firing simultaneously. The end-gas pressures and temperatures for these two cases then were determined as a function of the volume of charge burned. Finally, the previous calculations were combined to determine the pressure and temperature of the unburned gas as a function of time. The results indicated that when all plugs were fired together the end-gas pressure should be about 50 psi greater than that obtained when only plug 2 was fired. The corresponding temperature increase was approximately 40 F. These calculations were confirmed by the engine tests.

It seems reasonable that there was only a small increase in end-gas temperature with decreased combustion time when the particular operating conditions employed are considered. The longer combustion times were obtained by firing one spark plug, and the requirement of maximum power dictated that this plug be fired fairly early in the cycle, about 25 btdc. The shortest combustion time was achieved by the simultaneous firing of all plugs, and for maximum engine power ignition had to take place only a few deg before tdc. When the charge is ignited early in the cycle, the maximum cycle temperature tends to be higher than when the charge is ignited closer to tdc, because a burning mixture is being compressed for an appreciable period of time in the former case. However, the heat rejection to the coolant is greater when single ignition is used because the residence time of the flame is considerably longer than with multiple ignition. The greater heat rejection tends to lower the maximum cycle temperature and thus opposes the effect produced by early ignition. The net result is that the peak cycle pressures and temperatures should be about the same for single and multiple ignition, and thus for long and short combustion times. Since the end-gas pressure and temperature depend directly on the peak cycle pressure, they also should be about the same in the two extremes.

Engine Octane Requirement

The octane requirement when all plugs were fired always was significantly less than when any individual plug was fired, as illustrated in Table 5.

The difference between the octane requirement when all plugs were fired and that when only one plug was fired became larger as the engine speed decreased. Although the percentage reduction in burning time caused by firing all plugs was about the same at each engine speed (Table 4), the amount by which the combustion time could be reduced was much greater at low engine speeds, and hence a greater shortening of the end-gas residence time took place. Other factors being equal, a short residence time diminishes the like-

lihood of knock and thus acts to lower the octane requirement.

No definite relationship between combustion time and octane requirement was apparent when firing the various individual plugs. In general, there was only a small difference in octane requirement between the individual plugs at any operating condition despite the fact that there was usually a noticeable difference in the combustion time. Although the difference in combustion time between plugs 1 and 3 was nearly the same as that between plug 2 and all plugs, there was not a corresponding difference in octane requirement.

Effect of Spark-Plug Location. – When analyzing the octane requirement data it is necessary to consider the location of the end gas in the combustion chamber as well as the combustion time, since the position of the end gas with respect to regions of relatively high and low temperature is important in determining octane requirement. If a single spark plug is used, a substantial change in the combustion time can be obtained by altering the location of this plug, but this is accompanied by a change in the location of the end gas. It is apparent, therefore, that analyses of the results of differences in burning time between various individual plugs must take into account the location of the end gas in each of these cases.

The lowest octane requirement when firing any individual plug was observed consistently with plug 1 even though the use of this plug always resulted in the longest burning time. Plug 1 was so located that the end gas was in the portion of the combustion chamber away from the exhaust valve and thus in a region of relatively low temperature. Therefore, the temperature effect apparently reduced the knocking tendency more than the long residence time accelerated it. Ignition by plug 3, which resulted in the shortest burning time, gave the highest octane requirement in all but one instance. Plug 3 was situated so that the end gas was in the vicinity of the exhaust valve and the high local temperature in this region predominated over the short residence time and thus increased the octane requirement. Plug 2 was situated so that it was intermediate with respect to the above variables, and its use generally resulted in octane requirements intermediate between those obtained with the other two single plugs. The much shorter combustion time resulting from the use of multiple ignition appeared to overshadow any effect due to accompanying changes in end-gas location and thus became the dominant factor affecting octane requirement.

Influence of Combustion-Chamber Deposits

The combustion-chamber deposits which accumulate during the operation of automotive engines cause an increase in engine octane requirement. Since the work previously described was conducted using a clean combustion chamber, additional studies were carried out to ascertain the effect of combustion-chamber deposits on the octane requirement reduction obtained by a decrease in combustion time.

Table 5 – Effect of Combustion Time on Engine Octane Requirement

	Octane Requirement – Primary Reference Fuel Blends				
	900 Rpm		1800 Rpm		2700 Rpm
Plug	0.0663 F/A	0.10 F/A	0.0663 F/A	0.10 F/A	0.10 F/A
1	78	83	77	77	67
2	79	84	78	78	67
3	80	83	79	80	67
All	68	68	71	70	64

Combustion-chamber deposits were accumulated during 30 hr of engine operation at 1800 rpm using a fuel/air ratio of 0.075 and with other operating conditions as specified in Table 1. The fuel was a 100% catalytically cracked gasoline containing 3 ml tel per gal. Plug 2 was used for ignition. After the deposits had been formed, a complete set of new spark plugs was installed and data were taken when all plugs were fired together and also when plug 2 was fired.

As was the case in the clean engine, the effect of multiple ignition was to reduce combustion time at all engine speeds. Combustion-chamber deposits had the effect of reducing combustion time with single ignition and of increasing combustion time

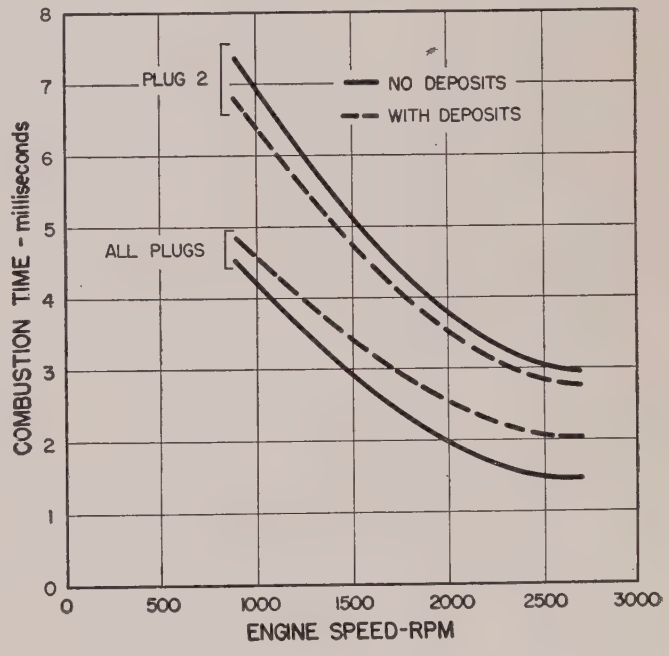


Fig. 6 – Effect of combustion-chamber deposits on combustion time – 0.10 F/A

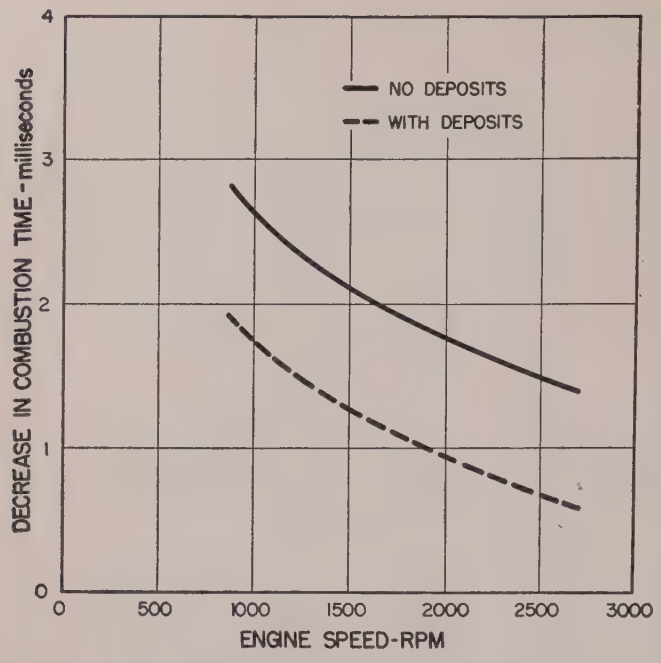


Fig. 7 – Effect of combustion-chamber deposits on decrease in combustion time obtained by firing all spark plugs – 0.10 F/A

with multiple ignition. These effects are shown in Fig. 6.

The octane requirement of the engine at the various test conditions is shown in Table 6.

The decrease in octane requirement attending the use of multiple ignition was reduced by the presence of combustion-chamber deposits. However, the decrease was still significant, especially at low speed. The octane requirement reduction was less when the engine contained deposits because the combustion time was reduced by a lesser amount under these conditions, as shown in Fig. 7.

The data of Table 6 show that at 900 rpm the presence of combustion-chamber deposits had no effect on the octane requirement when plug 2 was fired but increased the octane requirement four numbers when all plugs were fired together. When firing plug 2 the combustion time when deposits were present was less than when the engine was clean (Fig. 6), and thus the octane demand would tend to be lower in the presence of deposits. With multiple ignition, however, the combustion time with deposits in the combustion chamber was longer than with the clean engine (Fig. 6), and in this case the octane demand would tend to be higher in the presence of deposits. With either single or multiple ignition the thermal insulating effect of the deposits raised the charge temperature and this would act to raise the octane requirement. In the case of ignition by plug 2, the time effect apparently offset the temperature effect so that there was no octane requirement increase in the presence of deposits. On the other hand, with multiple ignition the time effect and the temperature effect were additive so that the octane requirement did increase when deposits were present.

At 1800 rpm the octane requirement increase caused by deposits was six numbers when plug 2 was fired and ten numbers when all plugs were fired simultaneously. When firing plug 2 at this speed the decrease in combustion time caused by deposits was less than at 900 rpm, while when using multiple ignition the increase in combustion time in the presence of deposits was greater (Fig. 6). Also, the temperature level of the engine was higher at 1800 rpm. Since when firing plug 2 there was a smaller decrease in combustion time caused by deposits and a higher temperature level at 1800 rpm than at 900 rpm, it might be expected that the octane requirement increase due to deposits would be greater at the higher speed. Similarly, with multiple ignition the larger increase in combustion time brought about by deposits together with the higher temperature level at 1800 rpm resulted in a greater deposit knocking harm at this speed.

Table 6 – Effect of Combustion-Chamber Deposits on Engine Octane Requirement

Engine Speed, rpm	Octane Requirement – Primary Reference Fuel Blends			
	No Deposits		With Deposits	
	Plug 2	All Plugs	Plug 2	All Plugs
900	84	68	84	72
1800	78	70	84	80

Conclusions

As a result of this investigation to determine the effect of combustion time on knock in a spark-ignition engine, the following conclusions can be drawn:

1. Under a variety of engine conditions generally representative of normal operation, a significant decrease in engine octane requirement can be realized when the combustion time is decreased by firing 17 spark plugs simultaneously instead of firing one plug in the normal location.

2. When the combustion chamber contains deposits, the octane requirement decrease obtainable by the use of multiple ignition is less than when deposits are not present.

3. When using single ignition, the location of the end gas is as important as the combustion time in determining the engine octane requirement.

Acknowledgment

The cooperation of the Champion Spark Plug Co. and its director of research, R. K. Christie, in designing and fabricating the special spark plugs is gratefully acknowledged.

Hydraulic Torque Converter— Common-Law Spouse of the Heavy-Duty Truck

Wilton B. Gibson, Twin Disc Clutch Co.

Paper presented at a meeting of the SAE British Columbia Section at Vancouver, Oct. 20, 1952.

THE title of this paper is the result of serious thought given to the requirements of the heavy-duty truck operator and not the facetious comment of a salesman.

Let us place ourselves in the customer's position – that of the truck operator – and take an objective look at the torque-converter transmissions in use today.

First, I wish to make it clear that no statements contain implied criticisms of a competitor's product. Their products and the application of them to trucks must be considered to obtain a rounded-out picture.

Second, I do not wish to imply that the truck operator knows the solution to our problem or even knows exactly what it is he wants. However, the heavy-duty truck operator is in a serious frame of mind. He knows he is part and parcel of a relatively young, virile, and rapidly growing segment of American industry. He is faced with serious problems and feels that, in general, standard truck components are being thrown together just to keep

THE case for the operator of the heavy-duty truck is considered in this paper. The author points out that the average driver feels that standard truck components are being assembled without regard for specific problems which must be met and solved on the road. An objective look at torque-converter transmissions in use indicates the serious need for specific study of truck applications.

The author describes the operation of logging and ore trucks in relation to loads and roads. He also gives performance data on short hauls handled by one driver, and on long runs where as many as 20 different drivers are required.

He states that trucks and related rubber-tired vehicles present one of the largest potential markets available to the torque-converter transmission manufacturer if he will but design for the specific needs of that market.

The Author

WILTON B. GIBSON is sales manager of the Hydraulic Division of Twin Disc Clutch Co. He attended the United States Military Academy at West Point, and studied engineering at George Washington University and Southern Methodist University. From 1942 to 1946 he served as Captain in the U. S. Army, as assigned chief engineer, Power Train Section, Armored Force Board, at Fort Knox.

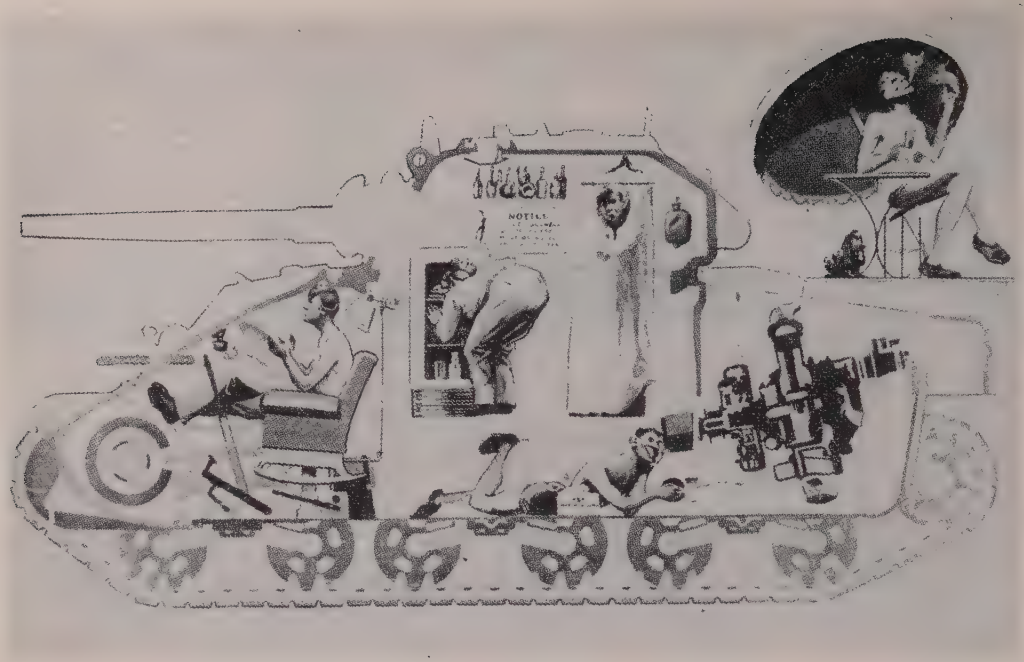


Fig. 1 – Demands upon Ordnance for impossible improvements in tank design led to basic design changes

him quiet. We must give him a sympathetic hearing and then do something about it.

Better Designs Urged

This situation reminds me of a personal experience during World War II when we in the Armored Force were making what appeared then to be impossible demands upon Ordnance for improved tank design. These requests were consolidated in the design shown in Fig. 1. I do not wish to be accused of plagiarism, but "we never had it so good." However, the tanks being produced today certainly show the results of demands from the field.

Now let us take that objective look at our problem and we will find that the hydraulic torque-converter transmission and the heavy-duty truck today are only living together out-of-wedlock to determine the practicability of a civil marriage ceremony. Neither the torque converter nor the associated mechanical transmission has had the benefit of specific design for truck application.

The heavy-duty trucking industry as we know it today had its beginning as an offspring of railroading. Those men who pioneered heavy trucking were forced to use more or less standard automotive components and heavy them up to suit their specific requirements. The volume market that attracts engineering talent and financing to produce a refined product didn't exist. However, the early operators persisted in their efforts and the results obtained justified their paying well for specialized equipment. This attracted manufacturers into producing custom-built trucks for the limited market.

The constant demand for higher production at lower costs brought about larger and larger vehicles, each step demanding higher and higher horsepower. Always, it appeared that the economic limit had been reached with each step. Also, the engineering problems increased with each jump in capacity. Tires grew larger and larger. Brake drums reached the physical limit of fitting within the wheel rims. Capacities of transmissions, drivelines, differentials, and axles were exceeded.

The work load on the driver increased with each increase in capacity. Coupled with this, the supply of skilled drivers was not keeping pace with the increased number of trucks being placed in service.

Some operators had been observing the operation of torque-converter transmissions in one make of heavy crawler tractor. The ease of operation appeared to be what they desired in their trucks and a new approach was made toward further increasing capacity and reducing costs.

Several years ago, prior to public introduction of Dynaflow, the first pilot installations of torque converters were made in the iron ore haulage trucks operating on the Mesabi Range in Minnesota. At that time the common size of truck was a 20- to 22-ton payload unit powered by a 200-hp engine. The trend today is toward 35-ton payload units powered by 400 hp in either single or dual engine installations. There are three experimental 50-ton payload units operating today, two with 550-hp butane engines and one with a 600-hp diesel engine. The year 1951 required over 1400 truck units to move 77.5% of the 231 million tons of iron ore and stripping handled. A considerable percentage of the larger truck units were equipped with torque converters. Gradients of 6 to 9% are common. Hauls are short – usually one or two miles round trip – with the loaded haul up adverse grades and

the empty haul down favorable grades. Top speeds range from 30 to 40 mph.

Mining Trucks

It wasn't long before similar installations were being made in the copper mines of the southwest. Operations were quite similar with but two exceptions. Many of the loaded hauls were down favorable grades while a few of the loaded hauls were up some rather severe adverse grades of 19 and 24%. It was in the copper pits of Chile that extreme high-altitude operations were encountered. These diversified operations clearly pointed out the need for maximum versatility of the torque converter properly to match both engine horsepower and operation requirements. Among the operating requirements was the urgent need for auxiliary downhill braking.

While the requirements of the mine-truck operator were being studied some salesman installed a torque converter in a logging truck in the redwood country of northern California. The requirements here forced the engineers to take another complete look. The loads were as heavy or heavier than those on the mine trucks. Gradients were as steep in some instances but the haul was 25 to 30 miles one way. The loaded haul was cycling between favorable and adverse grades and the speeds were higher. On the return run this same profile was negotiated empty. Here, the relative functions of the torque converter and the mechanical transmission were entirely different from those of the mine-truck units.

Logging Trucks

While engineering was pondering over the logging-truck application, sales felt they should include the western freight hauler and promptly installed several units for on-highway operation. Until now, we were dealing with trucks favored with assigned drivers where perhaps one and not more than three drivers were assigned to the truck and they operated close by a single service point. The drivers and maintenance crews took a certain pride in the experimental truck and their ability to make it work. The on-highway truck on certain overnight runs is similarly favored but the run from Denver to San Francisco and return is made without a shutdown and requires 20 different drivers. Each driver takes the next truck in line passing his station and is interested only in completing his run as fast as possible with a minimum of trouble and effort. Although he is always in a hurry on the road, the driver is not as much concerned about completing his leg short of eight hours as he is about having more time at the coffee stops to compete with the other "knights of the road" for the favors of the waitresses. In order to cope with the driver problem the converter transmission must be outwardly simple and mechanically foolproof.

The major interest of the on-highway truck operator in the torque-converter transmission is the vehicle-retarding or downhill-braking feature and the ease of handling in traffic with the improved acceleration afforded by the converter. Many operators anticipate that future traffic laws in the mountain states will require auxiliary braking devices or impose prohibitive limitations on the speed of trucks descending the long mountain grades. Vehicle weight is of such importance that a transmission offering adequate retarding characteristics will be acceptable provided its weight, above the present mechanical transmission, will be less than that of other types of auxiliary retarders.

Fuel Economy

Fuel economy is of such importance to the on-highway freight hauler, and highway gradients of such uniformity that practically all tractive performance will be in straight mechanical drive. This requirement will materially influence the determination of ratio-spread and the number of ratios in the following mechanical transmission. Furthermore, it appears that the on-highway mechanical transmission can and must be a lighter unit than its off-highway mate.

What advantages have the torque-converter manufacturers proved to date? There must be some real advantages for the mining and logging operators publicly to state they would rather have trouble with the torque converters than to have trouble without them.

First, we have increased production capacity for the equivalent horsepower either by increased unit-loading in the same cycle time or by handling the same unit-loading at a faster cyclic rate. This has been accomplished through the higher horsepower availability factor of the torque converter as compared to the mechanical transmission. The load variation factor of mining and logging trucks is quite high, requiring many gear shifts in a mechanical transmission to apply the maximum available horsepower of the engine to the wheels. Because of physical and practical limitations the driver generally is one or two gear steps below the optimum ratio for the load he is handling. The average power factor under the foregoing conditions ranges from 60 to 70% for the mechanical transmission. The torque-converter power factor, under like conditions, ranges from 70 to 85%. This is the first step in cost reduction.

Second, the service life of all power-train components – engine, transmission, driveline, differentials, axles, and tires – have been improved. R. M. Schaefer and J. A. Winter have presented excellent data on this phase of the converter function.[1] In addition, those torque-converter units providing the downhill retarding feature have brought about de-

[1] See SAE Transactions, Vol. 61, 1953, pp. 142-153: "Hydraulic Torque Converter – Its Effect on the Power Train," by R. M. Schaefer and J. A. Winter.

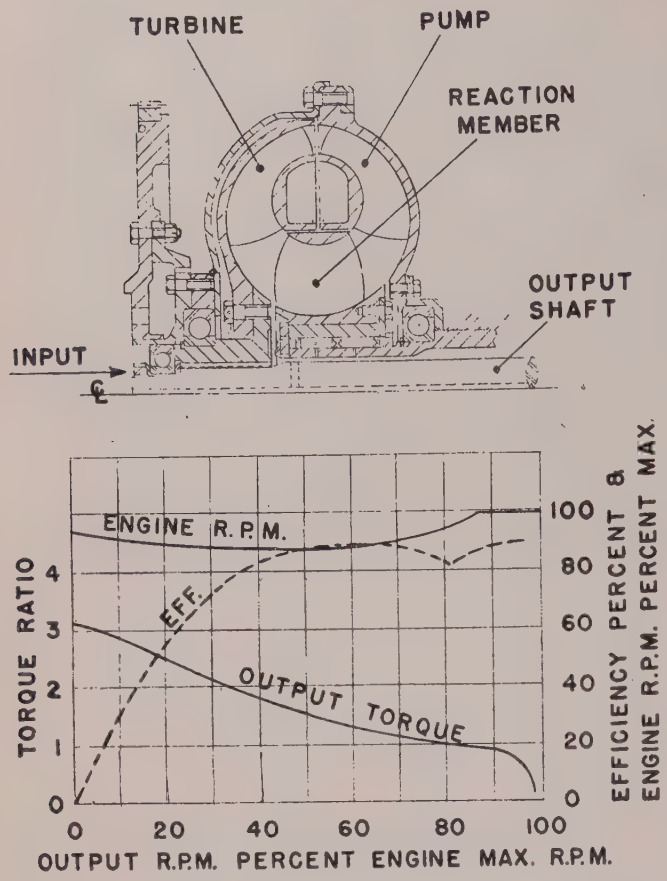


Fig. 2 – Basic single-stage torque converter showing relationship of engine speed loss to converter efficiency

cidedly improved brake and brake drum life. The reduction of maintenance cost has been of such magnitude as to be a major factor in the reduction of overall operating costs.

Third, an intangible but definitely recognized factor, is the reduced work load on the driver afforded by the torque converter. A driver, not fatigued at the end of his work day, is a more alert driver and less apt to abuse his equipment. One Mesabi mine superintendent offered the comment that the torque converters made it possible for his drivers better to enjoy their families after a day's work.

Competition Is Keen

At the time of writing this paper we cannot make any claims for the torque converters operating on-the-highway. These have been in service for one month and less. However, recently I spent two nights on the round-trip run from Los Angeles to Tulare and return over the famous Grapevine run, or Ridge Route. The driver, in this instance, is assigned to one truck and is responsible for obtaining the best performance possible. In this fleet, as in all others, each driver daily compares his performance against all other drivers as to total fuel consumed, time and speed on various legs of the round trip, and general performance of all components. Competition is high and no driver likes to be saddled with a "sick horse."

All trucks except the experimental unit are identical as to make and power and are equipped with water brakes or Hydrotarders for auxiliary braking. The experimental truck is identical as to make and power but is equipped with a 3-stage torque converter featuring a through mechanical drive and downhill braking followed by a Fuller transmission having five closely spaced ratios. All trucks are averaging 72,000-lb gross vehicle weight both ways. Top speed is 60 to 65 mph.

The run starts from Los Angeles at about 7 p.m. and crosses that city in the dinner and theater-hour traffic. It ends back in Los Angeles the following morning between 7 and 8 a.m. again crossing town during the morning office-hour traffic. At all times the experimental truck was able to keep abreast of passenger-car traffic by starting from standstill in third hydraulic, power-shifting to third mechanical, and then manually upshifting the mechanical transmission from third to fourth with the converter locked-out. City traffic time was cut from 10 to 15 min on each end of the run, leaving the standard trucks well behind. In traffic slowdowns due to congestion, the driver would leave the transmission in fourth gear, but drop back from mechanical drive to converter drive in the converter. The performance in this instance simulated the Dynaflow or Ultramatic.

Grapevine Route

As stated earlier, all open road driving was confined to mechanical drive except that occasionally the converter was used for short steep pitches and for acceleration after topping a grade. This fact accounts for the 94-gal fuel consumption for each round trip while all standard trucks were ranging from 86 gal to 110 gal. Incidentally, the driver with the 86-gal record required two engine overhauls recently and a trip to the front office.

The descent down the Grapevine going north is the worst braking requirement of the round trip, consisting of seven miles of 6 and 6.5% favorable grade. The converter holds the truck steady at 30 mph in third gear with only five light assists from the service brakes. The brakes were cold at the bottom of the grade. Engine water temperature was maintained at 175 deg without thermostats or shutters. Engine water temperature at all other times was 165 deg. Converter temperature stabilized at 220 deg during the braking period. The company passenger car following the truck had to use brakes

continuously to stay in line. A modification to the converter is being made that will permit the driver to come off this grade in fourth gear at 40 mph. No loss in truck acceleration is anticipated.

The driver is sold – the fleet owner noncommittal. The answer is 6 to 9 months off, at least. We have stayed under the combined weight of a standard transmission and water-brake system. Our fuel economy, so far, is slightly better than average for the mechanical units. Our braking appears to be adequate and our overall time is below average. Just what the score will be 6 to 9 months from now only time will tell. One thing is certain – there will be further changes to make.

Power-Train Components

Now what is wrong with the torque converter power-train today? Let us consider the components in this order – torque converter, engine, mechanical transmission, and truck chassis.

The two basic types of hydraulic torque converters commonly found in trucks today are the single-stage and the 3-stage. The single-stage converters are offered to the basic design by Schneider and the polyphase design by Allison. The 3-stage torque converters are offered by Twin Disc, while Spicer supplied a few during the initial experimental installations. All of these units are good in their particular field but each has serious deficiencies when considered for truck applications. Many of these deficiencies will be eliminated soon, while others will be minimized with years of refinement.

We, the torque-converter manufacturers, are in the process of eliminating some of the major deficiencies now but we must face a fact which I personally feel is obvious. The truck operator will go along with us for the present on those deficiencies less easy to eliminate or minimize because he is making a major stride forward. However, once the operator has become accustomed to the higher production at lower cost, he will establish this as a new floor or normal base. Then the pressure will surely be on to improve converter performance further.

It is not the purpose of this paper to repeat in detail the merits and operating characteristics of the several types of torque converters. A review of their general characteristics is required to develop a major point of this paper – the effect of torque-converter design on output horsepower performance.

Efficiency Curves

Published efficiency curves for all makes of converters are extremely accurate but it must be borne in mind that the efficiency shown relates strictly to input horsepower. Therefore, if the circuit design is such as to load the engine within a desirable working range of the converter, the input horsepower is correspondingly lower with a resultant lower output horsepower regardless of the converter efficiency. Conversely, circuit design can also permit high engine speed and high input horsepower while efficiency falls off to zero at just that converter range where high output horsepower is most desirable.

The single-stage and 3-stage torque converters respectively produce the results just mentioned. Fig. 2 illustrates the basic single-stage torque converter as produced by Schneider and the relationship of engine speed loss to converter efficiency. The loss of input horsepower offsets the peak efficiency of the torque converter. In this unit as well as the polyphase type the reaction member is freewheeled to provide a coupling phase in the high-speed portion of the torque converter range. In this design the engine is completely unloaded at zero output torque – a highly desirable feature.

In the Allison polyphase torque converter, Fig. 3, the reaction member is split to provide a partial converter range and maintain a more uniformly

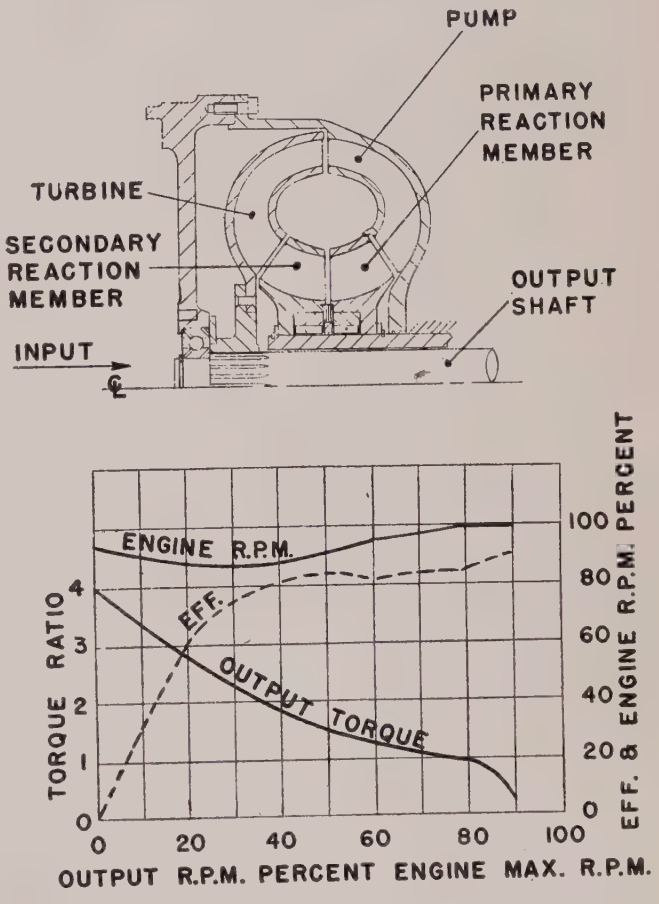


Fig. 3 – Polyphase torque converter

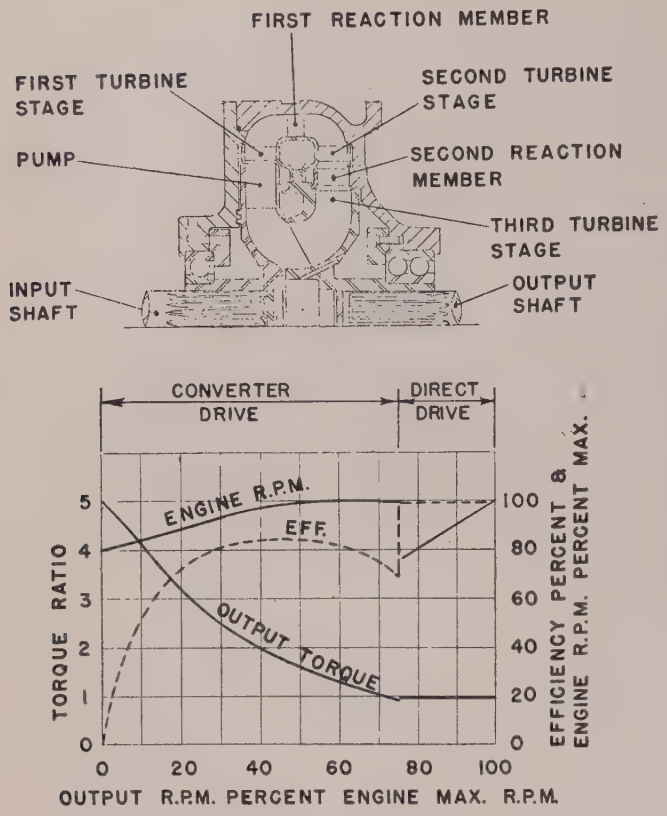


Fig. 4 – Characteristic curve for 3-stage converter

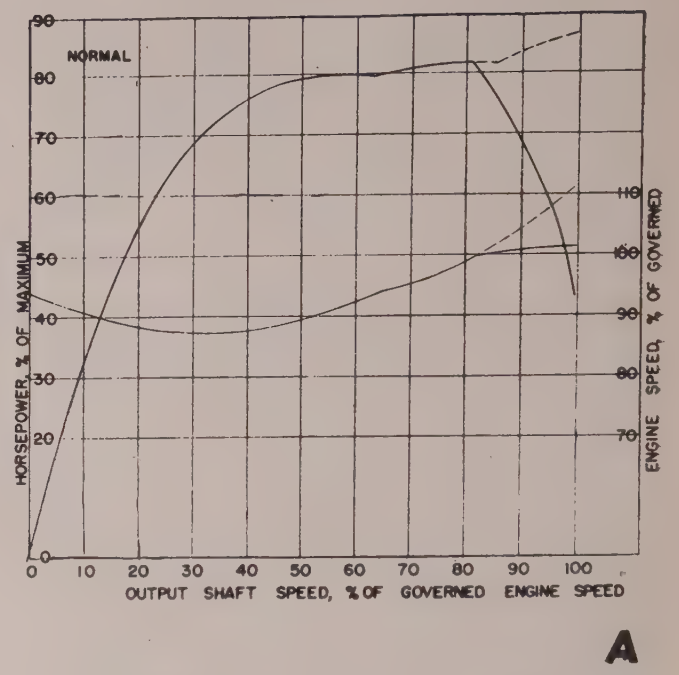


Fig. 5A – Polyphase single-stage torque converter matched to specific engine gives results shown

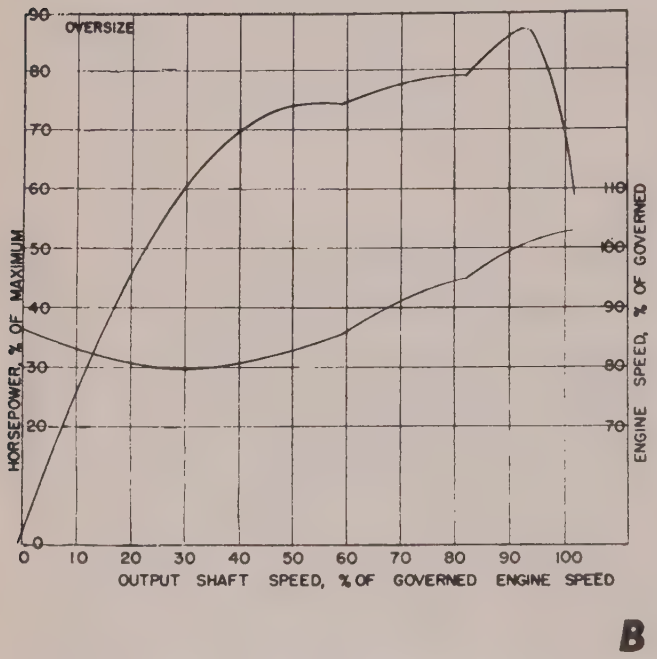


Fig. 5B – Illustration of result of coupling phase with an oversize hydraulic unit

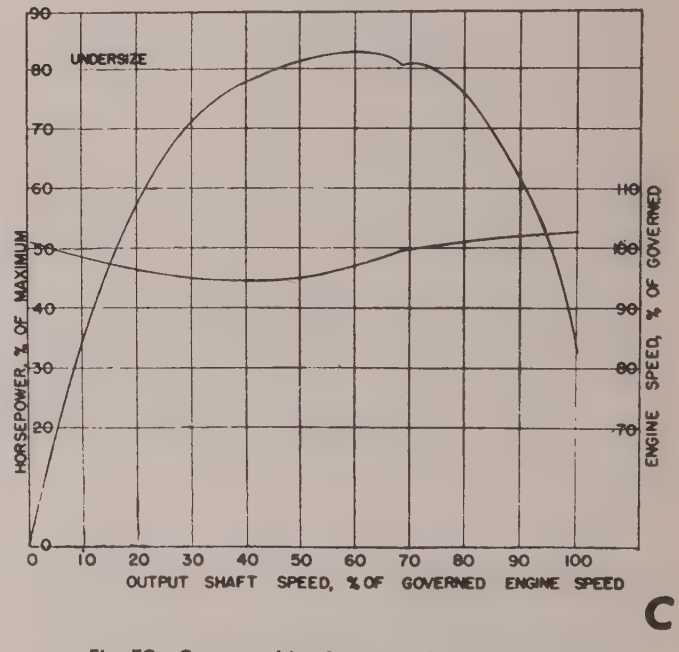


Fig. 5C – Curve resulting from use of an undersize unit

rising efficiency curve, particularly on the approach to the clutch point. The result is maintenance of higher output horsepower at a critical point of the converter range. However, you will note here again the typical pulldown of engine speed within the converter phase by the single-stage converter.

A typical characteristic curve for the 3-stage converter is shown in Fig. 4. Engine speed remains high, providing high input horsepower throughout the entire range of the converter. High output horsepower is then available through the left two-thirds of the converter range. The curve in Fig. 4 illustrates the direct-drive type converter, but without this feature the converter efficiency falls off to zero at approximately 100% of engine speed. Under these conditions, the engine speed is high and the engine fully loaded produces nothing but heat.

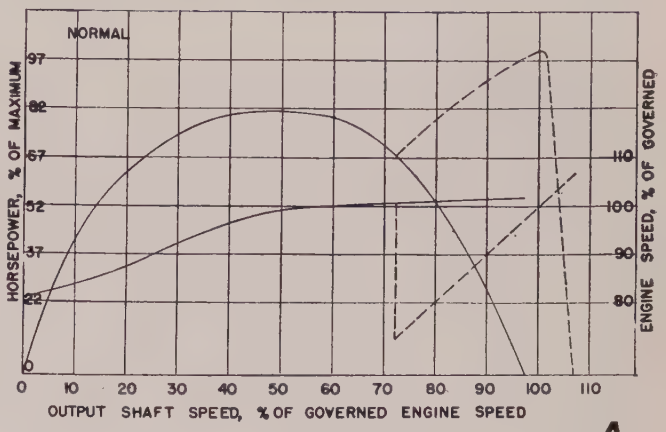
Basic Methods

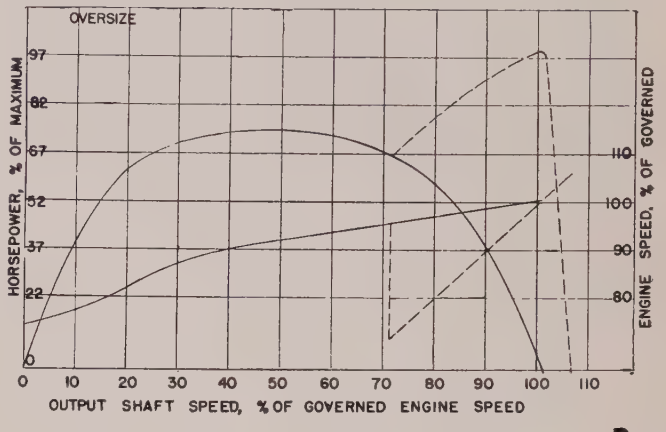
The torque-converter manufacturers have used two basic methods of improving output horsepower performance in the high-speed range of the converter. The 3-stage converter with its fixed reaction blades requires a mechanical by-pass to obtain the performance shown under the direct-drive portion of the curve in Fig. 4. The design is complicated and expensive, and the advantages of a fluid drive are lost when operating in mechanical drive. The single-stage converter design lends itself to the use of a fluid-coupling drive in the high-speed range, but optimum use of this feature imposes severe penalties on performance in the converter range. Fig. 5 and 6 will illustrate graphically our point. Each figure consists of three parts showing the effect of using normal, oversize, and undersize hydraulic units of a given design on an engine developing a specific horsepower at some governed speed. We have used the same engine in each example.

Fig. 5A shows the polyphase single-stage torque converter matched to our specific engine in the normal manner. Rather than show converter efficiency we have chosen to show output horsepower as a percentage of governed engine horsepower. This more nearly reflects what the customer gets for each 100 hp he purchased. Remember, the customer gets less horsepower when using a mechanical transmission if the load variation factor is high enough to justify the use of a torque converter. The driver cannot consistently use the optimum mechanical gear ratio for the load.

Coupling Phase of Converter

Engine speed and converter output shaft-speed each are shown as a percentage of engine-governed speed. The typical characteristic of engine pulldown is evident. If we follow the output horsepower curve we find it rising rapidly through the full converter phase and increasing at a slower rate in the partial converter phase. However, before we can get into the coupling phase the engine governor





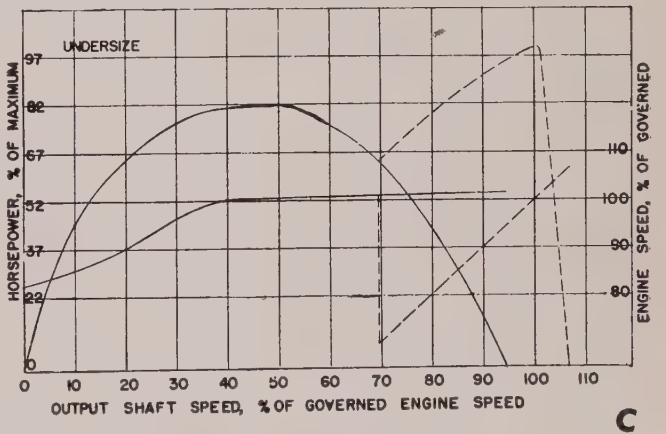


Fig. 6 – Normal application of 3-stage converter to engine is shown in curve in A. Effect of an oversize converter is illustrated in B, and performance curve with undersize converter is shown in C

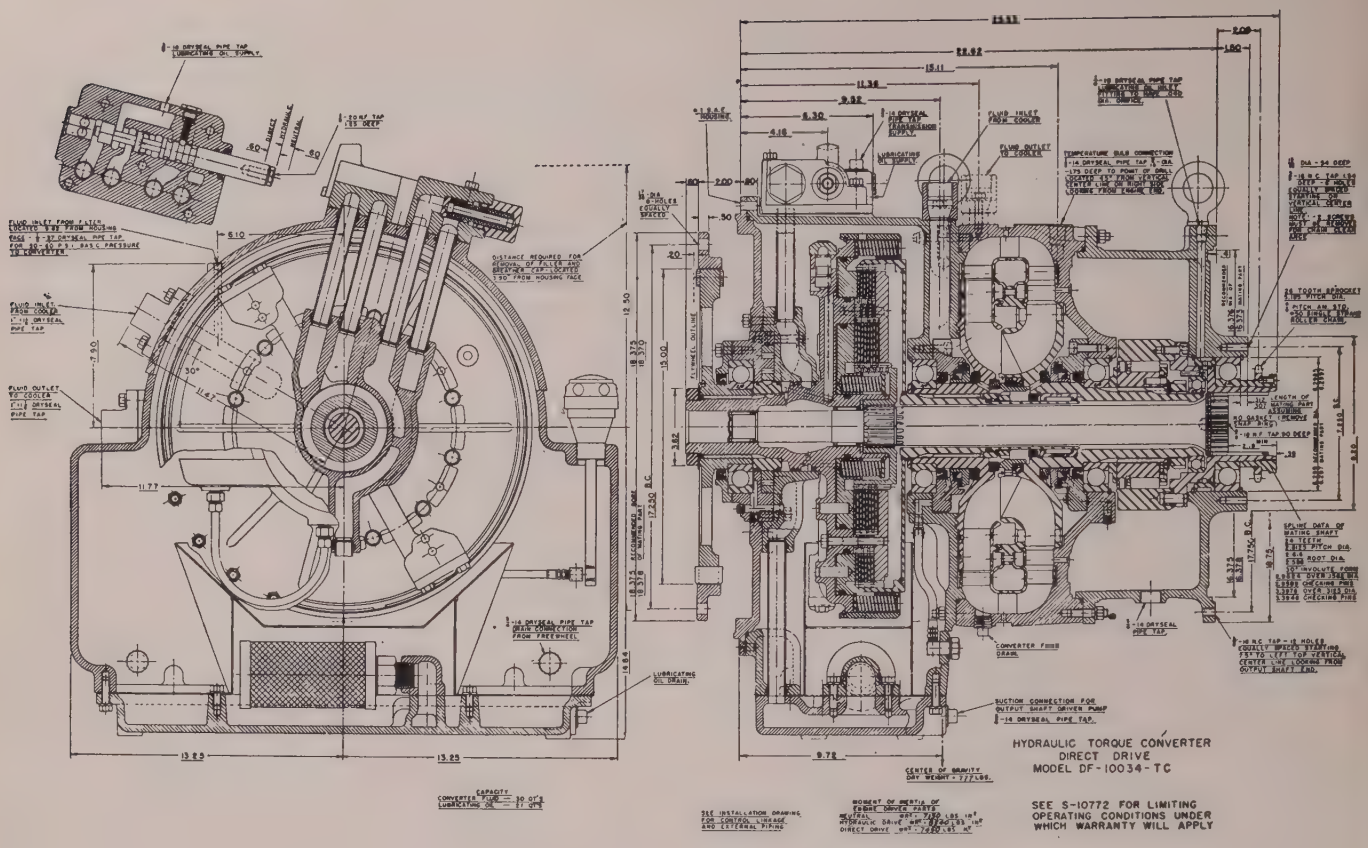


Fig. 7 – Mechanical design of 3-stage torque converter

has taken over, leaving us with only about 83% of maximum power. Why didn't we get into the coupling phase of the converter?

Our answer lies in the fact that when a torque converter and hydraulic coupling are combined into a single fluid circuit design one phase or the other will not have the capacity properly to match the horsepower delivered at the governed engine speed. Since the capacity of our units increases in relation to the cube of the input speed we could make use of the coupling phase provided the engine manufacturer would permit higher engine speeds. The results are shown by the extended dashed lines. Since the engine is usually set up to the top speed for the specific application we would be required to use a hydraulic unit of larger capacity in order to take full advantage of the coupling phase. Fig. 5B illustrates the results.

We now have an output horsepower of about 87% at a speed ratio of 94% but have paid for it in the converter range due to excessive pulldown of engine speed. This is an excellent curve for vehicles with high horsepower to weight ratios where fuel economy is not a major operating factor. The Dynaflow makes full use of this principle but Dynaflow is not involved in horsepower to weight ratios of 5 to 10 hp per ton.

Undersize and Oversize Units

Where the polyphase single-stage torque converter is required to deliver high output horsepower in the full converter phase, it is simply a matter of using an undersize or smaller capacity unit that will permit engine speed to remain high. This is shown in Fig. 5C. We now have a steep rising horsepower curve peaking at 83% at 60% speed ratio and practically all performance is in the full converter phase.

It is important to remember, when studying each of these three curves, that the free-wheeling stator members completely unload the engine under free-running load conditions. This is never true of the fixed stator design unless some mechanical means is provided to bypass the hydraulic unit.

Varying the capacity of the 3-stage converter does not have the direct beneficial results as shown for the single-stage converters in Fig. 5 because

the circuit design permits relatively high engine speed throughout the converter working range. Here we are more concerned with not using an oversize unit.

Fig. 6A shows a normal application of a 3-stage converter to the same engine. Engine pulldown occurs at stall of the converter, and peak engine torque times converter torque ratio is available for breakaway. Engine speed rises rapidly providing high output horsepower over a broad speed range of the converter. We have referred to the fall off of output power to zero caused by the fixed stator design and the requirement for the direct mechanical drive in the high-speed range. The optimum point for shifting from converter to mechanical drive is shown to be at the intersection of the two output horsepower curves. The location of this point is extremely important and should be as far to the right as possible to prevent excessive pulldown of engine speed at this point. Again, engine speed loss is horsepower loss and in this instance can impart a shock load to the driveline due to the use of power shifting.

Fig. 6B illustrates the effect of an oversize converter on our engine. Such an application should never be made since the power output drops from the 80% shown in Fig. 6A to 75% and our shift point for direct drive moved to the left. I used the word *should* purposely because a *normal* matching for sea-level operation would result in the performance shown in 6B if the truck were moved to some high altitude. This point is important when applying any make of converter.

Application of Converter

The torque converter is a closed fluid circuit and once applied is a "leech" as far as power absorption is concerned. Loss of engine power for any reason is compounded by the torque converter due to the effect on engine speed. When using a multi-speed transmission, the operator can select a gear ratio which will permit high engine speed and application of the maximum available power to the wheels. This is also true when using a torque converter-transmission package but the loss of power due to the converter characteristic is not regained without changing converters to suit the altitude, or repairing the engine. For this reason it is common practice to anticipate certain power losses when the engine application and location of operations are known in advance.

If these factors are known then an undersize unit can be applied which will provide a performance curve as shown in Fig. 6C. The horsepower output now peaks at 82% and 50% speed ratio with the shift point too far to the left. Our hope now is that the truck manufacturer ships the truck to the job for which it was figured.

The foregoing sounds complicated but in practice it is not too bad. The important point is to recognize the need for a torque converter that will provide the highest possible output power over the broadest possible operating range. The torque converter manufacturers can go only so far without some help from the engine manufacturers. This point we will cover later on.

Changeover Must Be Automatic

We must have a torque converter that will permit the engine to operate at high output from converter stall to full racing. The torque converter should provide a peak output from 86% to 90% maximum engine power at close to 85 or 90% speed ratio, but also maintain an output horsepower of 60% at 20% speed ratio. The high-speed range beyond the 85% speed ratio point should provide a minimum of 97% power output at 97% speed ratio. The changeover from converter to fluid coupling or mechanical phase must be automatic and not at the discretion of the driver. It must be impossible for the driver to have the option of starting his vehicle in either fluid coupling or mechanical drive even though he may set his selector valve. Such a design will then provide the automatic kick-down from fluid coupling or mechanical drive to converter drive when the load requires such a change, and engine speed will range about 80% of governed speed.

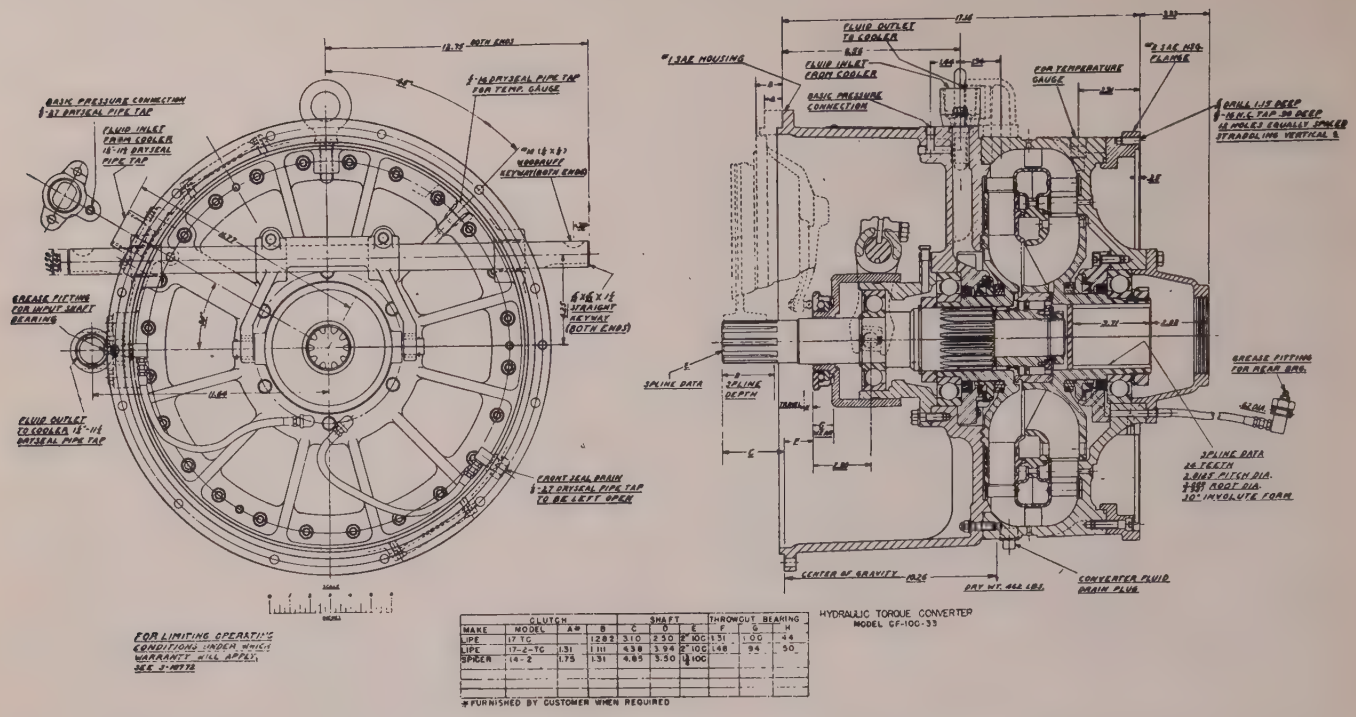
There is one more major point relating to torque-converter design to consider before moving on to the other components of our truck. This point concerns the utilization of the torque absorbing characteristics of the torque converter as a retarder during downhill operation.

The increasing size and speed of today's off-highway trucks make severe demands on the service brakes. Tire sizes have increased along with the brake drums until brake drum sizes have about reached a physical limit. The trucks are becoming underbraked. Since the torque converter is not particularly concerned with the source of power driving it, a logical step was to couple the rear wheels to the converter during downhill operation. Full utilization of the converter was accomplished – a power transmission on adverse grades – a vehicle retarder on favorable grades.

The mechanical problem during the retarder operation was to couple the rear wheels to the impeller of the torque converter rather than the turbine. The impeller is a more efficient pump than the turbine and can therefore provide the maximum fluid circulation within the converter. At the same time the stator blades must be either fixed or locked up by some clutch device.

Design Provides Solution

The mechanical design of the Twin Disc 3-stage torque converter provided a ready solution. Their Model DF, providing a direct mechanical drive, needed only a fourth position of the control valve causing simultaneous engagement of the converter



CLUTCH				SHAFT				THROWOUT BEARING		
MAKE	MODEL	A#	B	C	D	E	F	G	H	
LIPE	17 TC		1.282	3.10	2.50	2 10C	1.31	1.00	44	
LIPE	17-2-TC	1.31	1.111	4.38	3.94	2 10C	1.48	94	50	
SPICER	14-2	1.75	1.31	4.85	3.50	1 10C				

Fig. 8 – Mechanical design of converter showing retarding feature, obtained by use of a freewheel located between turbine and impeller

and mechanical drive clutches. This arrangement may be followed in Fig. 7.

First, consider only the direct drive or inner clutch engaged. The drive from the rear wheels is through the splined hub to the solid shaft, thence to the inner clutch hub. The engaged clutch drives the double clutch carrier, which in turn drives the input spider and the engine. Engine friction horsepower is then available for retarding.

Since the double clutch carrier is being driven by the rear wheels, engagement of the outer or converter clutch will cause the impeller also to be driven by the rear wheels. The turbine, until now, has not been in the picture due to the freewheel mounted between it and the splined output hub. Oil circulation by the impeller causes the turbine to rotate at near impeller speed.

The converter is operating at racing where, due to the fixed stator blades, the efficiency is zero but high horsepower is being absorbed just to circulate the oil against the resistance of the fixed stator blades. What was a problem in the transmission of power is now put to work as a retarder.

Retarding Feature

The retarding feature also proved to be desirable in the short-haul pit truck where the direct mechanical drive was not employed. The Twin Disc Model CF shown in Fig. 8 is equipped with a standard spring-loaded clutch. The retarding feature is obtained by the use of a freewheel located between the turbine and impeller in such a manner that when the rear wheels tend to drive the turbine, the freewheel engages the impeller. The two are then driven at identical speeds along with the engine. The impeller establishes oil flow within the converter. Again, the power absorption of the converter is added to engine friction horsepower for retarding purposes.

Although the retarding ability of both units is similar, the Model DF, Fig. 7, permits optional braking while braking is automatic with the Model CF, Fig. 8, at any time that engine speed drops to turbine speed. The hazard of driver abuse is minimized with the Model CF. Both units provide a push-to-start feature in the event this becomes necessary.

The horsepower available for retarding purposes varies with the capacity of the converter. This is obvious since the converter is first a power transmission device and therefore must be matched to the engine as described before. It also varies as the cube of the input speed and with speed ratio. A simplified horsepower absorption curve is shown in Fig. 9 where certain of the speed ratio capacities have been omitted.

In both models described above, the retarding horsepower is that absorbed by the converter at full racing. The right-hand curve represents this

horsepower value. The working range is from 200 to 300 rpm below engine-governed speed and lower. The rapid increase in horsepower absorption with increase in rpm decreases the hazard of overspeeding the engine. Some operators, cognizant of this fact, are permitting drivers to use higher engine speeds on favorable grades.

The retarding horsepower indicates directly the amount of converter cooling that must be provided under these conditions. At first, the requirement indicates an enormous engine radiator, but when the problem is analyzed, the solution is relatively simple. The engine radiator requirement is increased slightly for two reasons. First, to accommodate a higher average output of horsepower by the engine as compared to its average output when used with a mechanical transmission. Second, to accommodate the power loss within the converter during its function as a power transmission.

The increased horsepower transferred to heat during the retarding function is accommodated by that portion of the radiator capacity not being used by the engine, since during this function, the engine throttle is closed and little engine heat is developed. As a matter of fact, the heat from the converter has been found to maintain uniform engine temperatures without shutters during long periods of favorable grade operation. Therefore, the solution is simply to provide a heat exchanger of sufficient capacity to transfer the converter heat to the engine water. Normally, a 30% increase in capacity is ample.

Here again, close cooperation between the engine, converter, and truck manufacturers is required properly to engineer additional requirements along these lines that are surely just around the corner. While the above retarding feature has been made available without additional gadgets or cost, there is reason to question how long the braking capacity will be adequate.

Already, the single-stage converter manufacturers have units on test. The problem for them has been complicated by the mechanical design of their units but they have the brain power and the desire to solve it.

The aim is to cause the torque converter to relieve the service brakes of their load peaks and provide longer brake life without imposing a major cooling problem on the truck manufacturer. There are hydraulic retarders presently available to handle the extreme braking requirements.

The optimum converter performance just outlined cannot be obtained without some help from the engine manufacturer. He has long been accustomed to designing for mechanical transmissions and therefore has shaped his engine torque-speed curve to provide a peak torque rise of from 7 to 10% at about 70% of engine-governed speed. The engine torque curve should peak closer to 80 or 85% of engine-governed speed and the torque rise should be as high as possible but a minimum of 15% is required.

The engine manufacturer can also help to overcome the inherent speed lag within the torque converter by the use of lighter flywheels designed to make more use of the WR^2 of the torque-converter elements. A more rapid acceleration of the engine from idle or partial speed up to full speed will smooth out the dynamic output horsepower curve of the converter when viewing the overall operating requirements of the vehicle.

The design of the cooling water circuit, including the location of temperature control thermostats, radiator bypass lines and water-circulating pump, must take into account the torque-converter cooling requirement. The design must provide for the location of an adequate oil-to-water heat exchanger for cooling the torque converter during both functions of vehicle propulsion and retarding. The location of the heat exchanger should be such as to permit ready access to the engine for daily maintenance and minor servicing. These requirements are not going to be easily met because most changes will have to be made to engines already in production.

The transmission manufacturer has a different problem. He has to make a completely new transmission. First on the list of requirements is a full-torque shifting transmission. The Allison Division of General Motors is producing the only available unit today. This is an excellent unit for industrial applications where the selection of gear ratios may be made more leisurely or at less frequent intervals than is required in many truck applications. It is doing a good job in open-pit operations but so are some single-speed units because of the limited duty cycle of the operations. The extreme spread between gear ratios is excessive for general truck applications where gear-shifting requirements will be more frequent. Sooner or later, the driver will hang on too long before shifting and impose extreme loading on the driveline as the power shift is accomplished.

Transmission Applications

A transmission to cover the wide range of truck applications must consist of a minimum of four speeds and a 40% spread between ratios. The torque-converter characteristics may dictate five gears in some instances. There appears to be a requirement for a 4-speed transmission with a 60% ratio spread to accommodate the overall tractive effort requirements of the logging truck. The 60% ratio spread is about the maximum to permit power shifting of the transmission while the converter is in the coupling or mechanical drive range.

The on-highway truck will require no less than five speeds and preferably six with a maximum of 30% spread between ratios. The five speeds will be adequate provided the engine-converter package described earlier is made available. This is dictated by the fact that the on-highway truck will seldom use the converter range for truck propulsion.

Fig. 10 shows the general shape of a gradeability curve for a typical heavy-duty truck equipped with a direct-drive type 3-stage torque converter and a closely spaced 4-speed mechanical transmission. You will recognize the horsepower output curve from our discussion of Fig. 6. Those curves radiating upward and to the right from the lower left corner represent the power required for this specific vehicle to negotiate the various grades shown across the top at speeds indicated across the bottom. Gradient values shown are for a fully loaded vehicle.

These curves show rather clearly the relationship of performance through a mechanical transmission both with and without a hydraulic torque converter. If we will consider only the performance in direct mechanical drive it is obvious that gear-shifting frequency is a direct function of gradient requirements. However, optimum performance is a dream never realized because of practical considerations. Regardless of the care exercised in the construction and maintenance of the haul road, gradient variations and spot road conditions can force the driver to operate his truck at a power factor as low as 65%. This is caused by the driver's inability to shift on the grade. As a matter of fact, many operators prohibit shifting on certain grades because of the resultant maintenance cost.

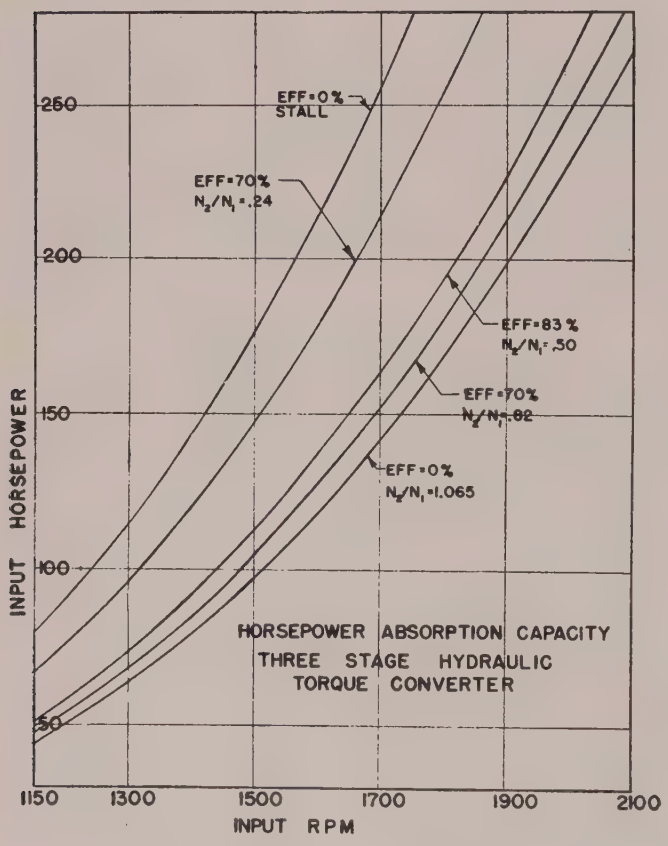


Fig. 9 – Simplified horsepower absorption curve

The performance in converter drive shows that a wide variety of grades may be negotiated without transmission shifting. The combination of the torque converter and mechanical drive provides an extreme range for any one gear ratio of the mechanical transmission.

The selection of the mechanical gear ratio should be at the discretion of the driver since he can anticipate these requirements. The overlap required in an automatic transmission for such heavy equipment would actually slow up overall trip time.

The performance shown in Fig. 10 does not alone indicate the requirement, in this instance, for four speeds or close ratio spacing. However, if you would stretch the performance out to 45 mph for the logging truck and 60 mph for the freight hauler with corrected gradients, a different picture would be obtained. Also, the retarding function requires consideration.

We have shown a typical performance curve for the retarding action of the converter in Fig. 11. The curves shown here are for the same truck as in Fig. 10, again fully loaded. The gradients are favorable and their values, as related to vehicle speed, are 4% higher due to the use of 40 lb per ton rolling resistance as an ignorance factor.

As described earlier, the rear wheels are now driving the converter as well as the engine since all are mechanically connected. You will note two curves, one above the other, for each gear ratio. The bottom curve represents the horsepower absorbed by the converter as illustrated with Fig. 9 while the top curve represents the sum of the horsepower absorbed by the converter and the frictional horsepower of the engine as furnished by the engine manufacturer. These curves are read similar to those in Fig. 10. For example, locate the favorable gradient the truck is descending and follow this curve to its intersection with the top curve of any one of the four curves. The speed at which the truck will be maintained without assistance from the service brakes, can be read directly below on the vertical ordinate. The retarding horsepower required is read to the left along the horizontal ordinate. At the same time the proper transmission ratio is indicated.

In the event no intersection is obtained you can immediately determine the value of horsepower required for any safe speed, as well as the transmission ratio that will be most effective. By following the vertical ordinate above the safe speed, you can determine the difference between the braking horsepower delivered by the engine-converter package and the braking horsepower required. This difference must be provided by the service brakes.

The shape of the converter retarding curve covers a limited vehicle speed range for each transmission ratio. Practical coverage of the entire vehicle speed range dictates the requirement for several gear ratios closely spaced. Such an arrange-

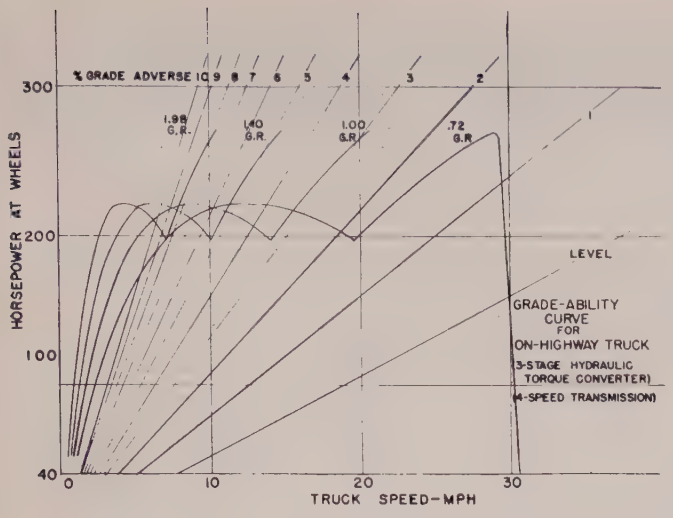


Fig. 10 – Gradeability curve for typical heavy-duty truck

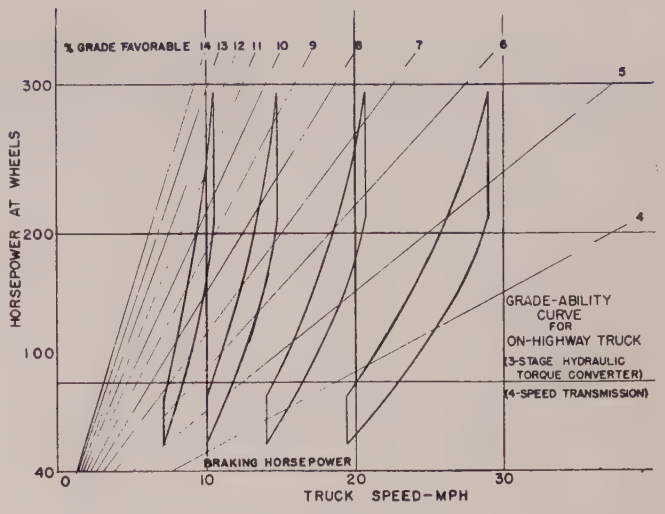


Fig. 11 – Typical performance curve for retarding action of converter

ment, with the transmission behind the converter, provides coverage in the low vehicle-speed range not now covered too well by the currently available water brakes. They are located on the main driveline and at low speeds their capacity falls below requirements.

The logging truck operating on private haul roads is carrying more and more weight. Gradient variations often are frequent, and cycling between adverse and favorable. The haul is a long one compared to mining operations. The closely spaced ratio transmission is a must here for optimum performance in converter range, over frequent gradient changes and for retarding on variable favorable grades. The long return haul empty can be made efficiently at high speeds using mechanical drive.

So much for the ratios and their spacing. Plumbing requirements for the torque converters are bad enough but when we get into hydraulically actuated transmissions the overall plumbing becomes a nightmare. A common fluid system is a must. Then to further reduce the plumbing requirements, the torque converter and transmission should be one package. This package must be as short as possible with only the two oil lines for connections to the heat exchanger showing outside the case. One control lever should control both the converter and transmission. The converter-transmission package should be equipped for amidship mounting in all but the shortest wheelbase trucks to facilitate servicing.

These are the transmission requirements. Actually two transmissions will be required – a heavy-duty 4-speed box with interchangeable ratios for off-highway duty and a light-duty box of 5 or 6 speeds for on-highway duty.

All the truck manufacturer has to do is to place the engine and transmission in his truck in such a manner that it will stay there with all kinds of abuse and be readily accessible for service. Actually this is quite a problem. However, the improvements discussed earlier will materially help.

More and more attention must be given to this matter of service. All too much truck down-time is being experienced because the various components cannot be readily removed for servicing and overhaul. We, the component manufacturers, are inclined to single out that exceptional unit which lasted so long and advertise it to the exclusion of those units which didn't do as well. Sooner or later they all must come out. The faster that can be accomplished, a swing unit installed and the truck placed back in service, the sooner the truck manufacturer can advertise minimum down-time on his truck. It will not be easy to accomplish this on trucks where each customer has special requirements but the ultimate torque-converter transmission described earlier can be so mounted to permit rapid exchange for a swing. Repairs on the transmission can then be made while the truck is at work.

Trucks and related rubber-tired vehicles present one of the largest potential markets available to the torque-converter transmission manufacturer if he will but design for the specific needs of that market. The civil marriage ceremony between the hydraulic torque converter and the heavy-duty truck will not take place until these design requirements are met.

CONTEMPORARY EUROPEAN

APART from airplane engines, which by the very nature of their cylinder arrangements and mountings provide particularly favorable cooling conditions, the aircooling of internal-combustion engines, up to the beginning of the second World War, had been almost entirely limited to small industrial and motorcycle engines of not more than 500 cc (approximately 30 cu in.) per cylinder. During the war, a number of aircooled diesel engines were developed in England, on the Continent, and in the United States, mainly on the instigation of military authorities who wanted small, light, self-contained engines developing from 6 to 15 hp for driving generators, pumps, concrete mixers, and so forth. The specifications demanded that these engines, usually with 1 or 2 cylinders and with a maximum speed of 1750 rpm, should work with a minimum of service and maintenance, be independent of water supply, and operate under extreme climatic conditions.

DURING World War II urgent military needs led to the rapid development of aircooled diesel engines. Specifications required that these engines, usually with one or two cylinders and a maximum speed of 1750 rpm, should be independent of water supply, should work under extreme climatic conditions, and should function with minimum maintenance.

This development is described in this paper, with particular reference to England and the Continent. The author points out that aircooled diesel engines are being successfully built in Europe to meet a wide variety of automotive and industrial uses. Cooling problems appear to have been satisfactorily solved. The author concludes with a comparison of liquid-cooled and aircooled engines.

The Author

WAYNE H. WORTHINGTON (M '17) is director of Engineering Research at John Deere Waterloo Tractor Works. Long associated with the design of farm tractors and equipment, he entered the employ of the J. I. Case Co. as a field engineer, following graduation from Oklahoma A. & M. College. Mr. Worthington joined the Waterloo Tractor Works in 1930 as research engineer in charge of design and development.

AIRCOOLED DIESEL-ENGINE PRACTICE

Wayne H. Worthington, John Deere Waterloo Tractor Works

This paper was presented at the SAE Annual Meeting, Detroit, Jan. 13, 1953.

In spite of prevailing contrary ideas, the diesel engine is naturally compatible with aircooling. A number of factors contribute to this condition:

1. Because of the excess air present, the controlled rate of combustion, and the high expansion ratio, the temperatures of the cylinder head and cylinder walls are lower than those of a spark-ignition engine of the same bore and displacement.
2. The percentage of heat rejected to the cooling media is less than with similar spark-ignition engines.

Furthermore, a number of comparatively recent developments have been particularly useful in promoting the development of the aircooled diesel engine. Of special significance can be included:

1. The swirl-chamber type of combustion chamber, which allows the use of large inlet and exhaust valves, operates with a pintle nozzle having no small holes to clog, and permits excellent cooling of the fuel injector.
2. Development of light metal alloys having high hardness values and resistance to wear.
3. The process of molecular bonding of aluminum alloys to cast iron, which increases heat conductivity at the union and permits choosing metals of high wear resistance for ring groove surfaces and cylinder walls.

As a result, the development of aircooled engines presented less difficulty than was originally anticipated. Large numbers of 1- and 2-cyl aircooled industrial diesel engines are currently produced in England and are giving yeoman service in many parts of the world. It is of especial interest that at least some of these are sold to replace the aircooled gasoline engines used to power American farm machines sold in the export market, such as combines, field or hay and roughage cutters, hay balers, and so forth. In Germany one firm, Kloeckner-Humboldt-Deutz A. G. of Cologne, has reportedly built over 65,000 aircooled diesel engines in 1, 2, 3, 4, 6 and 8-cyl types for farm tractor, road vehicle, and industrial purposes. Current production was recently announced to be at the rate of 7000 cyl per month. A number of smaller manufacturers contribute additional quantities to the total of German engines of this type. In view of the outstanding character of the Deutz operation, it will be discussed first.

During World War II, the severity of Russian winters and the tropical heat of North Africa developed requirements for aircooled diesel engines to power transport vehicles for the German military. As a result, in 1942, the German firm of Kloeckner-Humboldt-Deutz was commissioned to design and build aircooled diesel engines to power 3-ton military trucks. An interesting account of the development of these engines has been published by E. Flatz, managing director of the Deutz works, in the *Motortechnische Zeitschrift* (volume 7, number 3) and by R. Kloss, chief engineer of the same works.

In 1942, the Deutz works converted two existing water-cooled truck engines to aircooling; the 5-liter (305 cu in.) 4-cyl engine of a 3-ton truck, and the 7.5-liter (457 cu in.) engine of a 4.5-ton truck. Both engines had a bore of 110 mm (4.33 in.) and a

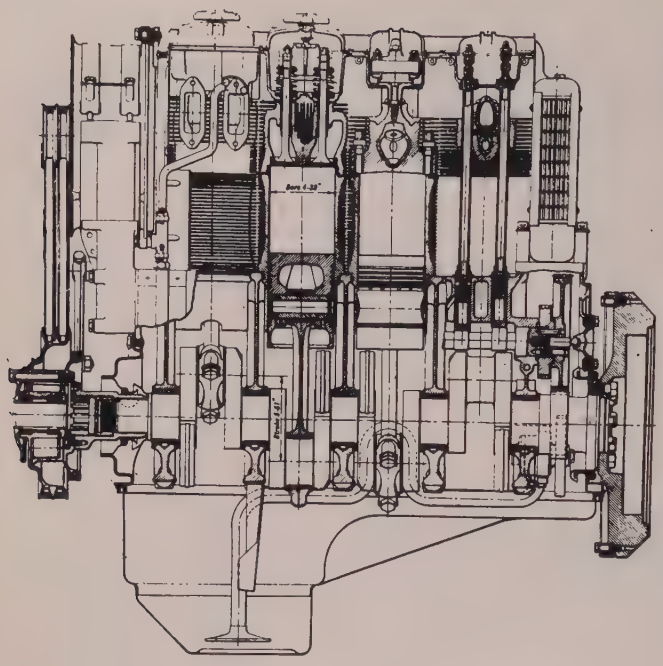


Fig. 1 – Deutz 4-cyl aircooled diesel engine – longitudinal-section

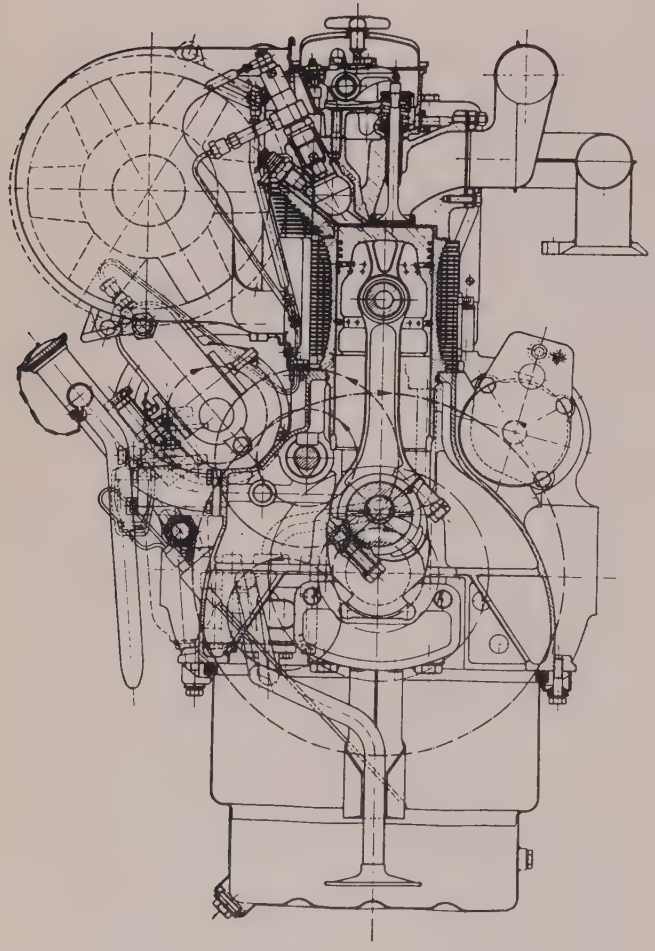


Fig. 2 – Cross-section of Deutz engine shown in Fig. 1

stroke of 130 mm (5.11 in.), and complete with all accessories including cooling blower, developed 80 bhp and 120 bhp respectively at 2250 rpm. Insofar as possible, use was made of the various components of the water-cooled engines, and the majority of castings, with the exception of cylinders and cylinder heads, were produced from existing patterns with only minor alterations. Redesign of the cylinder head provided an opportunity to replace the precombustion chamber of the liquid-cooled engine with a new swirl chamber, which some years previously had been tested experimentally, when it had been found to reduce fuel consumption by approximately 10% and to facilitate starting at low temperatures.

The first engines were running on the test stand within 4 months, and the 4-cylinder engine, which had been given first priority, developed almost from the beginning the required output of 70 bhp at 2250 rpm. In spite of the uncertainties in estimating the necessary areas of the cooling surfaces, as well as the volume and pressure of the cooling air to be delivered by the blower, the results proved to be most satisfactory. A further year of extensive development work ensued, until the engine could meet the severe military acceptance conditions, which included that of starting at a temperature of minus 40 F.

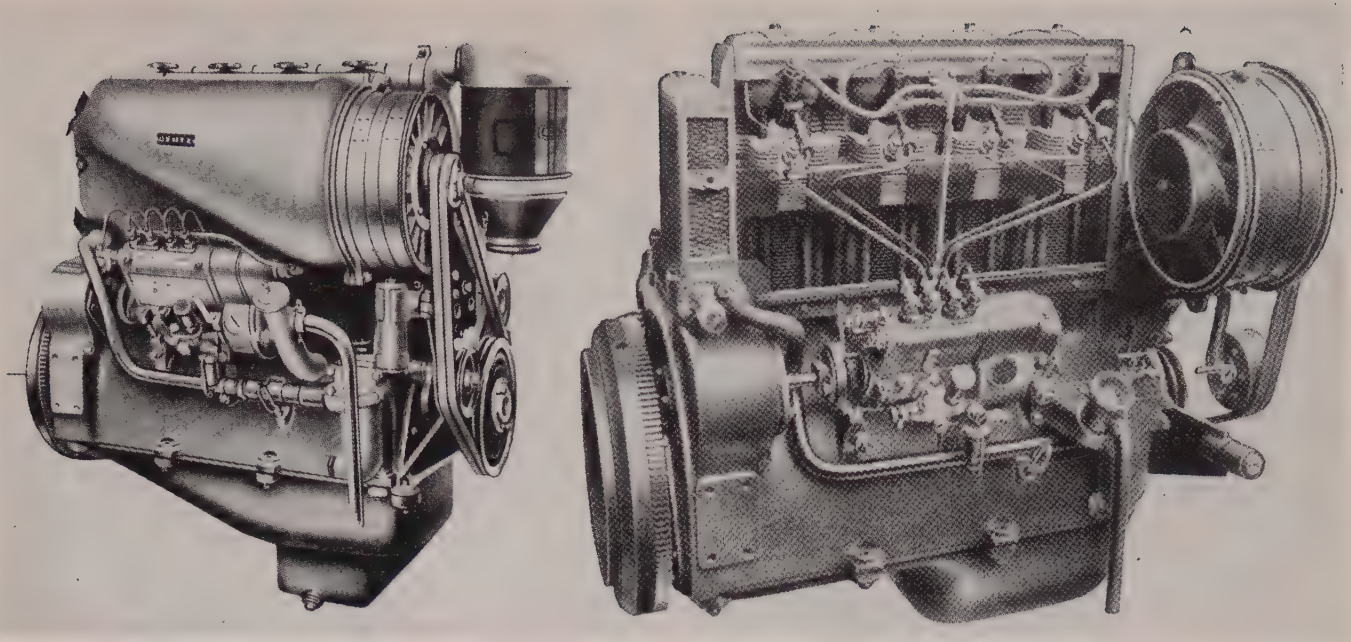
On completion of the actual development work, modified specifications called for an increase of the rated output from 70 bhp to 75 bhp. This was accomplished by the expedient of lengthening the stroke from 130 mm to 140 mm (5.51 in.), which increased the displacement of the 4-cyl engine to 5.3 liters (323 cu in.). Weight of the engine at the end of the development period was 1047 lb with all accessories attached. This resulted in a specific weight of 14 lb per bhp with a specific output of 14 bhp per liter displacement (0.229 bhp per cu in.).

Manufacturing conditions in Germany toward the end of the war made it impossible to produce these engines in any great quantity; but, in spite of these difficulties, more than 1000 engines were fitted into 3-ton trucks built by Magirus A. G., a subsidiary operation of the Deutz works. The results achieved by these military vehicles under severe battle conditions induced Magirus to adapt aircooled engines as standard equipment for their commercial and public service vehicles.

General Arrangement – The 1950 model of the 4-cyl Mark FL514 engine, which differs from its prototype only in small details, is shown in longitudinal section in Fig. 1 and in cross-section in Fig. 2. Additional views of this 4-cyl engine are shown in Fig. 3, right side view; Fig. 4, right side view with air ducts removed; Fig. 5, left side view.

The 8-cyl V-type engine is shown in Fig. 6.

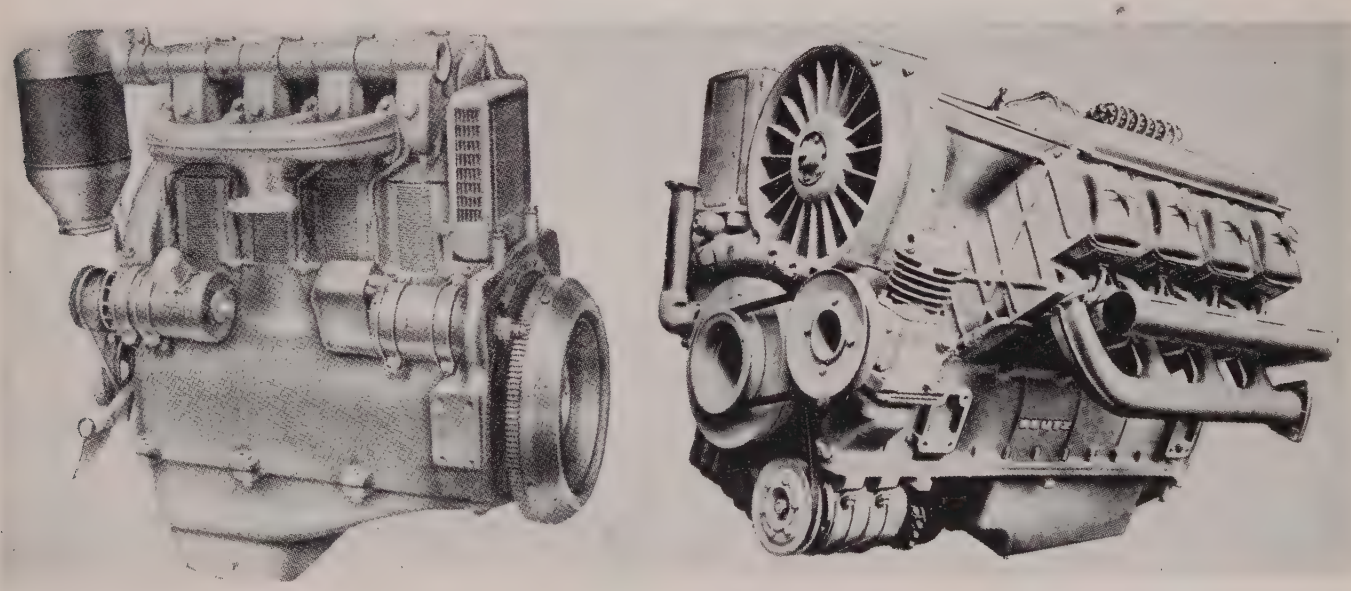
By reducing the weight of the crankcase and flywheel and replacing a number of gray iron castings, such as the blower bracket, inlet manifold, and other similar parts, with welded steel components,



Figs. 3 and 4 – Deutz 4-cyl aircooled diesel engine. Right side view shown in Fig. 3 (left) and same view with air duct removed shown in Fig. 4 (right)

it was found possible to reduce the overall weight of the 4-cyl engine from 1047 to 892 lb. This is equivalent to a weight less than 12 lb per bhp.

General Construction – Four finned cylinders, fitted with light metal heads, are bolted directly to the cast-iron crankcase. The impeller shaft of the axial-flow blower, arranged parallel to the crankshaft axis, is supported on a bracket at the front of the engine and is driven by multiple V-belts from sheaves flexibly mounted at the forward end of the crankshaft. The air duct, leading the cooling air from the blower to the cylinders, is made of sheet steel and may be easily removed. An extension of the duct directs some of the cooling air toward an oil-cooler mounted on the flywheel end of the engine immediately back of the rear cylinder. The induc-



Figs. 5 and 6 – Left side view of Deutz 4-cyl aircooled diesel engine shown in Fig. 5 (left). In Fig. 6 (right) is a general front view of Deutz 8-cyl aircooled diesel engine

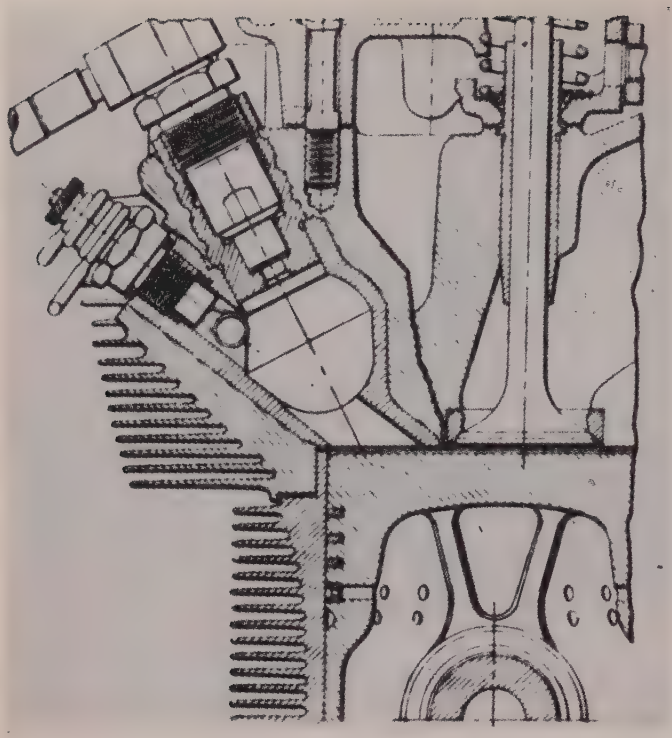


Fig. 7 – Section through combustion chamber of Deutz aircooled engine

tion-hardened crankshaft is supported by 5 lead-bronze main bearings. The camshaft is gear-driven from a driving pinion carried at the rear of the crankshaft.

The short connecting rods ($l/r = 3.9$) are split at the big ends at an angle of 45 deg to the center-line, in order that the rod and piston assembly may pass through the cylinder. The rods are fitted with lead-bronze steel-backed big-end bearings, with phosphor-bronze bushings in the small ends. The overhead valve gear is operated from the camshaft with push rods; it is totally enclosed and lubricated from the oil pump. Bosch fuel-injection equipment is used with the injection pump arranged at a slightly inclined position below the air duct. No timing device has been found necessary.

Cylinder Head – The cylinder head is a heavily finned die casting of low-expansion aluminum alloy. To facilitate molding, the valve gear casing is a separate gray iron part bolted to the cylinder head, with a detachable sheet-metal cover enclosing the rocker arms. The pear-shaped swirl chamber shown in Fig. 7, with volume comprising approximately 60% of the total compression space, is a cast-iron insert integral with the aluminum head. It is placed slightly to one side of the cylinder, with the throat entering the swirl chamber at an angle of about 30 deg to the cylinder axis. With this arrangement, it is possible to accommodate relatively large valves and thus maintain a high volumetric efficiency over the entire speed range.

The injectors are also inclined relative to the cylinder axis and are positioned in the intake end of the cylinder cowling in such a way that they receive the full flow of cold air entering from the blower duct. The main portion of the fuel spray is directed towards the throat of the chamber, with only a small portion impinging on the walls. The efficient cooling of the injector and the self-cleaning properties of the single-hole pintle nozzle reduce to a minimum the tendency to choke as the result of carbon deposits. The glow plug, which is generally used only for starting at temperatures below freezing point, is so arranged that it is not adversely affected by attacks of hot combustion gases. The valve guides are of cast iron pressed directly into the head, with valve seats of special cast iron shrunk in place. Each cylinder head, together with its cylinder, is bolted directly to the crankcase by four long through-studs.

Cylinder and Piston – The cylinder, the outside of which is machined all over, has a total of 28 annular fins extending the whole length of the cylinder. It is tightly fitted to the cylinder head so that both parts may be removed from and assembled to the crankcase without damaging the cylinder-head gasket.

As a cylinder spacing of the aircooled engine is the same as that of the water-cooled engine from which it was originally developed, a portion of the fins between the cylinders had to be considerably shortened. The total cooling surface is therefore somewhat reduced at this point. Although the rate of heat dissipation from the cylinder walls is not constant around the entire circumference, cylinder distortion is kept within tolerable limits.

The pistons are of aluminum-silicon alloy with flat crowns and are each provided with three pressure rings and two lap-joint scraper-type oil rings, one above and one below the fully floating piston pin. The piston clearance, amounting to 0.0043 in. in the water-cooled engine, has been increased to 0.0071 in. to better suit the higher operating temperatures. No special cylinder rings or ring carriers were found necessary to insure reliable continuous operation.

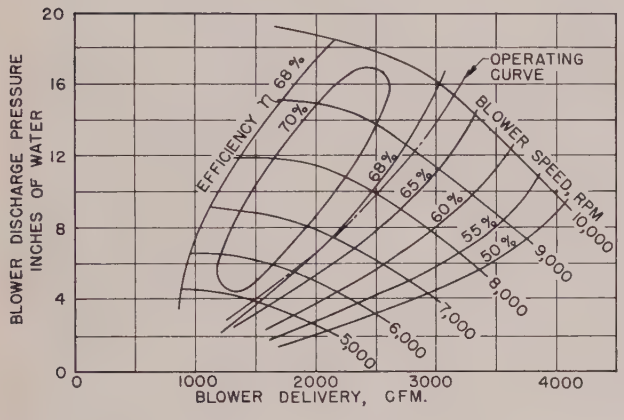
Blower – The blower of the 4-cyl engine delivers approximately 2560 cu ft of cooling air per min at an engine speed of 2250 rpm, while that of the 6-cyl engine delivers 3800 cu ft under the same conditions. In both cases, air is supplied from the blower at a pressure of 10½ in. of water. The axial-type blower was chosen for its simplicity in design, good efficiency, and more especially because it offers favorable conditions for smooth airflow, and good air distribution. The blower, mounted on a bracket at the front of the engine by means of steel bands, is driven by multiple V-belts from sheaves on the crankshaft. A spring-loaded idler pulley fitted with an oil damper serves to maintain belt tension. A safety device is also provided so that in the event the belts break the engine will

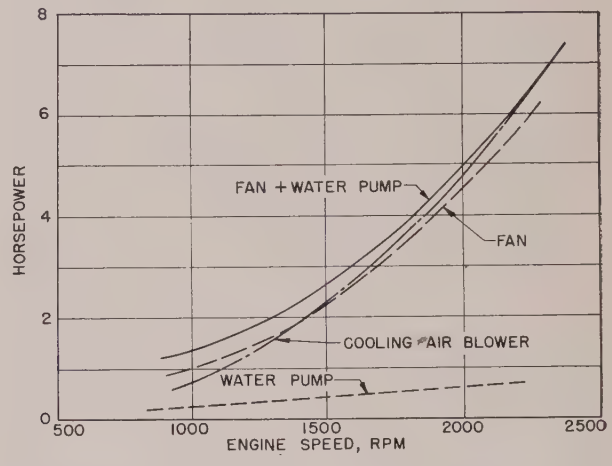
automatically stop.

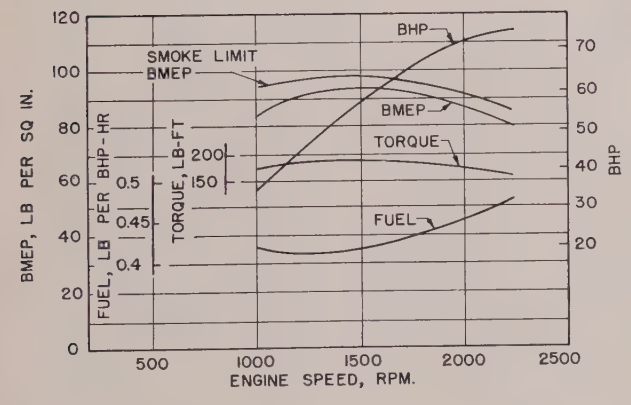
Fig. 8 shows the characteristic curves of the blower fitted to the 4-cyl engine. The contours of constant efficiency show that an efficiency of 70% can be maintained over a considerable speed range. The operating curve in this figure indicates the behavior of the blower when mounted on and tested in conjunction with the engine. It will be seen that the efficiency remains practically constant at 68% for blower speeds between 5000 and 10,000 rpm, corresponding to engine speeds over the range of 1500 to 2700 rpm. The blower for the 6-cyl engine differs from that of a 4-cyl engine only in having an impeller and guide vane annulus of larger diameter. The characteristic curves are similar but, as the influence of wall friction is less pronounced in the larger blower, a maximum efficiency of 75% can be obtained over a speed range of 6000 to 8000 rpm.

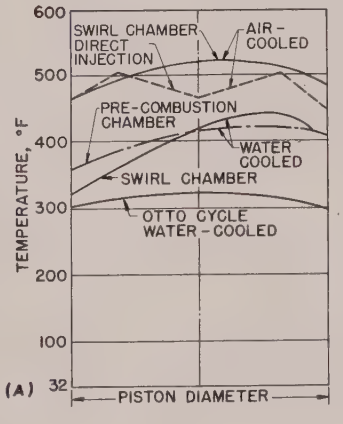
The airflow from the blower is conducted to the cylinders by tapering sheet-metal ducts. No deflecting plates, which tend to cause a pressure drop or induce turbulence, are employed inside the duct. Guide plates attached to the cylinders insure satisfactory air distribution. The power required for driving the blowers of these aircooled engines is much smaller than generally assumed. Fig. 9 shows the power absorbed by the blower of the 4-cyl aircooled engine in comparison with the power required by the water pump and the water-cooled engine. Contrary to widely held opinions, the power absorbed by the fan and the water pump of a similar liquid-cooled engine over the whole speed range is practically the same as that required by the blower of the aircooled engine.

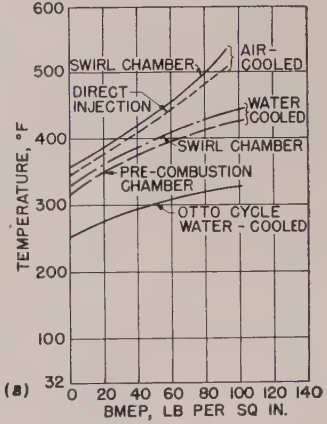
Oil Cooler – In these aircooled engines, the area of the convection surfaces transmitting heat into the cooling air is approximately one-fifth that provided in the radiator of a similar liquid-cooled engine. This results in a high intensity of heat flow, high mean temperatures of heat-conducting parts, and a great deal of heat transmitted into the lubricating oil. In a motor truck operating under normal conditions, the temperature of oil in the engine sump rarely rises more than 100 F above that of the ambient air. Nevertheless, it was



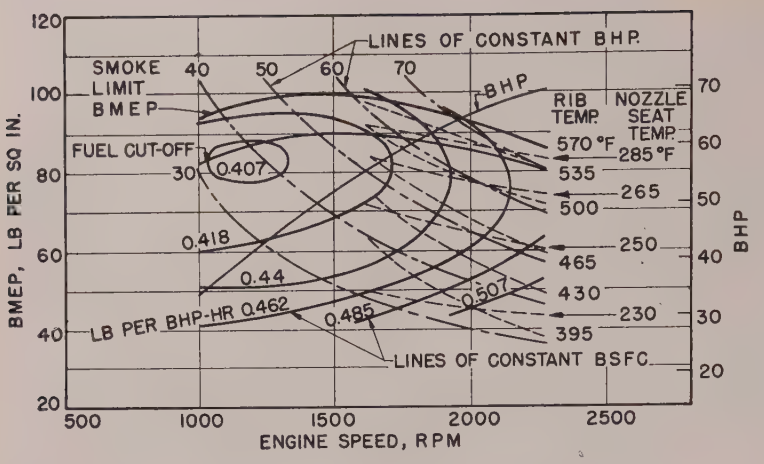


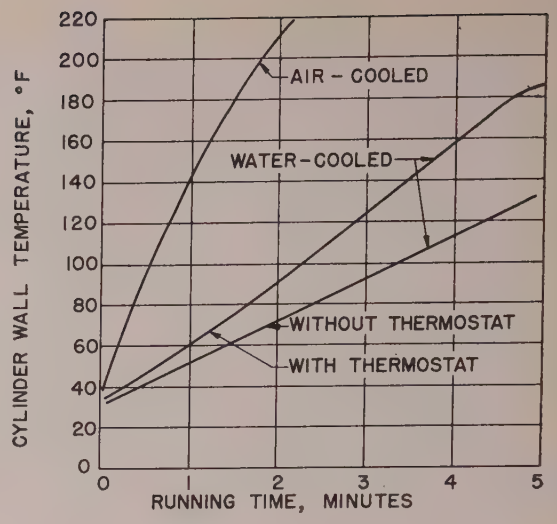






Figs. 8, 9, 10, 11A, and 11B – Blower characteristics of Deutz 4-cyl aircooled diesel engine are shown in Fig. 8 (upper left). Graph at upper right (Fig. 9) shows operating characteristics of same engine. Variable speed performance is illustrated in Fig. 10 (lower left). Fig. 11A (lower right) shows piston temperatures with various combustion systems in Deutz aircooled and water-cooled diesel engine. Fig. 11B (lower right) gives variations in peak piston temperatures with power for various types of combustion and two modes of cooling





Figs. 12 and 13 – Performance characteristics of Deutz 4-cyl aircooled diesel engine (Fig. 12, left). Cylinder-wall temperature after starting for Deutz aircooled and water-cooled diesel engines at rated load, and ambient air temperature of 65 F (Fig. 13, right)

felt that precautions must be taken against damaging oil temperatures. For this reason, an oil cooler is regularly mounted on the crankcase back of the rear cylinder. The necessary cooling air is provided from the blower through an extension of the air duct. To prevent damage to the oil-cooler tubes, as when starting the cold engine at low temperature, a spring-loaded bypass valve is provided.

Performance – The variable speed performance characteristics of the 4-cyl 4.33-in. by 5.51-in. engine, taken at the end of the official 100-hr nonstop acceptance test, are shown in Fig. 10. The torque curve shows excellent lugging characteristics and, at the clean exhaust limit, the bmep has its maximum value of 98 psi at 1500 rpm. The optimum specific fuel consumption of 0.407 lb per bhp-hr was attained at 1250 rpm, increasing to 0.425 lb at 1700 rpm. With the specific fuel consumption below 0.42 lb per bhp-hr over a speed range of 1000 to 1650 rpm, acceptable over-the-road fuel consumption is to be expected.

This specific fuel consumption is not particularly impressive when compared with that of some of the modern direct-injection engines, particularly those using a toroidal combustion chamber, which show optimum specific fuel consumption rates of 0.36 to 0.38 lb per bhp-hr. However, other Continental and most American diesel engine manufacturers, for a variety of reasons, favor the use of divided combustion space in engines for road vehicles.

Proponents of divided combustion chambers appear willing to sacrifice some measure of economy in return for other claimed advantages over the direct-injection engines, such as:

1. The use of larger valves, made possible by the swirl-chamber design, maintains high volumetric efficiency at the higher operating speed range, within the limits of clean exhaust. This is particularly true with cylinders of small displacement.

2. Greater smoothness at low speed and light loads.

3. Use of a pintle-type nozzle with a single large hole, which is less sensitive to carbon formation than the multihole nozzles of direct-injection engines.

4. Less sensitivity to different grades of fuel, particularly to those with a high sulfur content.

Extensive dynamometer and road tests have been run with both 4-cyl and 6-cyl engines using fuels of widely varying viscosities, specific gravity, and cetane numbers. Apart from the usual grades of commercial gas, oil, other fuels such as lignite, tar oil, shale oil, and kerosene were satisfactorily used. With all of these fuels, the rated output could be maintained with a clean exhaust, without the necessity of altering the injection timing.

Supplementary tests were also made with two engines converted to spark-ignition and operated on gasoline and producer gas. In these tests, the swirl chamber was blocked off, the injection pump replaced by a magneto, and new pistons were pro-

Table 1 – Spark-Ignition Engine Tests

Gasoline	
Compression Ratio	5.9/1
Output at 2250 Rpm, bhp	75.0
Specific Fuel Consumption, lb per bhp-hr	0.51
Producer Gas	
Compression Ratio	8.5/1
Output at 2250 Rpm, bhp	55.0
Specific Fuel Consumption, Btu per bhp-hr	10,700

vided to reduce the compression ratio from the normal diesel ratio of 18.51, as required.

The results obtained are given in Table 1.

In Germany, the political situation is unfortunately such that a continuing source of petroleum fuels is not assured. This introduces an imperative requirement that diesel engines be capable of modification so that producer gas may be used when petroleum fuels are not available, a condition to which the Deutz design is well-adapted. Such a situation militates against the use of farm tractors, particularly on the prevailing small farms on the Continent, where food can be grown for draft animals.

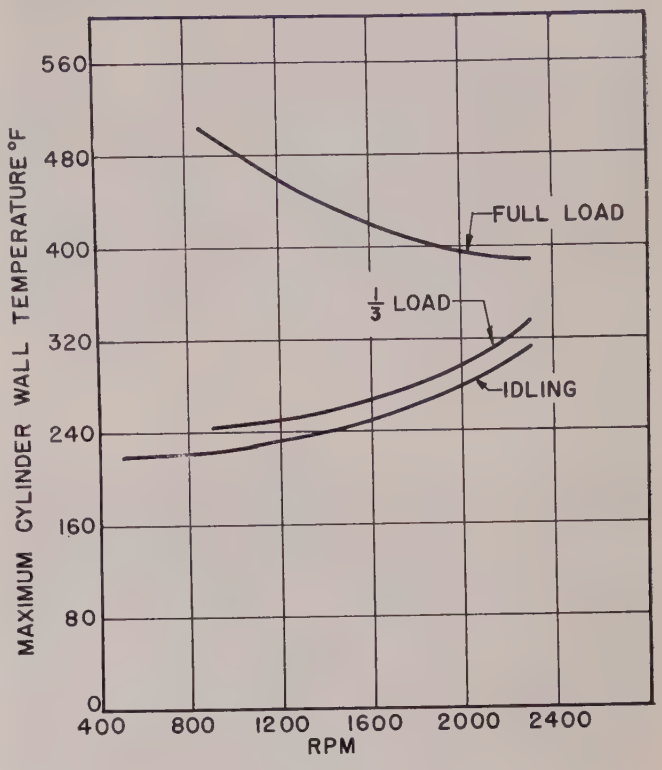
Thermal Performance – Since the Deutz aircooled engines are directly developed from similar water-cooled engines, it is possible to make direct comparisons of the thermal conditions prevailing in each. Fig. 11A shows the comparable temperature gradients across the crown of the pistons of aircooled engines using different types of combustion chambers in comparison with a spark-ignition gasoline engine. Fig. 11B shows the variation in the temperature of the hottest point on the piston crown, as a function of bmep. The authority for these data is A. Pischinger of the Institute for Internal-Combustion Engines, Graz, Germany.

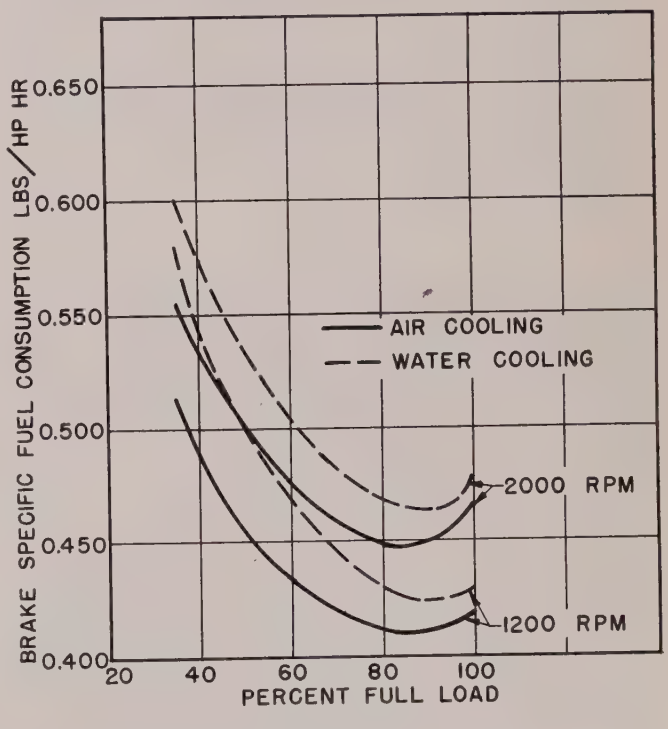
Two significant temperature conditions are shown in Fig. 12. One is the temperature of the fins between the inlet and exhaust ports, which, as determined by a great number of temperature readings, is the maximum temperature of the cylinder head. The other shows the temperature of the swirl chamber in the immediate vicinity of the injector, which indicates the maximum temperature of the nozzle seat. Both temperatures were measured by means of caulked-in copper-constantan thermocouples. The curves show that the temperature of the cylinder head falls with increasing engine speed, if the load changes under the action of the governor. The highest thermal strain to which the engine is subjected coincides with the maximum output, and not, as in the majority of aircooled gasoline engines, with the maximum torque. As the maximum engine speed can be kept safely under the control of the speed governor, overheating of the engine is unlikely to occur. It should be mentioned that the governor is so adjusted that over the whole speed range, the control of maximum mean effective pressure is kept below the smoke limit.

Fig. 13 shows warming-up rates of the cylinder walls after starting of comparable aircooled and water-cooled engines.

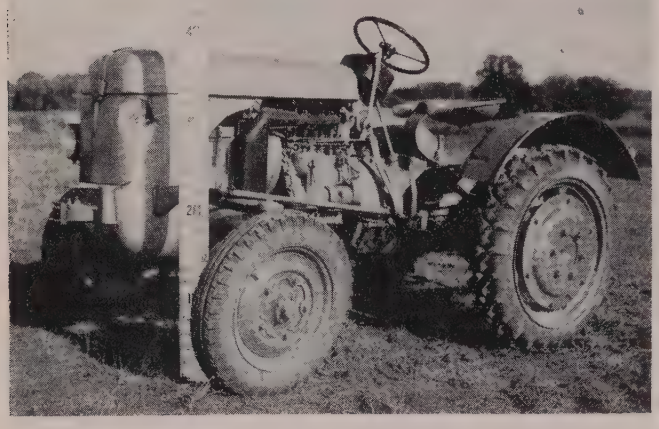
Fig. 14 shows the cylinder-wall temperatures at the hottest point, operating on the road at full load, one-third load, and no load.

Fig. 15 shows the part-throttle fuel consumption characteristics of the comparable water-cooled and aircooled 4-cyl Deutz engines, both with the same swirl-chamber combustion chamber.





Figs. 14 and 15 – Maximum cylinder wall temperature versus rpm at various loads under road operation – Deutz 4-cyl aircooled engine (Fig. 14, upper). Variable load fuel consumption characteristics with air-cooling and water-cooling in Deutz 4-cyl engine with swirl chamber (Fig. 15, lower)



Figs. 16 and 17 – Allgaier Porsche tractor. Left side view shown in Fig. 16 (upper), and left front view shown below (Fig. 17)

In general, the performance of the aircooled engines compare most favorably with similar water-cooled engines.

Road Performance and Wear Rates – During the early stages of development, road tests were carried out with two 4-cyl aircooled diesel engines mounted in 3-ton Magirus trucks, the total distance traveled by each truck exceeding 75,000 miles. Parallel tests were made over the same distance, and under similar operating conditions with other trucks fitted with water-cooled diesel engines and with aircooled engines running on producer gas. After completion of these tests, the engines were examined, new piston rings fitted, and the tests continued. The wear measured after 75,000 miles was found to be practically identical in the aircooled and water-cooled engines. In each case, the maximum cylinder wear, between the individual cylinders, was between 0.0110 in. and 0.118 in. It appears that the higher cylinder-wall temperature of the aircooled engine had almost the same effect upon cylinder wear as the low temperature of overcooled cylinder walls. This latter condition frequently exists in water-cooled engines when running for long periods at light or partial load.

In the absence of further knowledge as to the fuel used in these tests and the efficiency of the air cleaners, no intelligent comparison can be made with our own background data.

An examination of the aircooled spark-ignition engines running on producer gas showed very different results – the cylinder wear was very much smaller than that of the diesel engine and operation was maintained for 180,000 miles before overhaul. This, of course, may have been partly due to the lower thermal and the mechanical stresses to which the engine was subjected, but it also seems to indicate that the cylinder walls of the gas engine are immune from the corrosive effects of fuel condensation and that, for the same reasons, the lubricating oil is not affected by crankcase dilution.

To minimize the effect of corrosion and oil dilution, all Deutz aircooled engines are currently fitted with positive crankcase ventilation. Partly due to this positive crankcase ventilation and partly to modified materials with better wear characteristics, the cylinder life of the aircooled engine between reconditioning intervals is today about double that of the pre-war water-cooled engine. This low wear has been independently confirmed by conversations with a number of independent engineering authorities.

A further contributing factor accounting for the better cylinder wear of the aircooled engine might be its relative insensitivity to the very high sulfur content of present-day diesel fuels. On the Continent, sulfur content is reported to currently run as high as 2.3%. The aircooled engine warms up rapidly after starting and maintains a favorable mean temperature even at very high loads; the cylinder-wall temperature in all operating conditions is high enough to prevent any appreciable condensation of sulfur trioxide in the form of sulfuric acid, thus eliminating one of the causes of cylinder corrosion and oil contamination.

Behavior at Extreme Temperatures – Water at extreme temperatures is not an ideal cooling agent due to the risk of freezing and evaporation. It was to be expected that the aircooled engine would work satisfactorily at low atmospheric temperatures, whereas its behavior at very high temperatures was not easy to predict. In numerous tests with temperatures as low as —14 F, starting could generally be accomplished after preheating the swirl chambers for one minute by means of the electric heater plugs. At lower temperatures, however, it is necessary to warm up the engine with an air heater. This heater, in principle, consists of a series of guide plates which conduct a current of hot air from the blowtorch towards the discharge end of the cooling-air duct, thus warming both cylinders and cylinder heads.

At air temperatures of —40 F, the engine could be started after application of the blowtorch for 10 min, and preheating the swirl chambers for 1

min. It took about 5 more min for the engine to arrive at a safe operating temperature, that is, one which eliminated the risk of lubricating oil dilution. The total time required to start the engine from cold to the vehicle ready to move under its own power is only one-third of that necessary for water-cooled engines under these conditions.

At high temperatures, the aircooled engine also proved to be highly efficient and most reliable. In a 2-hr test, the engine was operated at full speed and full load with an ambient air temperature of 140 F, without excessive overheating. Although the oil in the engine sump reached a temperature of 220 F, no damage was done to the cylinder walls, piston rings, bearings, or other rubbing surfaces.

Noise – The absence of cooling jackets allows many of the internal noises of the engine to be transmitted directly into the air without dampening. The cooling-air blower is necessarily much noisier than a conventional cooling fan. However, with the engine compartment well-insulated against sound transmittal, the noise reaching the driver in a truck cab is sufficiently low that it was felt necessary to provide a safety stop, which stops the engine should the blower belts break, rather than to depend upon the operator to sense the failure.

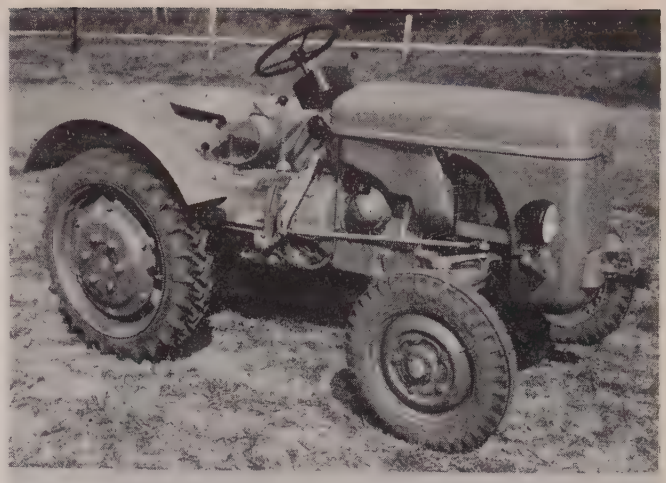
Maintenance Service – Road operations show that the servicing of aircooled engines is in no way different from that of their water-cooled counterparts. Many of the maintenance operations are simplified, as individual cylinders and heads are easier to dismantle and assemble than is the case with en bloc engines.

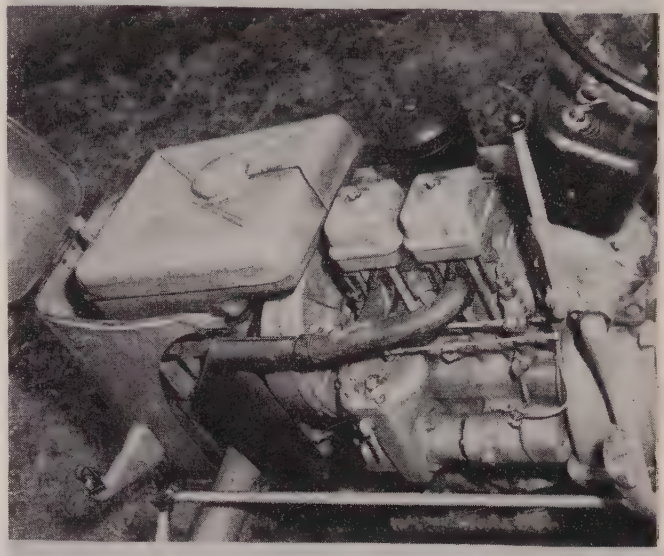
General – The use of aircooled engines will probably be restricted to small cylinder bores with mean effective pressures not exceeding 100 psi. The present development of the Deutz engine is very extensive, since all of their engines are built around a single-cylinder design with its individual head, combustion chamber, piston rings, and connecting rods.

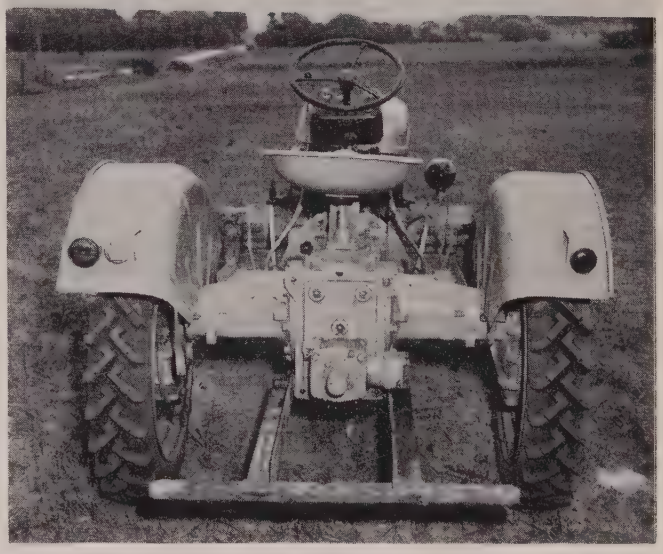
One- and two-cylinder engines are now available for small farm tractors, and the 8-cyl V-type engine is used to power the largest trucks and buses, as well as heavy-duty power units such as those used to power oil well drilling rigs. The *New York Times* of Nov. 5, 1951, reported that a Dutch Civil Defense order was placed for 1117 Deutz aircooled engines to power fire-fighting equipment.

Allgaier Porsche 2-Cyl 4-Cycle Aircooled Tractor Engine – One of the last designs of the noted German engineer, Ferdinand Porsche (designer of the original Volkswagen), was a small 2-cyl 4-cycle aircooled diesel-powered tractor built by Allgaier Werkzaugbau, G.M.B.H., with main offices in Uhingen, Wittenburg, Germany. This interesting little tractor is shown in Figs. 16 to 25. A cross-section is shown in Fig. 21.

The specifications of the engine are given in Table 2.







Figs. 18, 19 and 20 – Allgaier Porsche tractor. Fig. 18 (upper) shows right side view. View of engine with hood removed is shown in center picture (Fig. 19). Rear view of tractor is shown in lower picture (Fig. 20)

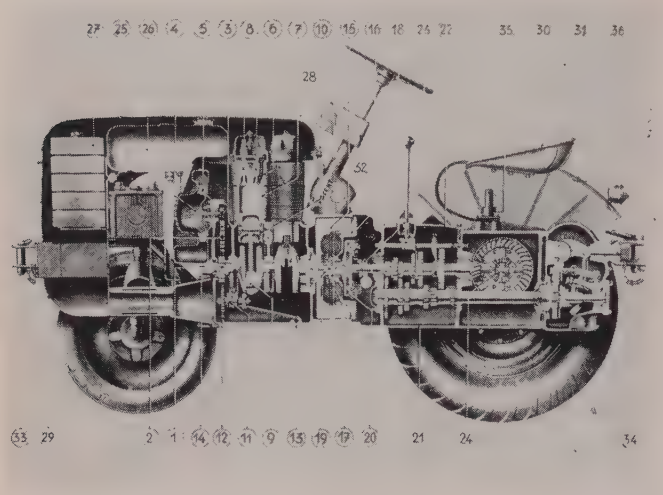


Fig. 21 – Sectional view of Allgaier Porsche tractor

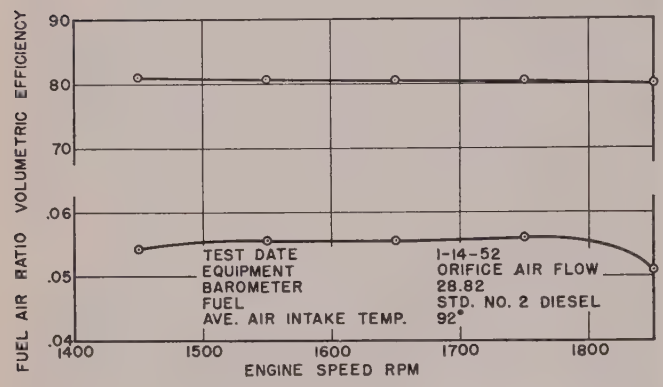


Fig. 22 – Volumetric efficiency and fuel/air ratio of Allgaier Porsche diesel tractor

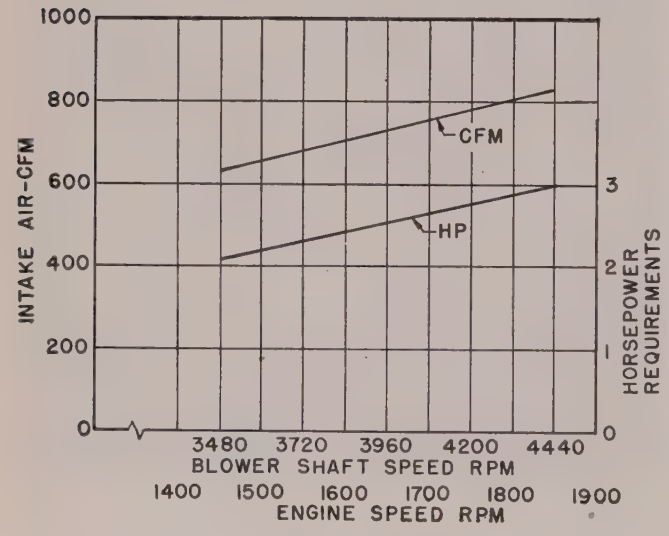


Fig. 23 – Cooling blower discharge characteristics of Allgaier Porsche diesel tractor

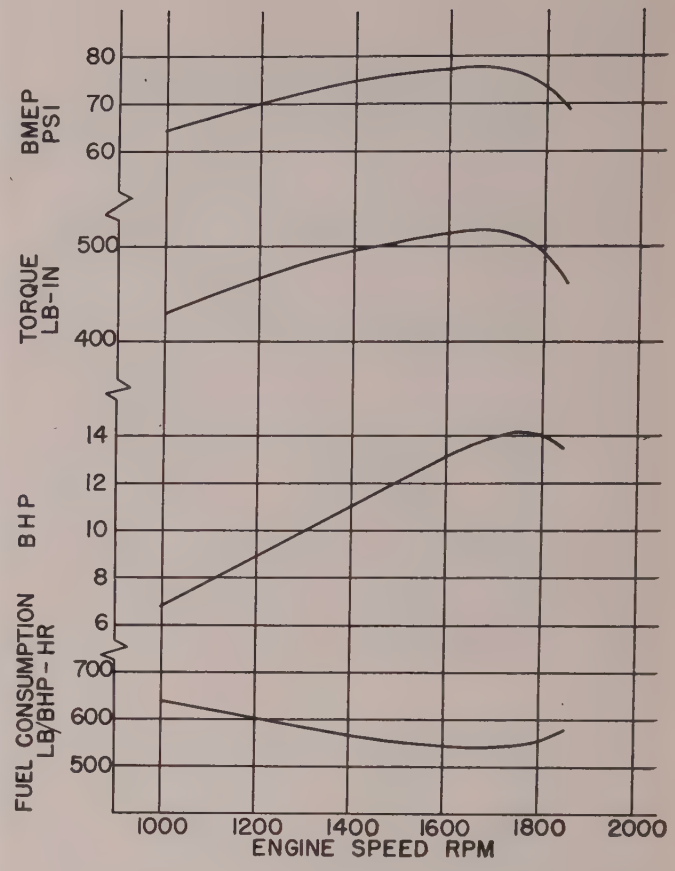


Fig. 24 – Variable speed characteristics of Allgaier Porsche diesel tractor

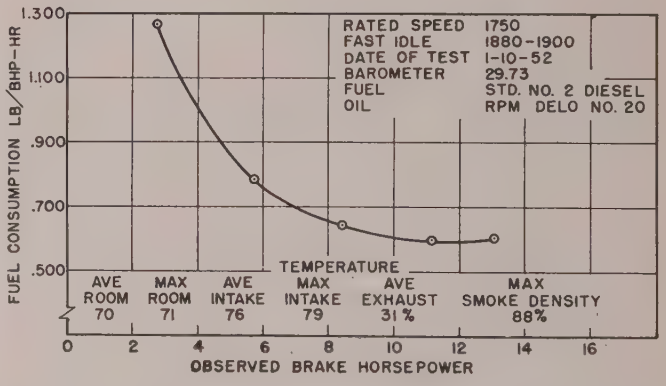


Fig. 25 – Variable load characteristics of Allgaier Porsche diesel tractor

Unfortunately, it was never possible to get the engine to deliver its advertised performance. After a 100-hr break-in period, followed by careful adjustment of tappet clearances and injection timing, the variable speed performance was that shown in Fig. 24. Corresponding volumetric efficiency and fuel/air ratio data are shown in Fig. 22.

Cooling air is blown to the cylinders and heads by means of a gear-driven centrifugal blower having power and output characteristics shown in Fig. 23. The individual cylinders are of cast iron, with sand-cast light metal head.

Individual Bosch injection pumps, operated directly from the camshaft, supply fuel to the pintle-type injectors of each cylinder. The combustion system is of the swirl-chamber type and is provided with an electric glow plug to facilitate starting.

Stihl Model 131 Single-Cylinder 2-Cycle Aircooled Diesel Engine – An interesting German-built single-cylinder 2-cycle aircooled diesel engine, which is provided with mountings adapting it to a wide range of stationary and industrial applications, is the Stihl Model 131 engine. Fig. 26 shows the general appearance of the engine, and Fig. 27 gives its general outside dimensions.

The specifications of this engine are given in Table 3.

It will be noted that this engine employs a belt-driven axial blower, which directs the cooling air at the exhaust and injector side of the cylinder head. Bosch fuel pumps and injectors are used, with injection directly into the combustion chamber.

The specific fuel consumption at speeds ranging from 1200 to 1950 rpm is shown in Fig. 28. The marked improvement in output from 1850 to 1950 rpm undoubtedly reflects the incidence of resonance in the inlet system. At maximum power output at 1950 rpm, the engine develops approximately 51 psi bmep, which is excellent for this type of scavenging.

British Aircooled Diesel Engines – While no automotive aircooled diesel engines have been developed in Germany, information has been made available on 1- and 2-cyl engines built by three long established British diesel-engine manufacturers, namely, Petters Small Engine Division, Brush Aboe, Ltd.; Enfield Industrial Engine Co. (subsidiary of the Enfield Cycle Co., Ltd.); and Armstrong Siddeley Motors, Ltd. Descriptions and performance data of these engines follow.

Petter Single- and Twin-Cylinder Aircooled Engine – This manufacturer produces 1- and 2-cyl vertical aircooled engines. The Model AVA-1 1-cyl engine, with a rating of 3 to 5¼ bhp at 1000-1800 rpm, is shown in Fig. 29. The AVA-2 twin engine, with a rating of 6 to 10½ bhp at 1000-1800 rpm, is shown in Fig. 30. A cross-section through the cylinder, applying to both, is shown in Fig. 31. The variable speed performance characteristics of the 2-cyl engines are shown in Fig. 32.

Brief specifications of both engines are given in Table 4.

Table 2 – Allgaier Porsche Tractor Engine

Number of cylinders	Two; individually cast
Timing of firing order, deg	180; 540
Arrangement	In line
Cooling	Air
Valve arrangement	Overhead, in-head
Types of heads	Individual
Operating cycle	4
Rated speed, rpm	1750
Bore, in.	3.54
Stroke, in.	4.25
Displacement, cu in.	83.25
Compression ratio	19.1/1
Maximum belt hp at rated speed	18.5 (advertised)
Oil cleaner	Centrifuge

Table 3 – Stihl Aircooled Diesel Engine

Bore, in.	3.55
Stroke, in.	4.75
Displacement, cu in.	46.36
Rated speed, rpm	1850–2000
Weight (less accessories), lb	209

Table 4 – Specifications for Two British Engines

Model	AVA-1	AVA-2
Number of cylinders	1	2
Bore, in.	3.15	3.15
Stroke, in.	4.33	4.33
Displacement, cu in.	33.9	67.8
Compression ratio	16.5/1	16.5/1
Rated speed, rpm	1800	1800
Piston speed, fpm	1300	1300
Maximum hp at 1800 rpm	7.5	15.0
Bmep at maximum hp	97.8	97.8
Maximum torque at 1300 rpm, lb-ft	22.0	44.0
Bmep at maximum torque, psi	98.1	98.1
Cyclic variation:		
1500 rpm	1/81.5	1/67
1800 rpm	1/119	1/99
Fuel tank capacity, U. S. gal	2.16	2.16
Sump oil capacity, U. S. pt	7.2	17.3
Net dry weight with standard equipment, lb	390.0	558.0

Cooling air blowers are of the flywheel type and deliver air through suitable ducts to the fuel injector side of the cylinders and heads. Individual fuel pumps, each operated by a lever directly from the camshaft, pump fuel through pressure lines of equal length directly to the injectors.

The fuel injectors and pump are manufactured by the British firm, Bryce Bergers, Ltd. Nozzles are of the multihole type. They are provided with three holes, arranged to give an equally spaced spray pattern. Of special interest is the finned aluminum sheath surrounding each of the fuel injectors, which conducts heat away from the injector and dissipates it into the cooling-air stream. It will be seen that the combustion chamber consists of a semispherical depression in the cylinder head.

The connecting rod and main bearings are of the steel-back precision type, with white metal lining for the main bearing and lower half of connecting rod position, and copper-lead lining for the upper half of the rod position.

The rating of these engines is in accordance with British Standard Specification 649-1949, with power deductions as follows:

A. For every inch reduction in atmospheric pressure below 29.5 in. mercury (corresponding to 500-ft altitude) deduct 4%, or at a rate of 4% per 1000-ft altitude above 500 ft, whichever is smaller.

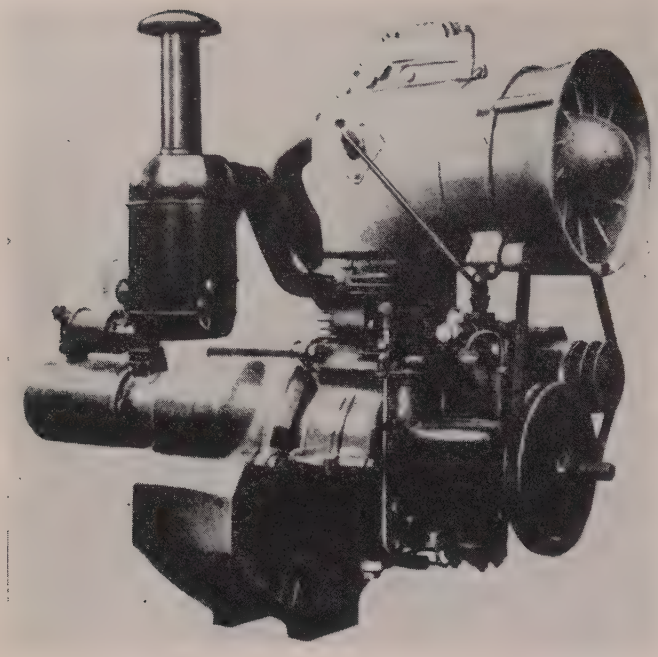


Fig. 26 – General view of Stihl 2-cycle diesel engine (model 131)

B. For any increase above the engine room (ambient) temperature of 85 F, deduction should be made at the rate of 2% per 10 deg F.

Continuous rating represents power available for continuous uninterrupted service, applicable to installations such as generators, centrifugal pumps, and so forth.

Intermittent rating represents power available for continuous periods not exceeding 1 hr, and is applicable for power shovels, saw mills, crushers, and so forth.

Gross rating represents the maximum power available on the dynamometer, and is applicable for automotive and tractor applications.

Enfield Single- and Twin-Cylinder Aircooled Diesel Engines – Through the courtesy of *Automotive Industries,* information has been received regarding the British Enfield 1- and 2-cyl aircooled diesel engines. The 1-cyl engine is of the conventional vertical type, but the twin is of the opposed type. Both are cooled by air supplied by centrifugal fans cast integral with the flywheel. The cooling is intended to be ample for tropical conditions with ambient air temperatures up to 130 F. The 1-cyl engine is shown in Fig. 33, and the twin engine in Fig. 34. A section illustrating the detail construc-

Table 5 – Specifications for Enfield Engines

	Single	Twin
Bore, in.	3.365 (88 mm)	3.365 (88 mm)
Stroke, in.	3.937 (100 mm)	3.937 (100 mm)
Displacement, cu in.	34.65	63.3
Compression ratio	19.5/1	19.5/1
Specific fuel consumption at 1500 rpm	0.44 lb per bhp	0.41 lb per bhp
Speed range, rpm	1200–1800	1200–1800
Piston speed, fpm at 1800 rpm	1180	1180
Maximum bhp at 1800 rpm, 1-hr rating	7.50	15.00
Bmep at maximum rating, psi	95.5	95.5
Net weight with accessories, lb	224.0	310.0

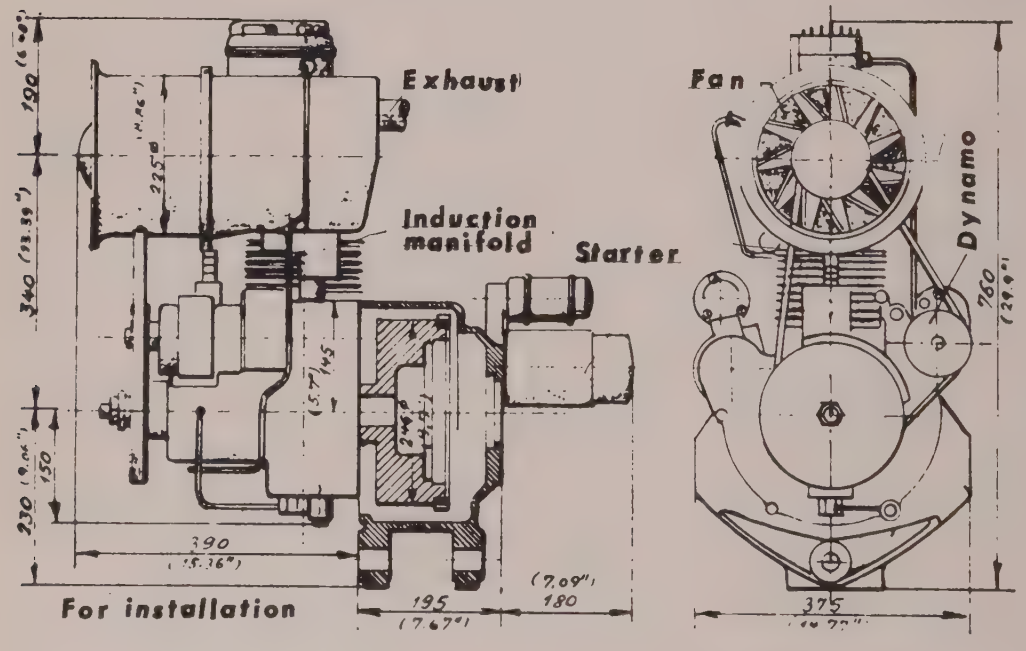


Fig. 27 – Dimensional drawing of Stihl 2-cycle diesel engine shown in Fig. 26

tion is shown in Fig. 35. The general details are shown in Fig. 36, and the performance characteristics are given in Fig. 37.

Specifications of these engines are given in Table 5.

Cylinder is heavily finned, of low-expansion aluminum alloy, with cast-in gray iron liner.

Cylinder head is heavily finned, of low-expansion aluminum alloy, with cast-in combustion chamber, valve seats, and inlet and exhaust ports of special iron.

Piston is of low-expansion silicon-aluminum alloy, with full-floating pin, three compression rings, and two scraper rings. The top compression ring is chrome-plated.

Enfield aircooled engines are designed for satisfactory operation in temperatures up to 130 F.

The fuel injection system is of C.A.V. (British) manufacture, with pintle-type, single-orifice nozzles.

The following information was also given:

A. Blower capacity is 45 cfm per maximum bhp.

B. Maximum temperature of piston crown and combustion chamber, 480 F above ambient air.

C. Rate of wear on cylinder walls is approximately 0.001 in. per 1000 hr. Average wear in similar water-cooled engines is from 50% to 100% higher, due to slower warm-up time.

Starting may either be manual, with a decompression gear, or a 12-v starter is available. An ether starting aid can be used to insure easy starting under very cold conditions.

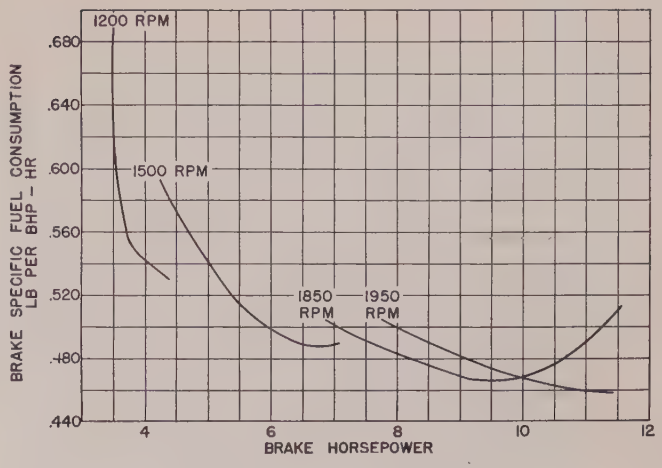
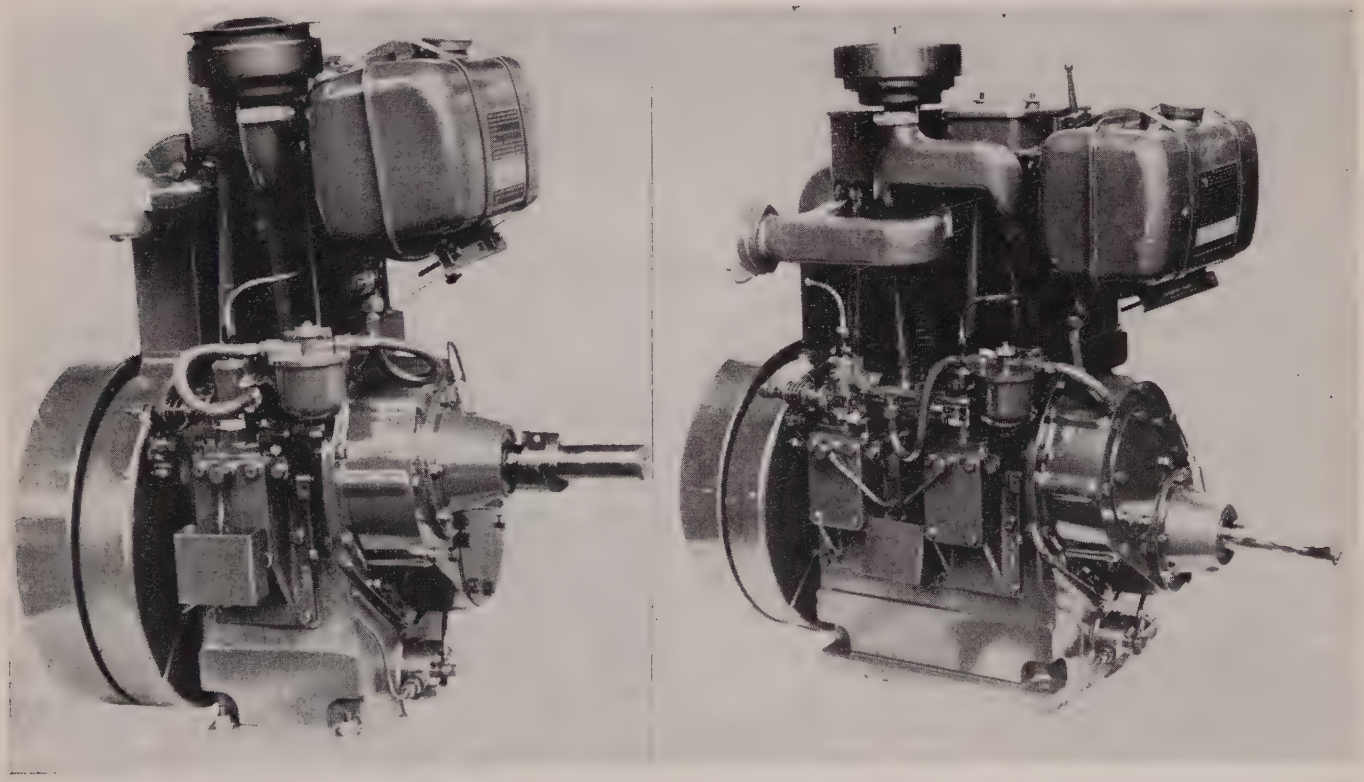


Fig. 28 – Variable speed fuel characteristics of Stihl 2-cycle diesel engine

Armstrong Siddeley Single- and Twin-Cylinder Aircooled Diesel Engines – Armstrong Siddeley Motors, Ltd. of Coventry are known over the world as builders of aircraft engines and motor cars. More recently, they have engaged in the indus-



Figs. 29 and 30 – Petter 1-cyl aircooled engine shown in general view at left (Fig. 29). At right, Fig. 30, is general view of Petter 2-cyl aircooled engine

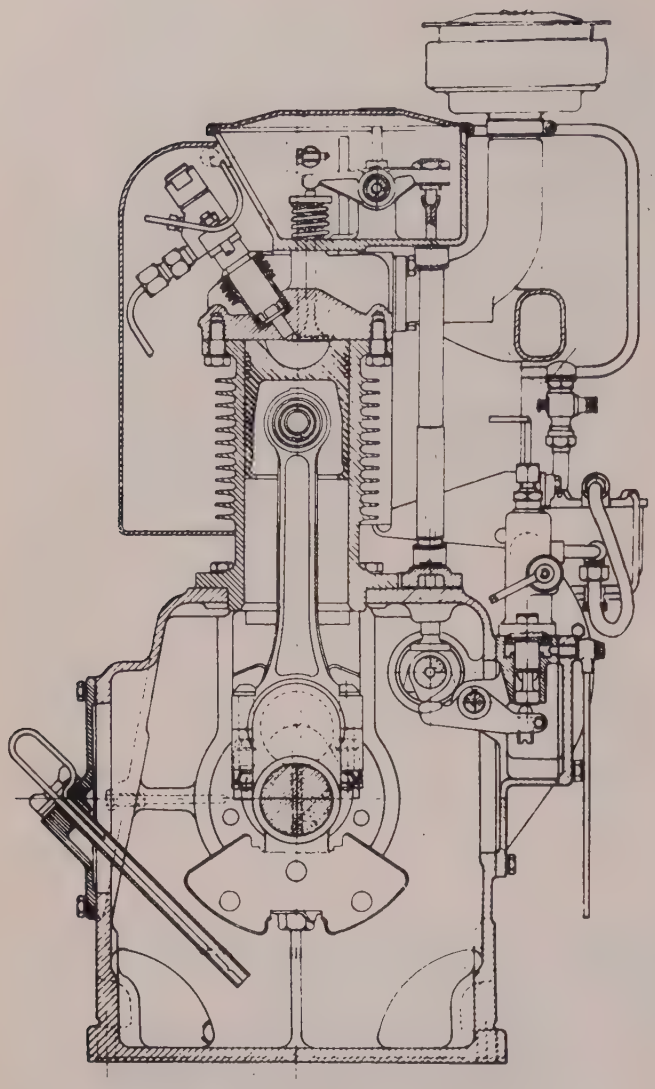


Fig. 31 – Cross-section through cylinder of Petter aircooled engine

Table 6 – Specifications for Armstrong Siddeley Engines

	Single	Twin
Number of cylinders	1	2
Bore, in.	4.25	4.25
Stroke, in.	4.25	4.25
Displacement, cu in.	60.29	120.58
Compression ratio	15/1	15/1
Rated speed, rpm	900-1500	1000-1500
Maximum torque, lb-ft	29.2	73.5
Speed at maximum torque, rpm	900	1000
Bmep at maximum torque, psi	73.2	92.0
Net engine weight with accessories, lb	540	700
Cylinder and head construction	Integral	Separate
Cylinder and head material	Gray Iron	Gray Iron
Piston material	Y Alloy	Silicon-Aluminum Alloy
Piston speed, at 1500 rpm	1060	1060
Horsepower maximum at 1500 rpm	8.0	20.0
Bmep maximum at 1500 rpm, psi	70.4	88.6
Starting means	Hand	Hand
Fuel tank capacity, U. S. gal	4.2	6.6
Oil sump capacity, U. S. gal	1.8	2.1

trial diesel-engine field. Considerable interest was aroused when the 1-cyl, 4-cycle aircooled diesel engine was introduced in 1946, because of their 100% adherence to the direct-cooling principle. In 1949, the twin engine was introduced.

The standard bore-and-stroke dimension for both engines is 4.25 in. (that is, a "square" engine), providing a displacement of 60.29 cu in. per cylinder. Cooling air is directed over the finned cylinder barrel, head, and finned sleeve for the injector, and on the twin engine, through tubes in the oil sump. Cooling air is provided by a simple fan integral with the flywheel. This construction, which involves no extraneous parts, holds the fan power requirement to a small value.

Specifications of these two models are as given in Table 6.

Both engines use steel-back precision type connecting rod bearings with indium lead-bronze in the rod half and babbitt in the cap half.

All fuel pumps and injectors are of C.A.V. make. Fuel nozzles are of the 3-hole type, injecting fuel at a pressure of 2975 psi directly in a semispherical combustion chamber in the piston crown.

The reliability of the engine is attested by the fact that the Dutch government has a number of the 1-cyl engines to power unattended light houses. Figs. 38 and 39 are general views of the 1-cyl engine and Fig. 40 is that of the twin engine, the performance of which is shown in Fig. 41. Figs. 42, 43, and 44 are sections and views respectively showing maximum operating temperatures during the 12-hr full-load run. All temperatures are given in deg C, and are based on an ambient air temperature of 35 C (95 F). The following notation applies: A, ambient air temperature, deg C; X, air entering the cooling fan at ambient temperature; Y, air leaving cylinder-barrel discharge outlet at temperatures above ambient value of: (a) 29 C at 18 bhp; (b) 35 C at 206 bhp; (c) 41 C at 226 bhp; E, exhaust; I, inlet.

Sales literature from Armstrong Siddeley features the use of their twin engine as a replacement for the V-type aircooled gasoline engines used on American-built farm equipment shipped overseas.

In order to better evaluate the data on operating temperatures of engine components, data is given below on temperature maxima published in recent papers by well-known authorities. In most internal-combustion engines of today, the piston assembly is the source of the severest problems affecting continuity of service. Satisfactory functioning of pistons, rings, and cylinder liners is, in a large measure, the criterion of successful behavior of any reciprocating internal-combustion engine. The temperatures shown in Figs. 42, 43, and 44 range from 200 to 250 C at various points on the piston, occurring at the highest 2-hr rating specified for the engine and with an ambient air temperature of 35 C (95 F).

In a paper presented in 1948 to the British

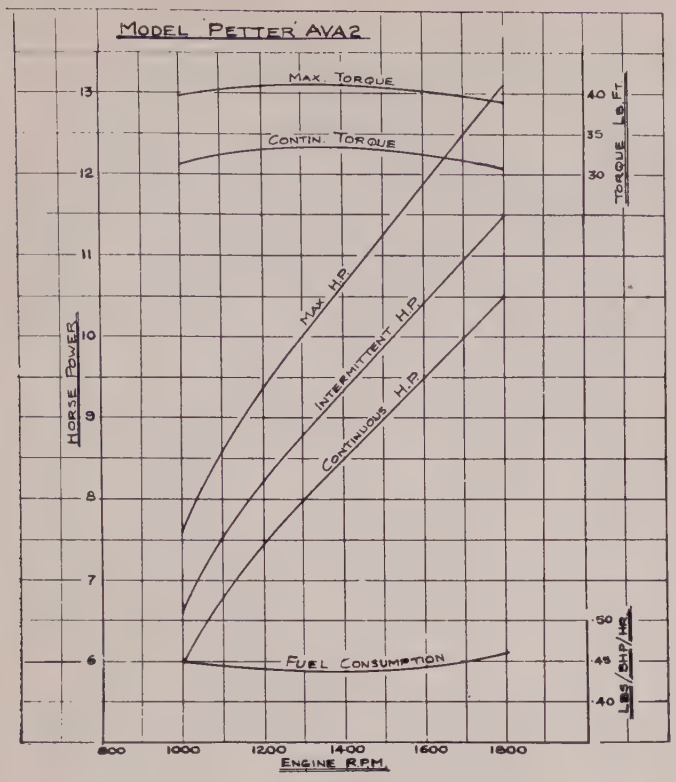


Fig. 32 – Variable speed performance characteristics of Petter 2-cyl aircooled diesel engine

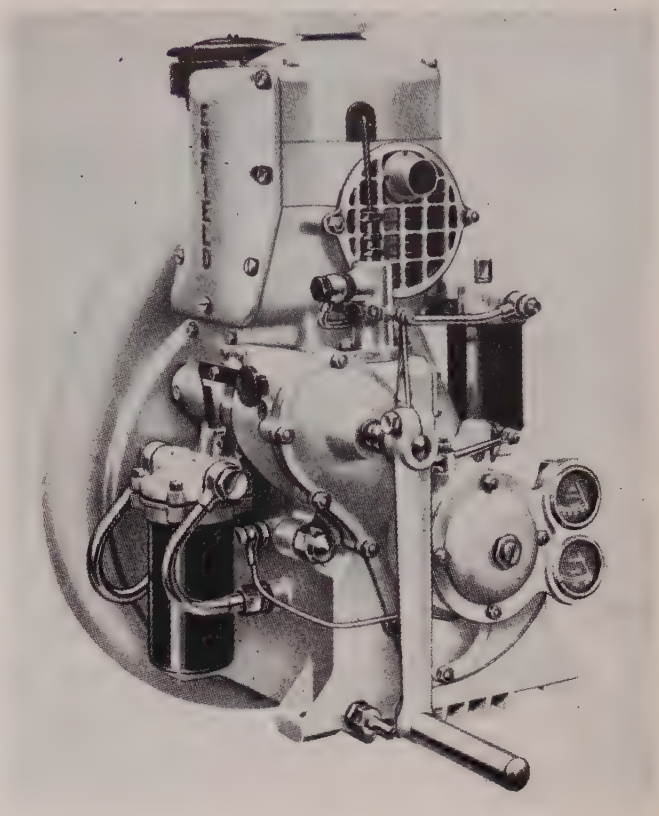


Fig. 33 – General view of Enfield 1-cyl aircooled diesel engine

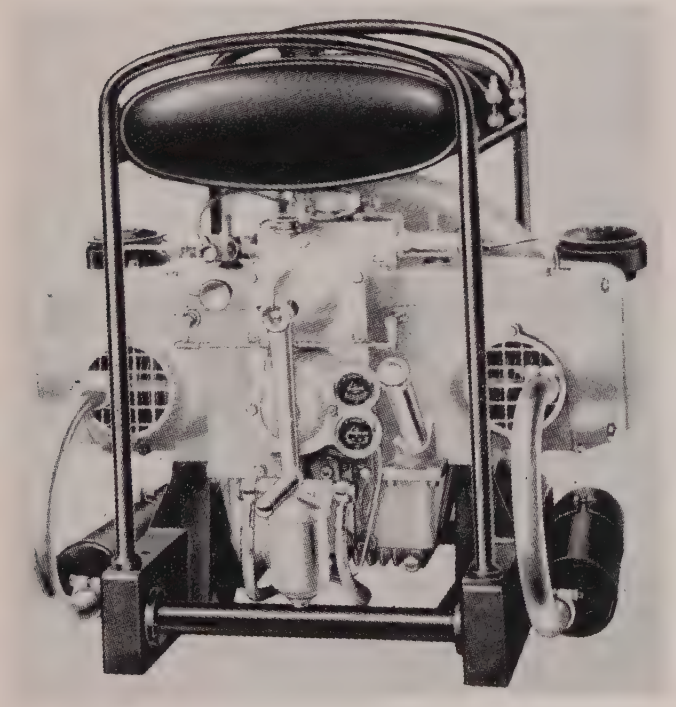


Fig. 34 – General view of Enfield 2-cyl aircooled diesel engine

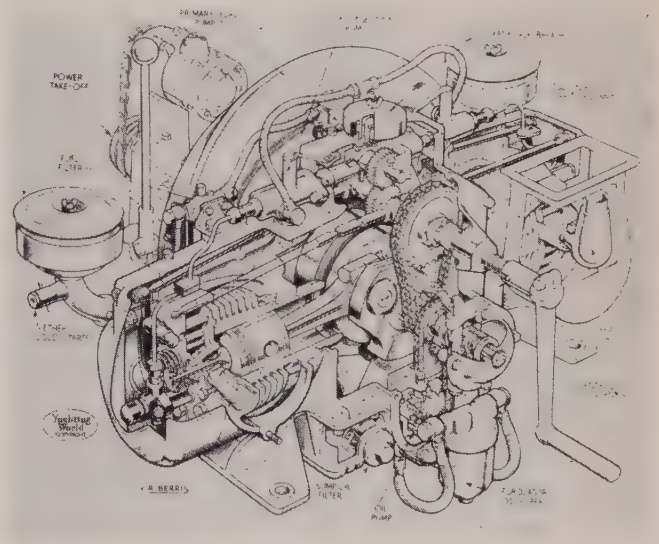


Fig. 35 – Sectional drawing of Enfield 2-cyl marine engine

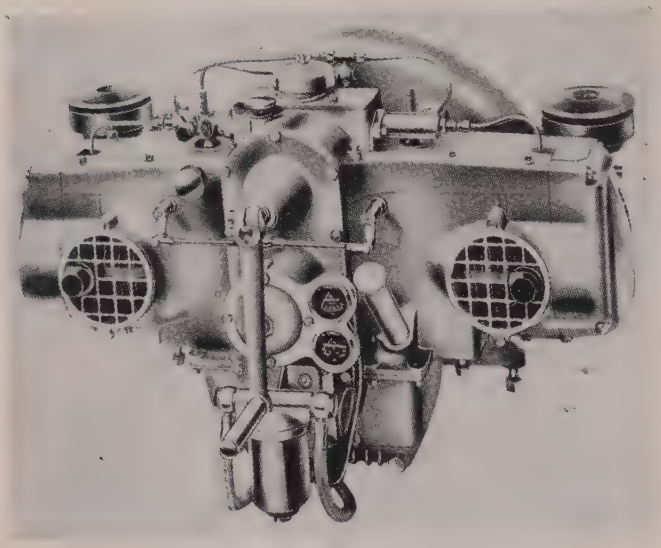


Fig. 36 – Enfield 2-cyl aircooled diesel engine showing general details

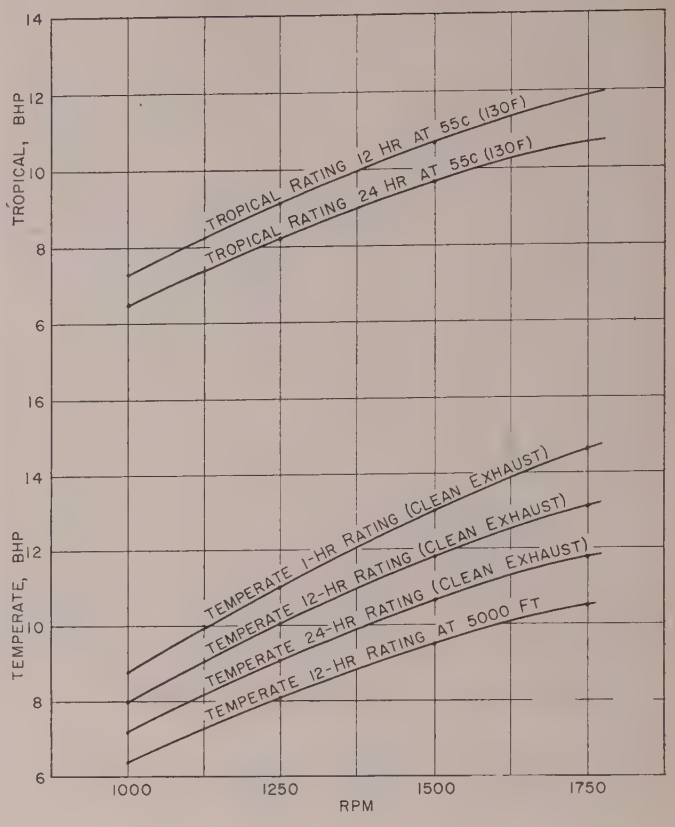


Fig. 37 – Diesel horsepower ratings of Enfield 2-cyl aircooled engine

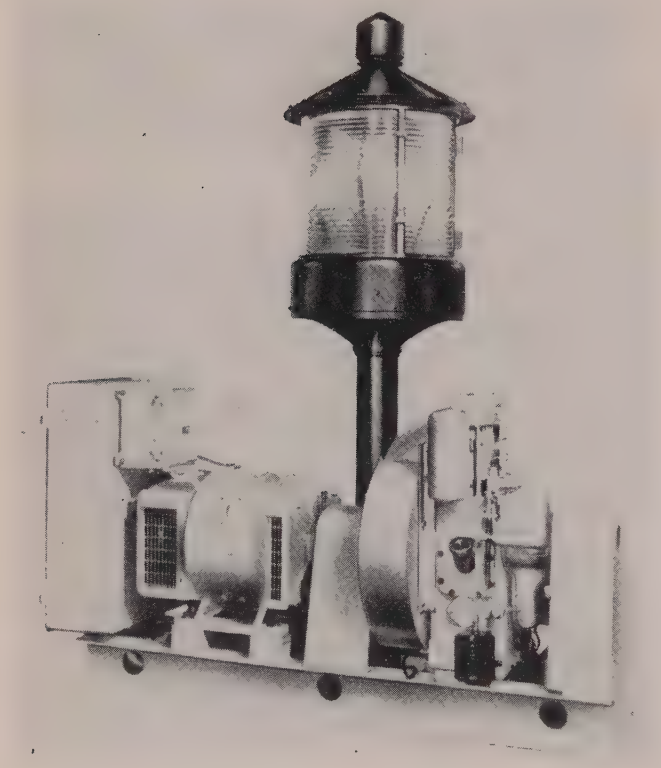


Fig. 38 – Armstrong Siddeley 1-cyl aircooled diesel engine showing general view with generator for unattended lighthouse

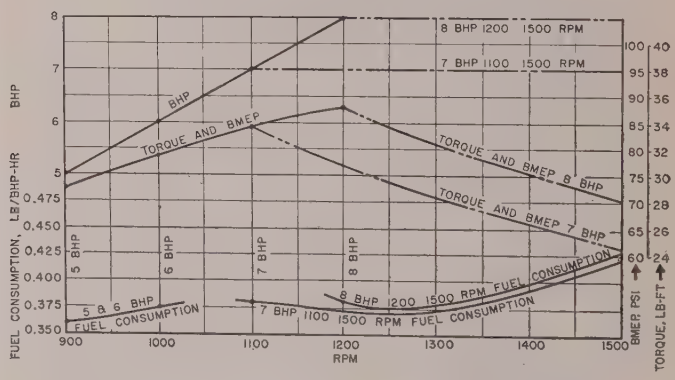


Fig. 39 – Performance curves of Armstrong Siddeley 1-cyl aircooled diesel engine

Association for the Advancement of Science, Sir Harry Ricardo suggested that the maximum safe temperatures for an aluminum-alloy piston in a diesel engine are: crown, 370-400 C; top ring groove, 200-200 C; gudgeon-pin bosses, 200-270 C. In a paper presented to the Institution of Mechanical Engineers in 1949, G. L. Hepworth said that in piston assemblies for transport oil engines, the ring groove temperature should not exceed 250 C, and that if the limit be set at 220-225 C, ring sticking can generally be eliminated.

In April, 1945, C. F. Russel and E. B. Graham (of Specialloid, Ltd.) presented to the Institution of Automobile Engineers a paper entitled "Light-Alloy Pistons." Therein, the authors gave an interesting chart of typical temperatures for the piston of a compression-ignition engine. These range from 150 C at the base of the skirt to some 400 C at the point at the top of the crown opposite the throat of the combustion chamber. Just below the piston-ring groove immediately above the gudgeon pin, the temperature is 175 C, ranging up to 225 C at the piston-crown periphery. These readings are concerned with water-cooled engines.

A study of the foregoing figures, in comparison with those given on the drawings of the Armstrong Siddeley engine, shows that the temperatures of the critical piston components are conservatively within the safety zone. The temperatures for the engine under consideration are no higher, if as high, as in many water-cooled compression-ignition engines of similar cylinder size and piston speed.

Swiss Aircooled Diesel Engines – The Swiss Locomotive and Machine Works of Winterthuer, Switzerland, have recently announced a series of 4-cycle aircooled diesel engines with 1 to 12 cylinders, and developing from 18 to 220 bhp. These engines are all built around a common cylinder and piston assembly having a bore of 4.43 in. (110 mm), a stroke of 5.51 in. (140 mm), and a displacement of 81.2 cu in. The compression ratio is 18/1. The rated speed for continuous heavy-duty service is 2000 rpm, which results in a piston speed of 1840 fpm. For automotive service, a maximum speed of 2400 rpm is permitted, which increased the piston speed to 2208 fpm.

In the 1-cyl engine, the cooling fan is integral with the flywheel, but in all others, the cooling air is provided by a belt-driven axial-flow blower. A cross-section of the 4-cyl engine is shown in Fig. 45, and the performance characteristics of the 12-cyl engine are shown in Fig. 46.

The engine has a number of unique features. The combustion chamber is of toroidal configuration placed somewhat eccentrically in the piston head. Each individual cylinder with heads is secured to the crankcase by means of four long through-bolts. The cylinder head is an outstanding example of foundry craftsmanship. This one casting contains the inlet and exhaust passages, the bore for the injector, an indicator opening, ribs sufficient to

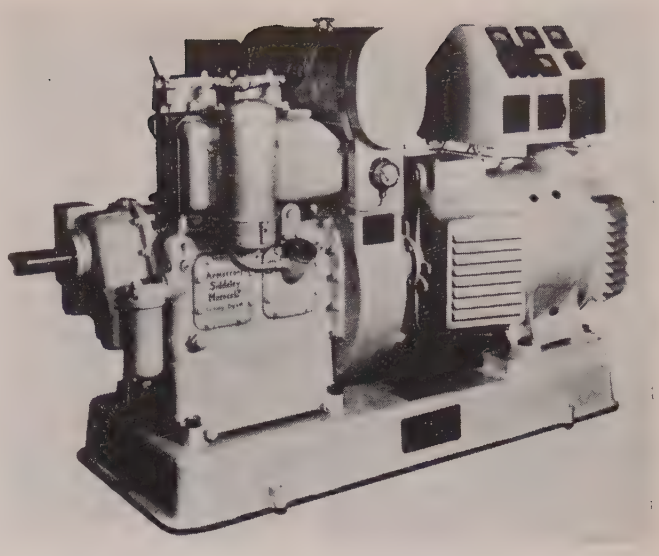


Fig. 40 – General view of Armstrong Siddeley 2-cyl aircooled diesel engine with generator

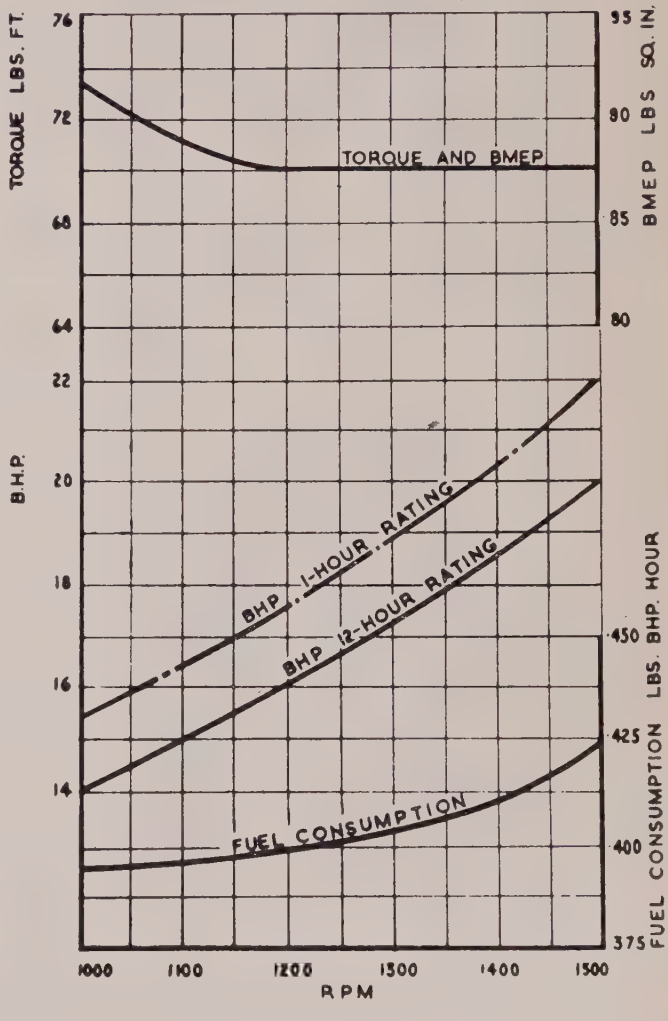


Fig. 41 – Performance curves of Armstrong Siddeley 2-cyl aircooled diesel engine

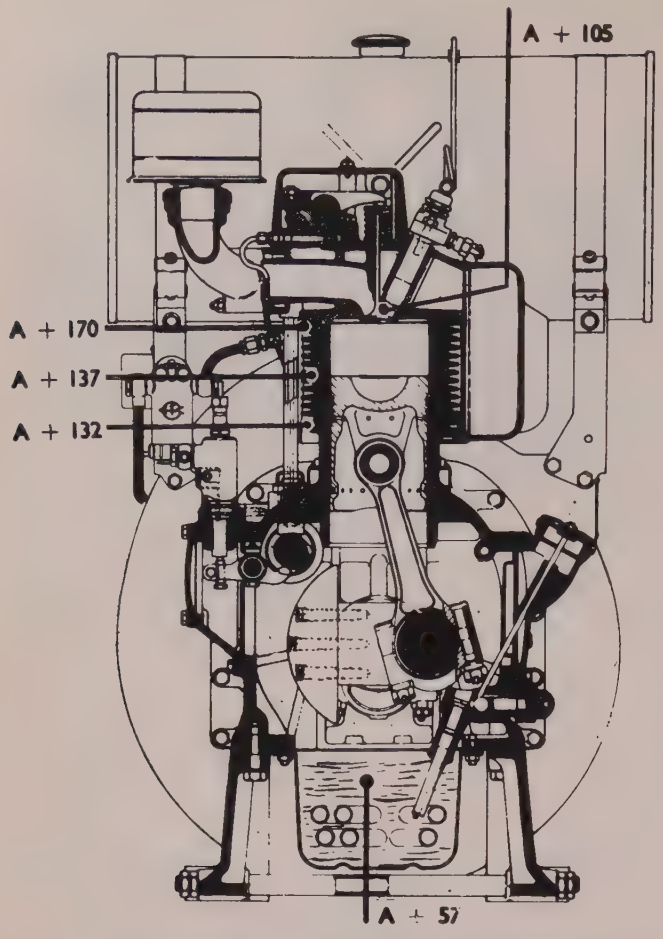


Fig. 42 – Cross-section of Armstrong Siddeley aircooled diesel engine showing cooling rib temperatures. All temperatures are given in deg C based on (A) ambient air temperature of 35 C

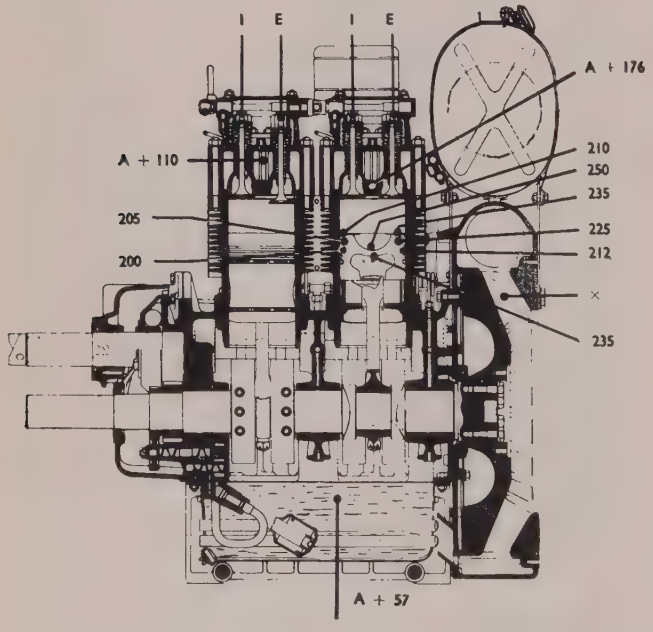


Fig. 43 – Longitudinal section of Armstrong Siddeley aircooled 2-cyl diesel showing working temperatures of piston and head. (I) inlet, (E) exhaust, (X) air entering cooling fan at ambient temperature. All temperatures given in deg C based on (A) ambient air at 35 C

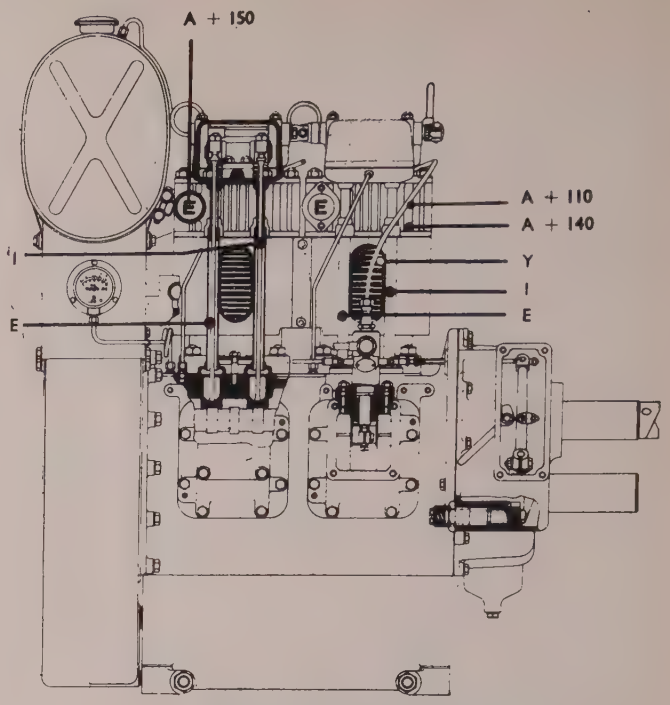


Fig. 44 – Armstrong Siddeley engine showing external working temperatures. (I) inlet (E) exhaust, (A) ambient air at 35 C; (Y) air leaving cylinder barrel discharge outlet at temperatures above ambient value of (a) 29 C at 18 bhp, (b) 35 C at 206 bhp, (c) 41 C at 226 bhp. Temperatures in figure given in deg C

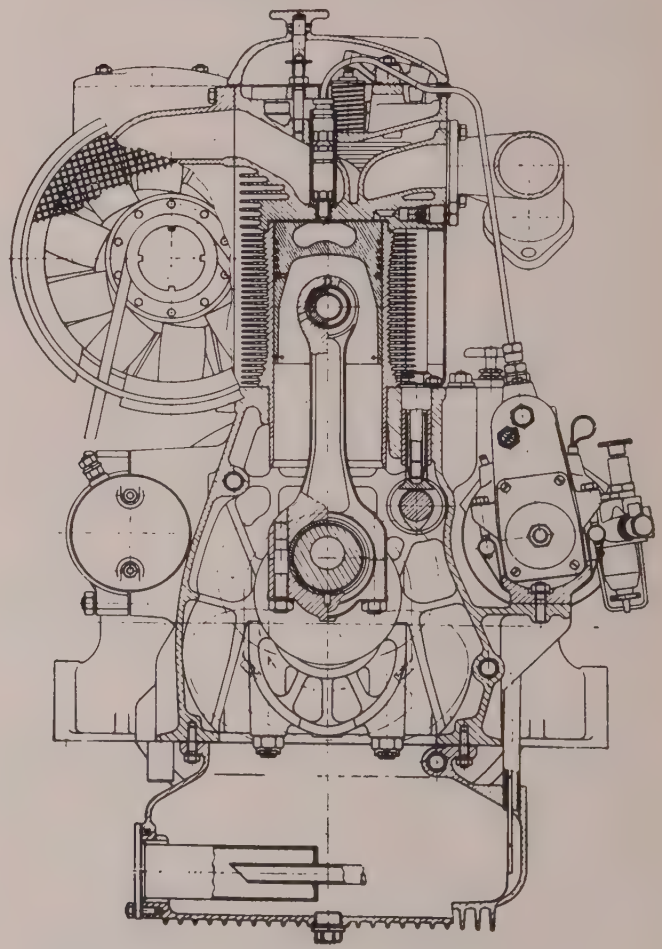


Fig. 45 – Swiss aircooled locomotive diesel cylinder section

carry off the necessary heat and air passages affording ample airflow, all within a space limited by the requirement of holding the overall engine length to a minimum. The point between cylinder and head is sealed with a narrow copper ring. The inlet and exhaust valves are both provided with bronze guides. The exhaust valve seat is replaceable, and when worn, the inlet valve seat may be replaced with a renewable ring.

The aircooling system is especially interesting. On the intake side of the engine, a cast-aluminum cover serves to enclose the cylinders and to distribute the cooling air. In front, this cover joins into the blower housing and at the rear, into the air-cleaner housing. The blower runs on roller bearings and is driven from the crankshaft by a pair of V-belts. Belt tension may be adjusted by means of the swinging electrical generator, which serves as a belt-lightener. The axial-flow blower rotor is a 1-piece light metal casting. The air is conducted through a diffuser, then through an oil cooler with metal fins, then deflected at 90 deg to the axis of the engine, past the cylinders and head and is finally discharged on the exhaust side. At low operating temperature conditions, suitable dampers can reduce the cooling airflow as required.

Air for combustion is taken from the cooling airstream through an integral air cleaner. The pressure head in the cooling-air duct overcomes the drop through the air cleaner, so that the slight supercharge is provided at the inlet ports, which insures good scavenging and high volumetric efficiency. The air cleaner consists of a cyclone-type precleaner, which has been found to remove all but about 10% of the dust in the incoming air, and an oil washdown element which operates with a high degree of efficiency to remove dirt left by the precleaner. Cooling of the injector is simply and effectively accomplished by surrounding the body of the injector by the air intake passage in the head, in such a way that heat from the injector is dissipated into the airstream.

The unusually low specific fuel consumption and high bmep are both characteristic of the toroidal combustion chamber, while the torque characteristics would excellently meet the requirements of tractor operation. The output compares favorably with that of water-cooled engines.

Unfortunately, the only illustrations of these very excellent-appearing engines are not suitable for making reproductions.

Conclusions

Aircooled diesel engines are being successfully built in Europe to meet a wide variety of automotive and industrial uses.

With one exception, they are all of the 4-cycle type.

Cooling problems appear to have been satisfactorily solved and in a number of different ways.

Three different types of combustion chambers are employed, using both pintle and multiple-orifice types of nozzles.

Specific output and specific fuel consumption are both comparable with the performance of similar combustion systems in water-cooled engines.

Cold-weather starting problems are stated to have been solved in a measure that compares favorably with liquid-cooled engines.

The wear rate is generally regarded as less than that of liquid-cooled engines, using fuels of equal grade and quality.

The efficiency of the cooling-air blowers appears to be uniformly high.

The overall length and height dimensions of aircooled engines can be maintained equal to or less than those of liquid-cooled engines of equal displacement when the space requirement of the radiator is considered.

Acknowledgment

The author wishes to thank the following for their invaluable help and cooperation, without which the preparation of this paper would have been impossible. Kloeckner-Humboldt-Deutz A.G.; Detroit Diesel Division, General Motors Corp.; *Automotive Industries;* Armstrong Siddeley Motors, Ltd.; Enfield Industrial Motors, Ltd.; Brush Aboe, Inc.; American Bosch Corp.; and John Deere Waterloo Tractor Works.

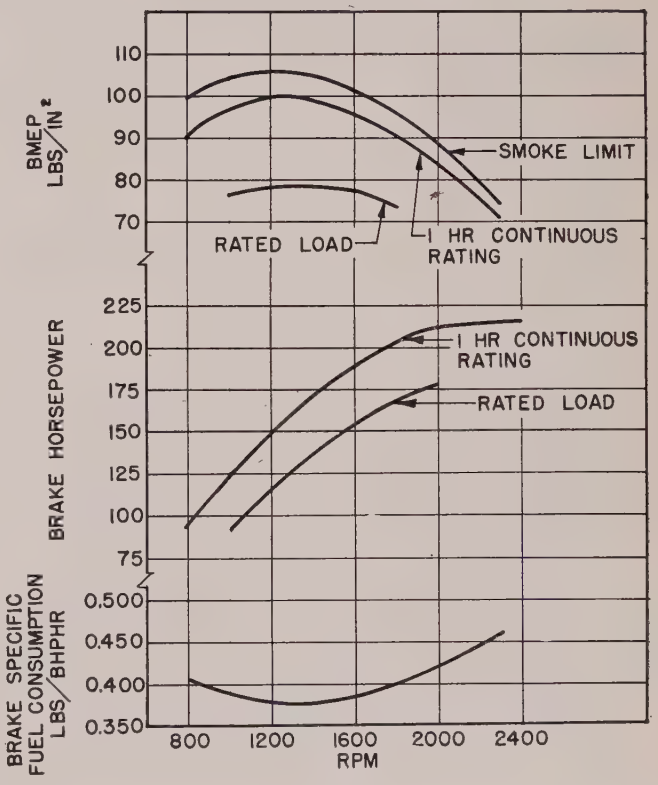


Fig. 46 – Performance characteristics of 12-cyl Swiss aircooled diesel engine shown in Fig. 45

Mode of Formation of

SINCE the discovery of the antiknock action of tetraethyl lead by Midgley in 1923 this compound has been widely used for the production of high octane fuels. Unfortunately the beneficial effects of lead are accompanied by several disadvantages, due to the formation of inorganic deposits in the combustion chamber, including the fouling of spark plugs, particularly in aero engines, and the corrosion and burning of exhaust valves. These deposits, in common with other deposits, give rise to an increase in octane requirements and power loss in automotive engines. These disadvantages have been to some extent minimized by the use of halide scavengers, more particularly ethylene dibromide and dichloride, but the problem is nevertheless still a serious one. In recent years efforts to overcome these deleterious effects of lead deposits have been intensified and work is proceeding to this end in several research establishments. Efforts are being made to modify the deposits by

DEVELOPMENT of an effective technique for the study of the formation of combustion-chamber deposits is described in this paper. The method is based on the extraction of samples from the end gas of an engine cylinder by means of an electromagnetic sampling valve, arranged to open over a narrow range of crank angles.

Lead compounds in the sample are deposited on a nickel foil placed in the body of the valve. Subsequently, they are identified by X-ray diffraction or, if necessary, by analytical chemical methods. The samples provide data on the mechanism of scavenging, particularly on the various steps occurring between the first breakdown of tetraethyl lead and the formation of the final products.

The technique also provides data on the exact mechanism of the antiknock action of tel.

The Author

J. C. STREET is a senior research chemist at the Thornton Research Centre, Shell Petrolum Co., Ltd. He studied chemistry at Cambridge University, England, taking Honours degree in 1944. He then joined Shell where he has since worked at the Thornton Research Centre in Cheshire, England, on general and fundamental combustion studies and specialized methods of gas analysis.

Lead Deposits in Gasoline Engines

J. C. Street, Thornton Research Centre, Shell Petroleum Co., Ltd.

Paper presented at SAE National Fuels and Lubricants Meeting, Tulsa, Nov. 7, 1952.

the use of additional scavenging materials, but such work has been handicapped by a lack of fundamental data on the exact mode of formation of the various types of deposit. In particular the various steps occurring between the first breakdown of tetraethyl lead and the formation of the final products have not been established. Information is also required as to the time in the cycle at which various scavengers exert their effect and the types of compound formed. This paper describes a technique which has been developed for a study of the above reactions and describes some of the preliminary results obtained.

The method involves extracting samples from the end gas of an engine cylinder by means of an electromagnetic sampling valve, arranged to open over a narrow range of crank angles, which has been developed at the Shoreham Works of Ricardo & Co., Ltd. The lead compounds in the sample are deposited on a nickel foil placed in the body of the valve and are subsequently identified by X-ray diffraction or where necessary by analytical chemical methods. Three unknowns, represented by the letters L, T, and M, still remain to be definitely identified.

In addition to obtaining data on the mechanism of scavenging, the technique described also provides a knowledge of the lead compounds present in the end-gas region at the time in the cycle at which lead is exerting its antiknock effect, and thereby enables further light to be thrown on the exact mechanism of the antiknock action of tel.

Apparatus

The apparatus used for obtaining the samples is shown diagrammatically in Fig. 1. The engine used was a variable compression Ricardo E.35 having a bore of 4½ in. and a stroke of 8 in. For this investigation the compression ratio was fixed at 5.0. In order to obtain the best cyclic regularity of combustion two spark plugs were used and the conventional contact breaker was replaced by an electronic triggering mechanism activated by a sliding pickup mounted above the rim of the flywheel.

The valve was fitted in the end-gas zone approximately midway between the spark plugs. It consisted of a stainless steel needle which, when the valve closed, fitted in a cast-iron nozzle. The movement of the needle was governed by a solenoid which was energized by the discharge from a bank of condensers, the timing of which was controlled by a contact breaker driven from the crankshaft, and adjustable to close at any required crank angle. To ensure that the valve opened only once during each cycle a second contact breaker, connected in series with the first and driven from the camshaft, was used. The time of opening of the valve, and also the duration, were indicated on an oscilloscope by means of a Sunbury magnetic pickup operating from an extension to the needle. The opening period of the valve used was 3 deg and this gave a sample gas flow rate of 200 ml per min. A more complete description of the valve has been published elsewhere by Downs, Walsh, and Wheeler.[1]

The time of flame arrival at the sampling valve was also indicated on the oscilloscope by the use of an ionization gap placed in the combustion-chamber wall adjacent to the sampling valve.

The sampled lead compounds were collected on a sleeve of removable nickel foil placed in the body

[1] See Philosophical Transactions of the Royal Society of London, Series A, Vol. 243, 1951, pp. 463-524: "A Study of the Reactions that Lead to Knock in the Spark-Ignition Engine," by D. Downs, A. D. Walsh, and R. W. Wheeler.

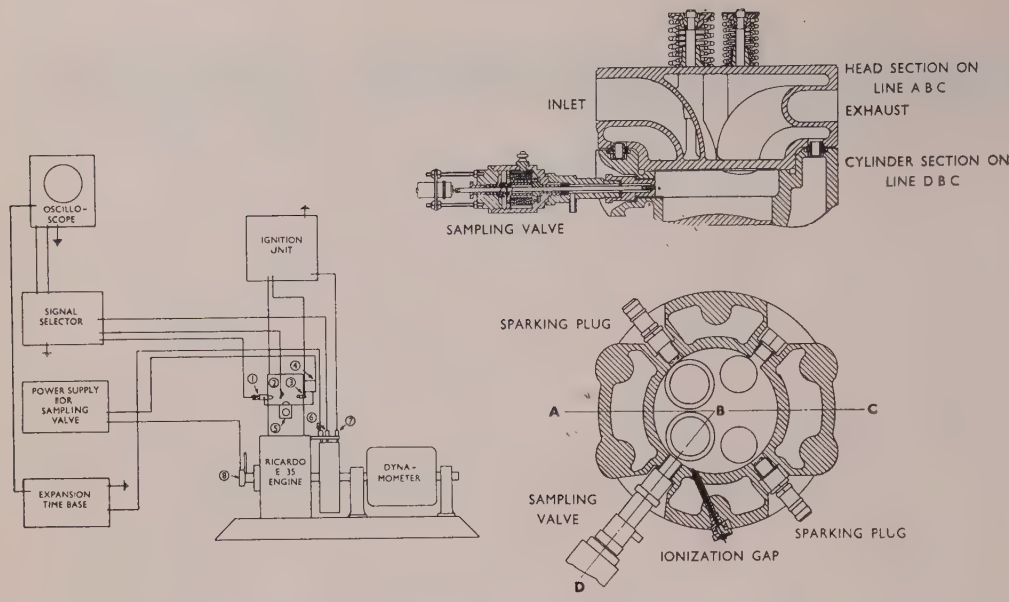
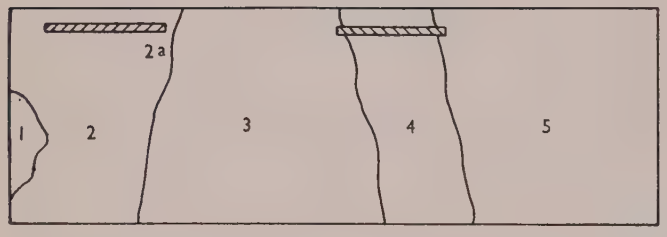


Fig. 1 – Sampling apparatus as fitted to Ricardo E.35 engine. Key to figure at left: (1) sampling valve; (2) ionization gap; (3) spark plug; (4) stroke selector for valve; (5) carburetor; (6) Sunbury pickup units for time-base sweep and degree marker; (7) timing pickup unit for ignition unit; (8) contact breaker controlling valve operation



ZONE	DESCRIPTION	X-RAY ANALYSIS
1	GREY	$Pb + PbBr_2$ (SOME)
2	BLUE BLACK	$Pb + PbBr_2$
2a		$Pb + \alpha NH_4 Br.2 PbBr_2$
3	DULL BRICK RED	$Pb + 2NH_4 Br.Pb Br_2$
4	DEPOSIT FADE OUT	$+ \alpha NH_4Br.2Pb Br_2$
5	DEPOSIT END	$NH_4 Br + 2NH_4 Br.Pb Br_2$
		NH_4Br

CHEMICAL ANALYSIS

		Pb	Br
▨	=	71%	16%
▨	=	26%	24%

Fig. 2 – Typical variation of deposit along length of sampling foil, obtained at 30 deg late on 100/130 grade aviation gasoline

TEST CONDITIONS

SPEED	: 1500 r.p.m.
JACKET TEMPERATURE	: 60°C
COMPRESSION RATIO	: 5·0
INLET AIR HEAT	: 1350 watts at 20°C
IGNITION ADVANCE	: 30°
SAMPLING RATE	: 200 ml/min

FUEL

100/130 GRADE AVIATION GASOLINE
Sp.Gr. 0·718 at 15°C

FUEL/AIR RATIO : 0·075

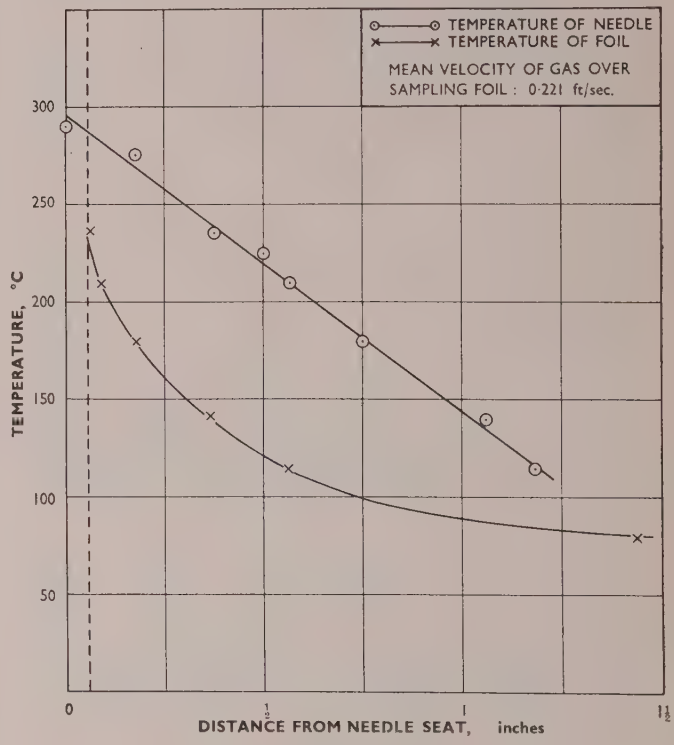


Fig. 3 – Temperature of sampling-valve needle as measured by Thermindex paints

Table 1 – Test Conditions

Engine Speed, rpm	1500
Engine Coolant Temperature, C	60
Heat to Intake Air, at 20 C, watts	1350
Compression Ratio	5.0/1
Fuel/Air Ratio, Rich	0.075, ignition advance 29 to 30 deg early (except Test 3)
Fuel/Air Ratio, Weak	0.060, ignition advance 40 deg early

Table 2 – Fuels Tested

Test 1	(a and b) 100/130 grade aviation gasoline (80-octane clear gasoline + 4.6 ml tel per U. S. gal + 1 Theory (1-T) ethylene dibromide)
Test 2	80-octane gasoline + 4.6 ml tel per U. S. gal (no scavenger)
Test 3	Isopropyl alcohol + 3.2 ml tel per U. S. gal + 0.5-T ethylene dibromide
Test 4	80-octane gasoline + 4.6 ml tel per U. S. gal + 1-T ethylene dibromide + 0.2-T tricresyl phosphate tcp
Test 5	80-octane gasoline + 4.6 ml tel per U. S. gal + 1-T ethylene dibromide + 0.5-T tcp
Test 6	80-octane gasoline + 4.6 ml tel per U. S. gal + 1-T ethylene dibromide + 1-T tcp
Test 7	80-octane gasoline + 4.6 ml tel per U. S. gal + 1-T tcp

of the sampling valve. X-ray diffraction patterns of the samples were obtained photographically, using a Debye-Scherrer camera of radius 57.3 mm and Cu $K\alpha$ radiation.

Test Procedure

In order to obtain a representative analysis by X-ray methods a minimum deposit of approximately 5 mg was required. This necessitated a test run of 3 to 25 hr depending on the fuel used and the crank angle at which sampling took place. On removal the foil was found to contain deposits in well-defined zones of varying colors, specimens being taken from each of these zones for analysis.

A few of the diffraction patterns so obtained were not listed in either the ASTM index of powder diffraction data or a library of diffraction patterns of lead compounds compiled at Thornton Research Centre. In these cases a longer engine run was carried out to obtain about 50 mg of deposit which was then analyzed chemically. On the basis of this analysis possible compounds were synthesized and their X-ray diffraction patterns obtained. This has enabled the majority of these deposits to be identified.

Some tests were carried out in which an electrostatic precipitator placed downstream of the sampling valve was used to collect any compound escaping deposition on the nickel foil. These tests showed that approximately 80% of the lead was deposited on the nickel foil and also that the deposit collected on the precipitator did not include any compound not also found on the foil. In view of this, the precipitator was dispensed with for the majority of the work.

Where possible the gas flow through the valve was maintained at a rate of 200 ml per min but at sampling angles at which the pressure in the cylinder was low the sampling rate fell to, at the worst, 15 ml per min even for maximum valve opening.

The conditions used throughout the work are listed in Table 1.

The fuels tested are listed in Table 2. Test 1 was carried out at both (a) rich and (b) weak mixtures. The other fuels were tested only at the rich mixture.

In Test 3 (Table 2), with isopropyl alcohol, because of the greater rate of fuel consumption, the lead concentration was reduced to give the same rate of lead consumption per unit time.

At the completion of a test on any one fuel, as listed in Table 2, the engine was dismantled and the deposit scraped from the combustion chamber, and subsequently analyzed both chemically and by X-ray diffraction.

Results

Fig. 2 depicts pictorially a typical pattern obtained on the sampling foil. In this case the sample was obtained at 30 deg late when running on 100/130-grade aviation gasoline. Generally speaking the heavier particles separated out earlier on the sampling foil, suggesting that gravity was one factor controlling the variation in type of deposit along the length of the foil. However, there was no evidence of the deposit being thicker on the lower surface of the foil than on the upper, as would be expected if gravity were playing a part. It appears probable that thermal factors are largely responsible for the actual mechanism of deposition. Thus Fig. 3 shows the approximate temperature distribution along the foil and also on the valve needle when sampling at tdc. These temperatures were obtained by the use of Thermindex temperature paints. It can be seen that the temperature halfway along the foil had dropped to a value less than 80 C, below which the temperature could not be measured by this means. The effect of sampling at earlier or later angles was to reduce these temperatures, the greatest effect being on the hottest part of the foil. Thus at 270 deg late the temperature of the hot end of the foil was 180 C compared with 235 C when sampling at 2 deg late. The maximum temperature difference between the needle and the foil occurred about halfway along the foil and was approximately 100 C. Bredl and Grieve[2] have described a thermal precipitator using a temperature difference of 150 C to 180 C for optimum results, so it appears probable that some such thermal mechanism is playing a part here.

The results obtained with the various fuels are tabulated in Tables 3 to 10 and depicted graphi-

[2] See *Journal of Scientific Instruments*, Vol. 28, January, 1951, pp. 21-23: "A Thermal Precipitator for the Gravimetric Estimation of Solid Particles in Flue Gases," by J. Bredl and T. W. Grieve.

Table 3 – X-Ray Analysis of Foil Deposits – Test 1a

Fuel, 100/130 aviation gasoline
Engine Speed, rpm 1500
Compression Ratio 5/1
Flame Arrival, deg late 10 (approximate)
Ignition Advance, deg early 30
Jacket Temperature, C 60
Fuel/Air Ratio 0.075
Mixture Strength, % rich 12
Inlet Air Heat, at 20 C, watts 1350

Sampling Angle, deg early = E deg late = L	Sample, Number	Gas Flow Through Sampling Valve, ml per min	Test Duration, hr	Deposit Analyses
130 E	70	24	21.5	$PbBr_2$ + Fe
90 E	44	52	25.25	$PbBr_2$ + NiO
	63	47	22.0	SiO_2 + $PbBr_2$ + Pb → $PbBr_2$
	83	50	20.0	$PbBr_2$
60 E	43	91	24.0	$CaCO_3$ → $CaCO_3$ + $PbSO_4$ → $CaCO_3$ + SiO_2 + $PbSO_4$
	62	89	11.3	SiO_2 + $PbBr_2$ → $PbBr_2$
30 E	42	200	11.25	$PbBr_2$ → $PbBr_2$ + some $PbSO_4$
25 E	86	200	8.0	PbO · $PbBr_2$ R-phase + $PbBr_2$
20 E	85	180	7.3	PbO · $PbBr_2$ R-phase
15 E	84	200	4.1	PbO · $PbBr_2$ R-phase + some $PbBr_2$
10 E	9	200	6.0	β3PbO · $PbBr_2$
	112	200	6.35	β3PbO · $PbBr_2$ + trace 2PbO · $PbBr_2$
7.5 E	18	200	6.5	β3PbO · $PbBr_2$
5 E	13	200	7.0	β3PbO · $PbBr_2$
tdc	82	200	5.7	β3PbO · $PbBr_2$ + Pb + PbO · $PbBr_2$
	110			R-phase
2 L	10	200	3.0	β3PbO · $PbBr_2$
10 L	19	200	3.25	3PbO · $PbBr_2$
	78			
15 L	15	200	6.5	$PbBr_2$ → $PbBr_2$ + βNH_4Br · 2$PbBr_2$ + αNH_4Br · 2$PbBr_2$
	114	200	6.20	Pb(OH)Br + $PbBr_2$
30 L	6	200	3.0	Pb → Pb + $PbBr_2$ → αNH_4Br · 2$PbBr_2$ → 2NH_4Br · $PbBr_2$
	16, 17	200	24.3, 36.3	Pb + $PbBr_2$ → αNH_4Br · 2$PbBr_2$ → Unknown *L*
90 L	11	100	4.0	Pb → $PbBr_2$ → αNH_4Br · 2$PbBr_2$
135 L	7	60	7.5	$PbBr_2$ + little βNH_4Br · 2$PbBr_2$ → $PbBr_2$ → $PbBr_2$ + αNH_4Br · 2$PbBr_2$
270 L	8	17	15.0	$PbBr_2$

Note: → indicates increasing distance along foil from sampling orifice.

Table 4 – X-Ray Analysis of Foil Deposits – Test 1b

Fuel, 100/130 aviation gasoline
Engine Speed, rpm 1500
Compression Ratio 5/1
Flame Arrival, deg late 10
Ignition Advance, deg early 42
Jacket Temperature, C 60
Fuel/Air Ratio 0.06
Mixture Strength, % weak 10.5
Inlet Air Heat, at 20 C, watts 1350

Sampling Angle, deg early = E deg late = L	Sample, Number	Gas Flow Through Sampling Valve, ml per min	Test Duration, hr	Deposit Analyses
10 E	20	150	13.0	$PbCl_2$ · $PbBr_2$ mixed crystals + β3PbO · $PbCl_2$ · 3PbO · $PbBr_2$ mixed crystals → β3PbO · $PbCl_2$ · 3PbO · $PbBr_2$ mixed crystals
tdc	21	200	7.0	β3PbO · $PbBr_2$
5 L	22	200	6.0	β3PbO · $PbBr_2$
12 L	23	200	6.0	β3PbO · $PbBr_2$
30 L	24	201	6.0	$PbSO_4$ → $PbBr_2$ → αNH_4Br · 2$PbBr_2$
90 L	25	90	24.25	$PbBr_2$
135 L	26	50	22.25	$PbBr_2$
270 L	27	50	.....	$PbBr_2$

Note: → indicates increasing distance along foil from sampling orifice.

Table 5 – X-Ray Analysis of Foil Deposits – Test 2

Fuel, 80-octane base + 4.6 ml tel per U. S. gal
Engine Speed, rpm 1500
Compression Ratio 5/1
Flame Arrival, deg late 10 (approximate)
Ignition Advance, deg early 29
Jacket Temperature, C 60
Fuel/Air Ratio 0.075
Mixture Strength, % rich 12
Inlet Air Heat, at 20 C, watts 1350

Sampling Angle, deg early = E deg late = L	Sample, Number	Gas Flow Through Sampling Valve, ml per min	Test Duration, hr	Deposit Analyses
10 E	28	200	8.0	4PbO · $PbSO_4$
2 L	29	200	8.0	4PbO · $PbSO_4$
15 L	30	200	6.5	4PbO · $PbSO_4$
30 L	31	200	8.0	4PbO · $PbSO_4$
90 L	32	100	8.0	Possibly PbO[a] → traces 4PbO · $PbSO_4$
135 L	33	80	15.75	Possibly PbO[a] → traces 4PbO · $PbSO_4$
270 L	34	...	...	Ni + NiO

[a] The pattern resembles that of orthorhombic PbO, but with the absence of reflections from planes perpendicular to the C axis.
Note: → indicates increasing distance along foil from sampling orifice.

Table 6 – X-Ray Analysis of Foil Deposits – Test 3

Fuel, Isopropyl alcohol + 3.2 ml tel per U. S. gal + 0.5-T ethylene dibromide
Engine Speed, rpm 1500
Compression Ratio 5/1
Flame Arrival, deg late 9
Ignition Advance, deg early 31
Jacket Temperature, C 60
Fuel/Air Ratio 0.108
Mixture Strength, % rich 12
Inlet Air Test, at 20 C, watts 1350

Sampling Angle, deg early = E deg late = L	Sample Number	Gas Flow Through Sampling Valve, ml per min	Test Duration, hr	Deposit Analyses
30 E	128	159	5.55	Amorphous. Contains a little unknown *T*
20 E	129	194	4.50	β3PbO · $PbBr_2$
10 E	130	200	5.25	β3PbO · $PbBr_2$
8 L	131	200	5.50	β3PbO · $PbBr_2$
15 L	132	200	5.35	β3PbO · $PbBr_2$
30 L	133	200	4.55	R form of PbO · $PbBr_2$ + Pb → Pb + $PbBr_2$ → $PbBr_2$ + Pb
130 L	134	98	14.25	β3PbO · $PbBr_2$ + some unidentified compound → 3PbO · $PbBr_2$
270 L	135	21	16.25	β3PbO · $PbBr_2$

Note: → indicates increasing distance along foil from sampling orifice.

Table 7 – X-Ray Analysis of Foil Deposits – Test 4

Fuel, 80-octane base + 4.6 ml tel per U. S. gal + 1-T ethylene dibromide + 0.2-T tricresyl phosphate
Engine Speed, rpm 1500
Compression Ratio 5/1
Flame Arrival, deg late 10 (approximate)
Ignition Advance, deg early 30
Jacket Temperature, C 60
Fuel/Air Ratio 0.075
Mixture Strength, % rich 12
Inlet Air Heat, at 20 C, watts 1350

Sampling Angle, deg early = E deg late = L	Sample Number	Gas Flow Through Sampling Valve, ml per min	Test Duration, hr	Deposit Analyses
130 E	99	30	13.9	$PbBr_2$ + some $3Pb_3(PO_4)_2$ · $PbBr_2$ → Pb(OH)Br
30 E	98	200	7.0	$PbBr_2$ + some $3Pb_3(PO_4)_2$ · $PbBr_2$ → $3Pb_3(PO_4)_2$ · $PbBr_2$
20 E	97	200	7.0	$PbBr_2$ + trace $3Pb_3(PO_4)_2$ · $PbBr_2$
10 E	47	200	7.25	$PbCl_2$ · $PbBr_2$ mixed crystals → $3Pb_3(PO_4)_2$ · $PbBr_2$ · $3Pb_3(PO_4)_2$ · $PbCl_2$ mixed crystals
5 E	48	200	7.0	$3Pb_3(PO_4)_2$ · $PbBr_2$
2 L	49	200	7.0	$3Pb_3(PO_4)_2$ · $PbBr_2$
8 L	50	200	7.5	$3Pb_3(PO_4)_2$ · $PbBr_2$
11 L	100	200	4.40	β3PbO · $PbBr_2$ + trace of 2$PbBr_2$ · $Pb_2P_2O_7$ β3PbO · $PbBr_2$ → β3PbO · $PbBr_2$ + $PbBr_2$ + trace of 2$PbBr_2$ · $Pb_2P_2O_7$
15 L	51	200	7.5	$PbBr_2$ + 4PbO · $PbSO_4$ → $PbBr_2$ + αNH_4Br · 2$PbBr_2$
30 L	52	200	4.0	Pb + some $PbBr_2$ → αNH_4Br · 2$PbBr_2$ → βNH_4Br · 2$PbBr_2$
	127	200	37.10	$PbBr_2$ + Pb + some 2$PbBr_2$ · $Pb_2P_2O_7$ → $PbBr_2$ + Pb + 2$PbBr_2$ · $Pb_2P_2O_7$ → $PbBr_2$ + αNH_4Br · $PbBr_2$ + βNH_4Br · 2$PbBr_2$ + 2NH_4Br · $PbBr_2$ + unknown L → αNH_4Br · 2$PbBr_2$
90 L	53	112	4.75	$PbBr_2$ → $PbBr_2$ + αNH_4Br · 2$PbBr_2$ → αNH_4Br · 2$PbBr_2$
138 L	54	75	13.5	$PbBr_2$ → αNH_4Br · 2$PbBr_2$ → αNH_4Br · 2$PbBr_2$
270 L	55	18	15.5	$PbBr_2$

Note: → indicates increasing distance along foil from sampling orifice.

Table 8 – X-Ray Analysis of Foil Deposits – Test 5

Fuel, 80-octane base + 4.6 ml tel per U. S. gal + 1-T ethylene dibromide + 0.5-T tricresyl phosphate
Engine Speed, rpm 1500
Compression Ratio 5/1
Flame Arrival, deg late 9
Ignition Advance, deg early 30
Jacket Temperature, C 60
Fuel/Air Ratio 0.075
Mixture Strength, % rich 12
Inlet Air Heat, at 20 C, watts 1350

Sampling Angle, deg early = E deg late = L	Sample Number	Gas Flow Through Sampling Valve, ml per min	Test Duration, hr	Deposit Analyses
30 E	87	150	7.0	$PbCl_2$/$PbBr_2$ mixed crystals
20 E	88	135	7.0	$3Pb_3(PO_4)_2$ · $PbBr_2$ + $PbBr_2$
10 E	89	200	6.0	$3Pb_3(PO_4)_2$ · $PbBr_2$
5 E	90	200	6.0	$3Pb_3(PO_4)_2$ · $PbBr_2$
2 L	91	170	4.5	$3Pb_3(PO_4)_2$ · $PbBr_2$
8 L	92	200	4.0	$3Pb_3(PO_4)_2$ · $PbBr_2$
15 L	93	200	2.35	2$PbBr_2$ · $Pb_2P_2O_7$ · → 4$PbBr_2$ · $Pb_2P_2O_7$ + trace $3Pb_3(PO_4)_2$ · $PbBr_2$ → αNH_4Br · 2$PbBr_2$
30 L	94	200	6.0	2$PbBr_2$ · $Pb_2P_2O_7$ · → αNH_4Br · 2$PbBr_2$ → αNH_4Br · 2$PbBr_2$ + trace 2NH_4Br · $PbBr_2$
135 L	95	63	13.5	2$PbBr_2$ · $Pb_2P_2O_7$ · → αNH_4Br · 2$PbBr_2$ + 2NH_4Br · $PbBr_2$ → βNH_4Br · 2$PbBr_2$
270 L	96	16.5	19.9	$3Pb_3(PO_4)_2$ · $PbBr_2$ + $PbBr_2$ → $3Pb_3(PO_4)_2$ · $PbBr_2$ + αNH_4Br · $PbBr_2$

Note: → indicates increasing distance along foil from sampling orifice.

cally in Figs. 4 and 5. The results are purely qualitative and the presence of any substance on a diagram gives no indication of its relative proportion in the total sample.

One further reservation must be kept in mind. The results quoted, especially the absence of any compound, are limited by the limits of detection of the various compounds by the X-ray technique used. In the case of points of particular interest, that is, at or near crank angles at which compounds abruptly appeared or disappeared, the above limits were reduced to a minimum by processing the films in a quinine thiosulfate intensifier solution. By the use of standard mixtures limits of detection were established as listed in Table 11.

In drawing conclusions from the following observations these limits must be borne in mind.

Tests with Ethylene Dibromide ($EtBr_2$) and Tel – In Test 1a the fuel used was normal aviation fuel, consisting of a base fuel of 80-octane gasoline containing 4.6 ml tel per U. S. gal and 1-T of $EtBr_2$. The fuel/air ratio was 0.075.

From the earliest point sampled (130 deg early) to about 30 deg early the main product was lead bromide, together with a little lead sulfate in the region 60 deg early to 30 deg early. Starting at or about 30 deg early lead oxybromides made their first appearance in the form of $2PbO \cdot PbBr_2$ and were accompanied by a little lead bromide. In the region 10 deg early to 10 deg late the sole products, with the exception of one point which showed a trace of metallic lead, were lead oxybromides mainly of a composition referred to as $\beta 3PbO \cdot PbBr_2$. The actual diffraction pattern obtained is given by a hydrate of composition $3PbO \cdot PbBr_2 \cdot H_2O$ to $3PbO \cdot PbBr_2O \cdot 5H_2O$ at up to 180 C. In actual fact, although referred to as $\beta 3PbO \cdot PbBr_2$, it is believed that the phase can exist over a range of molecular ratios of PbO to $PbBr_2$ in the neighborhood of 3/1. At or about the time of flame arrival at the sampling valve (10 deg late) the oxybromides abruptly and completely disappeared from the sampled deposit. The deposits from that point to about 150 deg late consisted of lead bromide, metallic lead, and various ammonium lead bromides. These last had the composition $2NH_4Br \cdot PbBr_2$ or $NH_4Br \cdot 2PbBr_2$, the latter occurring in two crystalline forms, a brown orthohombic form which is referred to a $\alpha NH_4Br \cdot 2PbBr_2$, and a grey tetragonal form referred to as $\beta NH_4Br \cdot 2PbBr_2$. The diffraction pattern of the latter form agrees with the X-ray data given by Powell and Tasker[4] for $NH_4Br \cdot 2PbBr_2$. In the region 150 deg late to the latest point sampled (270 deg late) the sample consisted entirely of lead bromide.

Test 1b at a weaker mixture strength (F/A

[3] See Atti della Reale Accademia dei Lincei, Vol. 21, Serie Quinta, Rendiconti, Semestre II, 1912 (Rome), pp. 768-774: "Sulla Tendenza a Combinarsi tra Alogenure e Fosfati dello Stesso Metallo; Fluoruro, Cloruro, e Fosfato di Piombo," by Mario Amadori.

[4] See *Journal of the Chemical Society* (London), Part I, 1937, pp. 119-123: "The Valency Angle of Bivalent Lead: The Crystal Structure of Ammonium, Rubidium, and Potassium Pentabromodiplumbites," by H. M. Powell and H. S. Tasker.

Table 9 – X-Ray Analysis of Foil Deposits – Test 6

Fuel, 80-octane base + 4.6 ml tel per U. S. gal + 1-T tricresyl phosphate + 1-T ethylene dibromide

Engine Speed, rpm	1500	Jacket Temperature, C	60
Compression Ratio	5/1	Fuel/Air Ratio	0.075
Flame Arrival, deg late	10 (approximate)	Mixture Strength, % rich	12
Ignition Advance, deg early	30	Inlet Air Heat, at 20 C, watts	1350

Sampling Angle, deg early = E deg late = L	Sample Number	Gas Flow Through Sampling Valve, ml per min	Test Duration, hr	Deposit Analyses
130 E	105	23	31.17	$3Pb_3(PO_4)_2 \cdot PbBr_2 + PbBr_2$
30 E	104	171	4.30	$3Pb_3(PO_4)_2 \cdot PbBr_2 + PbBr_2$
20 E	103	200	4.30	$3Pb_3(PO_4)_2 \cdot PbBr_2 + PbBr_2$
10 E	35	200	7.25	$3Pb_3(PO_4)_2 \cdot PbBr_2$
5 E	45	200	6.83	$3Pb_3(PO_4)_2 \cdot PbBr_2$
2 E	36	200	7.25	$3Pb_3(PO_4)_2 \cdot PbBr_2$
8 L	46	200	6.33	$3Pb_3(PO_4)_2 \cdot PbBr_2 \rightarrow \alpha NH_4Br \cdot 2PbBr_2$
15 L	37	200	4.41	$3Pb_3(PO_4)_2 \cdot PbBr_2 \rightarrow 3Pb_3(PO_4)_2 \cdot PbBr_2 + \alpha NH_4Br \cdot 2PbBr_2 \rightarrow \alpha NH_4Br \cdot 2PbBr_2$
30 L	38	200	3.41	$3Pb_3(PO_4)_2 \cdot PbBr_2 \rightarrow \alpha NH_4Br \cdot 2PbBr_2 \rightarrow \beta NH_4Br \cdot 2PbBr_2 \rightarrow$ unknown M
90 L	39	100	5.5	$3Pb_3(PO_4)_2 \cdot PbBr_2 \rightarrow 3Pb_3(PO_4)_2 \cdot PbBr_2 + 2NH_4Br \cdot PbBr_2 + \alpha NH_4Br \cdot 2PbBr_2 \rightarrow 3Pb_3(PO_4)_2 \cdot PbBr_2 \rightarrow 2NH_4Br \cdot PbBr_2$
135 L	40	75	5.5	$3Pb_3(PO_4)_2 \cdot PbBr_2 \rightarrow 3Pb_3(PO_4)_2 \cdot PbBr_2 + 2NH_4Br \cdot PbBr_2 \rightarrow 3Pb_3(PO_4)_2 \cdot PbBr_2 + 2NH_4Br \cdot PbBr_2 +$ trace $\alpha NH_4Br \cdot 2PbBr_2$
270 L	41	19	15.75	$3Pb_3(PO_4)_2 \cdot PbBr_2 \rightarrow 3Pb_3(PO_4)_2 \cdot PbBr_2 + \alpha NH_4Br \cdot 2PbBr_2$

Note: → indicates increasing distance along foil from sampling orifice.

Table 10 – X-Ray Analysis of Foil Deposits – Test 7

Fuel, 80-octane base + 4.6 ml tel per U. S. gal + 1-T mix tcp

Engine Speed, rpm	1500	Jacket Temperature, C	60
Compression Ratio	5/1	Fuel/Air Ratio	0.075
Flame Arrival, deg late	9	Mixture Strength, % rich	12
Ignition Advance, deg early	29	Inlet Air Heat, at 20 C, watts	1350

Sampling Angle, deg early = E deg late = L	Sample Number	Gas Flow Through Sampling Valve, ml per min	Test Duration, hr	Deposit Analyses
10 E	107	200	2.5	$Pb_3(PO_4)_2$ high temperature form[a] + $PbO \cdot Pb_3(PO_4)_2$
5 E	72	200	3.0	$Pb_3(PO_4)_2$ high temperature form + $PbO \cdot Pb_3(PO_4)_2$
2 L	73	200	2.5	$Pb_3(PO_4)_2$ high temperature form + $PbO \cdot Pb_3(PO_4)_2$
8 L	74	200	3.0	$Pb_3(PO_4)_2$ high temperature form + some $PbSO_4 \cdot Pb_3(PO_4)_2$ + trace $PbO \cdot Pb_3(PO_4)_2$
15 L	75	200	3.0	$PbSO_4 \cdot Pb_3(PO_4)_2 + Pb_2(PO_4)_2$ high temperature form + trace $PbO \cdot Pb_3(PO_4)_2$
	106	200	32.5	$PbSO_4 \cdot Pb_3(PO_4)_2 + PbO \cdot Pb_3(PO_4)_2 \rightarrow PbSO_4 \cdot Pb_3(PO_4)_2 + Pb_3(PO_4)_2$ high temperature form + trace $PbO \cdot Pb_3(PO_4)_2$
90 L	76	123	7.0	$PbO \cdot Pb_3(PO_4)_2 + Pb_3(PO_4)_2$ high temperature form
270 L	77	22	14.5	Probably ($PbO \rightarrow PbO + PbO \cdot Pb_3(PO_4)_2$. (Pattern too diffuse for certain identification)

[a] This refers to the form stable at high temperatures referred to by Amadori.[3]
Note: → indicates increasing distance along foil from sampling orifice.

Table 11 – Limits of Detection

Lead bromide ($PbBr_2$) in the presence of 3/1 lead oxybromide ($3PbO \cdot PbBr_2$), %	10
Lead oxide (PbO) in the presence of $3PbO \cdot PbBr_2$, %	2½
$PbBr_2$ in the presence of 1/3 lead bromophosphate ($3Pb_3(PO_4)_2\ PbBr_2$), %	10
PbO in the presence of 1/3 lead bromophosphate ($3Pb_3(PO_4)_2\ PbBr_2$), %	10

0.060) showed little change in the above results with the exception that the range of occurrence of the ammonium lead bromides was markedly reduced.

With the fuel containing tel alone (Test 2) the earliest point of sampling was 10 deg early when the sample consisted entirely of lead oxysulfate ($4PbO \cdot PbSO_4$). This persisted till about 50 deg late after which the deposit was mainly lead oxide together with a small amount of the oxysulfate.

The sulfur in the oxysulfate was presumably derived from the base fuel which for all tests contained approximately 0.005% by weight of sulfur.

Test 3 was performed with a sulfur-free fuel, isopropyl alcohol, to investigate the effect of reducing the scavenger concentration to 0.5-T. With the exception of the samples obtained at 30 deg early and 30 deg late the sole product throughout the range investigated (30 deg early to 270 deg late) consisted of $3PbO.PbBr_2$. The sample at 30 deg early has not so far been identified. At 30 deg late the products were lead bromide, metallic lead, and the R-phase of lead oxybromide.[5]

In addition to the foil sampling work with the normal aviation fuel a few subsidiary tests were carried out to obtain information on the point in the cycle at which ethylene dibromide and tel first decomposed. A polished silver foil was placed in the sampling valve, and the first point in the cycle at which it became tarnished, due to the presence of free bromine, was noted. This occurred at tdc, suggesting that ethylene dibromide did not decompose before this point. In the range 90 deg early to 30 deg early two traps, cooled to —60 C, were placed in series with the sampling valve. The condensates were subsequently analyzed for ethylene dibromide and tel. The analyses showed that in this range the ethylene dibromide was recovered unchanged. Unfortunately the analyses for tel were partially invalidated by the procedure used for washing out the traps. It is hoped to overcome this and explore this aspect of the problem more fully in future work.

Tests with Tricresyl Phosphate (tcp) – Fig. 5 shows the results obtained in Test 4 when 0.2-T tcp was added to the normal aviation fuel.

From the earliest point sampled (130 deg early) until 10 deg early the sample consisted of $PbBr_2$ and lead bromophosphate, $3Pb_3(PO_4)_2.PbBr_2$. From 10 deg early till the arrival of the flame (approximately 10 deg late), the sole product was $3Pb_3(PO_4)_2\ PbBr_2$. At 11 deg late the product obtained was $\beta 3PbO\ PbBr_2$. Over the rest of the range, 15 deg late to 270 deg late, with the exception of the result at 30 deg late, the products were lead bromide and ammonium lead bromides. At 30 deg late there was in addition a compound referred to as lead bromo-pyro-phosphate which has the approximate composition $2PbBr_2.Pb_2P_2O_7$. The problem of what has happened to the phosphorus in the range 15 deg late to 270 deg late is being investigated further by examining the sampled gases.

When the concentration of tcp was increased to 0.5-T (Test 5) the bromophosphate first appeared at approximately 20 deg early. From 10 deg early to 10 deg late this was the sole product but at later angles, 10 deg late to 270 deg late the sample consisted of lead bromo-pyro-phosphate $2PbBr_2.Pb_2P_2O_7$, $3Pb_3(PO_4)_2.PbBr_2$, and ammonium lead bromides. Lead bromide only occurred at 270 deg late. Metallic lead was not detected at all with this fuel.

Raising the concentration of tcp to 1-T (Test 6) caused the lead bromophosphate to be present from the earliest point investigated, that is, 130 deg early. Otherwise the results are much the same as with the 0.5-T tcp except that no oxybromides were detected, the amount of ammonium lead bromide was much reduced, and free lead bromide was not detected at all after flame arrival in the end gas.

In the case of the fuel containing tel and 1-T tcp only, the earliest point investigated was 5 deg early, since the pattern obtained at earlier angles was not sufficiently clear for identification. The products obtained over the range 5 deg early to 90 deg late were lead phosphate ($Pb_3(PO_4)_2$), lead oxyphosphate ($PbO.Pb_3(PO_4)_2$), and lead sulfate orthophosphate ($PbSO_4.Pb_3(PO_4)_2$). At 270 deg late the product was a mixture of lead oxide and lead oxyphosphate.

The analysis of the combustion-chamber deposits from all tests is summarized in Table 12.

Discussion

Test with Ethylene Dibromide and Tetraethyl Lead – The lead bromide sampled during the early stages of the compression stroke, when running on aviation gasoline, must come either from deposits swept off the walls or from the residual gases left over from the previous cycle, since it has been shown that ethylene dibromide does not decompose until approximately tdc. It is unlikely that any free bromine survives from the previous cycle as this will combine with the tel thus reducing its antiknock efficiency. That this does not occur is shown by the approximate equal antiknock effectiveness of ethyl fluid and pure tel.

It would appear probable that the first appearance of lead oxybromides coincides with the point in the cycle at which tel first starts to decompose. No free lead oxide is detected before or during the appearance of lead oxybromides, and this would tend to indicate that immediately lead oxide is formed from the tel it combines with the lead bromide already present from the previous cycle to form lead oxybromide. However, one important reservation must be kept in mind. This concerns the existence of complex lead salts in the end gas.

[5] See *Analytical Chemistry*, Vol. 23, October, 1951, pp. 1388-1397: "Analytical Research in the Petroleum Industry (Symposium)," by F. W. Lamb and L. M. Niebylski.

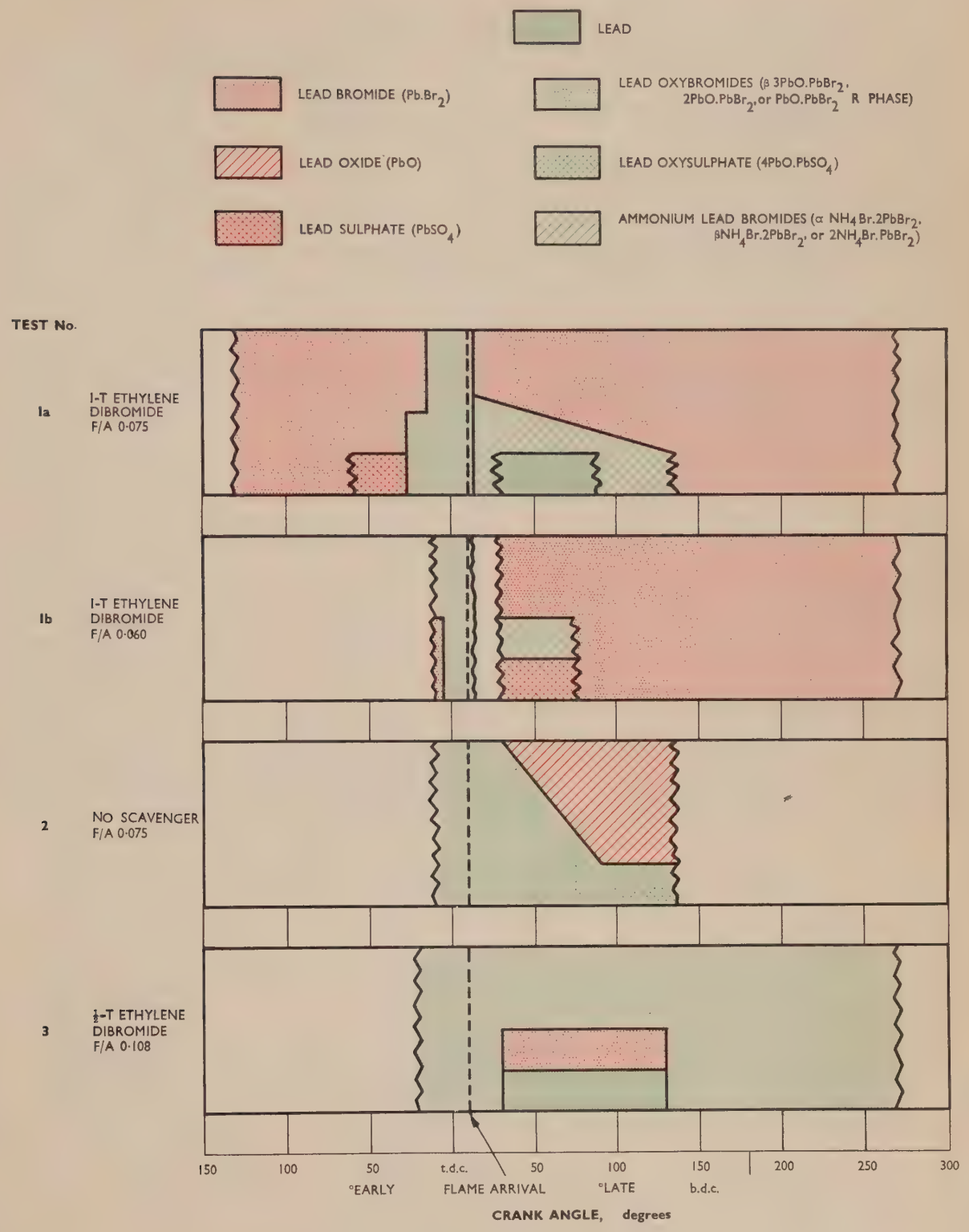


FIG. 4—Lead sampling tests with Ethyl fluid and pure T.E.L.

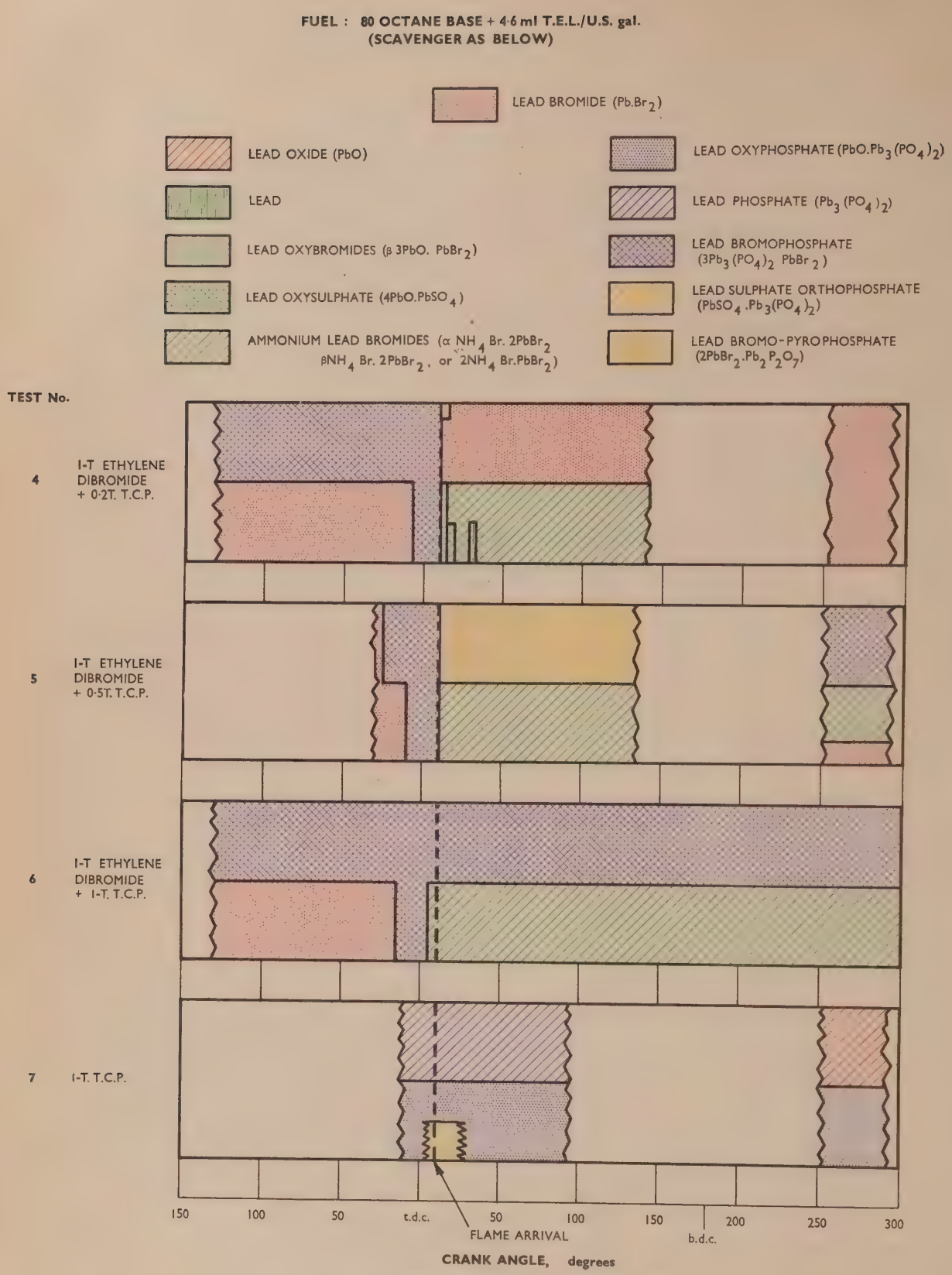


FIG. 5—Lead sampling tests with T.C.P. mixes

It is not yet certain whether the oxybromides occur as such in the engine cylinder, or whether lead oxide and bromide occur separately and combine to form oxybromides on their way to the sampling foil or on the foil itself. If this were the case, however, it would appear remarkable that no free lead bromide or oxide are found on the foil and that the amounts of free lead bromide and oxide emerging from the cylinder are in exactly the right proportions to react with each other. However, further work is in progress to establish this critical point. In particular the effect of variations in foil temperature is to be studied.

The abrupt disappearance of the oxybromides is also remarkable. This could be due to the sudden reduction in oxygen content at the time of flame arrival, although the test at weaker mixtures would appear to indicate that this is not the major factor. It may be that the additional amount of bromine liberated due to the decomposition of ethyl bromide at or about tdc also plays some part in the sudden reduction of the oxybromide concentration. The results obtained when the ethylene dibromide concentration was reduced to a ½-T would tend to support this mechanism.

The narrowness of the range of occurrence of the lead oxybromides is also of interest from another point of view. Combustion-chamber deposits collected at the end of the test definitely contain oxybromides and as it is improbable that these are only laid down in the narrow range in which oxybromides occur on the sampling foil it appears that their formation on the combustion-chamber walls involves reaction at the surface. This agrees with the conclusions of Lamb and Niebylski.[5]

The occurrence of ammonium lead bromides on the foil is also interesting. These compounds would not however be expected to exist in the combustion chamber as the temperature would be too high. Analyses of the fuel showed that the nitrogen was not derived from this source and hence must have come from the atmosphere. This is not surprising in view of the known presence of ammonia in the gases sampled from the engine cylinder.[1]

The fact that no free lead oxide has been detected in the range 10 deg early to 10 deg late has an important bearing on the mechanism of the antiknock action of tel. This aspect is discussed later.

It is also noteworthy that in the case of the normal aviation fuel there is no lead sulfate in the samples obtained after flame arrival. This contrasts with the fuel containing no bromine scavenger although the sulfur content of both fuels is the same, that is, approximately 0.05% by weight. This indicates that the bromine from the ethylene dibromide competes successfully with the SO_3 for the lead. It also indicates that the oxysulfates found on the exhaust-valve deposits are again probably formed mainly by reaction at the surface.

No free lead was detected when running on the fuel without bromine scavenger in contrast to the run with aviation gasoline. It may be that the free lead in the latter case is due to reduction of lead bromide on the surface. The reduction of lead oxide and lead sulfate is not so readily accomplished.

Tests with Tcp

The most significant effects with tcp are as follows:

1. Both the range of occurrence and the quantity of lead oxybromides are markedly reduced even with 0.2-T tcp. At the higher concentrations of tcp lead oxybromides are completely eliminated.

2. The amount of lead bromide formed is not reduced appreciably at the low concentration of tcp (0.2-T) but is very much reduced at higher concentrations.

3. With 0.2-T tcp the occurrence of metallic lead is markedly reduced and with the higher concentrations it is completely eliminated.

Table 12 – Analysis of Combustion-Chamber Deposits

Test No.	Scavenger	Fuel/Air Ratio	Piston: Pb (Metallic)	Br	PO_4	SO_4	C	X-Ray Analysis	Exhaust Valve: Pb (Metallic)	Br	PO_4	SO_4	C	X-Ray Analysis
1a	1-T $EtBr_2$	0.075	50.2	25.3	....	2.5	15.9	$PbBr_2 + 2PbO \cdot 3PbBr_2$	72.0[a]	....	....	4.6	6.8	.....
1b	1-T $EtBr_2$	0.060	41.9	24.3	....	1.6	23.0	$PbCl_2/PbBr_2$ mixed crystals	75.0[a]	....	....	8.3	6.1	$PbO \cdot PbSO_4$
2	None	0.075	70.0	....	....	7.0	13.8	Mixed crystals	70.0[a] (20.0)	....	....	3.8	17.7	.....
3	0.5-T $EtBr_2$	0.108	....	....	....	....	....	$2PbO \cdot PbBr_2$ + trace $PbO \cdot PbSO_4$	....	....	....	....	....	$2PbO \cdot PbBr_2$ on seats; $4PbO \cdot PbSO_4$ on heads
4	1-T $EtBr_2$ + 0.2-T tcp	0.075	47.0	14.6	6.1	3.5	22.7	$3Pb_3(PO_4)_2 \cdot PbCl_2/3Pb_3(PO_4)_2$ $PbBr_2$ mixed crystals → $PbCl_2/PbBr_2$ — mixed crystals	73.0	1.7	18.0	1.1	....	$3Pb_3(PO_4)_2 \cdot PbBr_2$
5	1-T $EtBr_2$ + 0.5-T tcp	0.075	42.0	10.2	8.3	2.0	....	$3Pb_3(PO_4)_2 \cdot PbBr_2 + PbBr_2$	51.7[b]	1.1	10.2	2.2	....	$3Pb_3(PO_4)_2 \cdot PbBr_2$
6	1-T $EtBr_2$ + 1-T tcp	0.075	44.0	4.0	13.0	0.8	25.0	.....	73.7	1.0	23.8	....	....	$Pb_3(PO_4)_2$ (high-temperature form)
7	1-T tcp	0.075	42.6	....	13.4	....	20.8	$Pb_3(PO_4)_2$ (high-temperature form)	67.0[a]	....	9.3	....	4.5	$3Pb_3(PO_4)_2 \cdot PbO$

[a] Corrected for corrosion products.
[b] Uncorrected although corrosion had occurred.

From the point of view of spark-plug fouling all of the above effects are of course desirable as the above compounds all have high electrical conductivities and low melting points.

Antiknock Action of Tel

Fundamental investigations into the mechanism of knock, involving the sampling of peroxides and aldehydes extracted from the end gas of a combustion chamber during the preflame period have shown that the period in the cycle at which tel exerts its antiknock action is between tdc and 10 deg late under the present test conditions.[1] In this work, the only compounds detected at this time in the cycle have been lead oxybromides, lead oxysulfate, lead bromophosphate, lead oxyphosphate, lead sulfate orthophosphate, and lead phosphate, according to the fuel under test. At no time has free lead oxide been detected. This would appear to be at variance with the modern view that at the time lead exerts its antiknock effect it is present in the form of lead oxide.[6] Even if the above compounds are dissociated in the end gas there would still be no lead oxide in the case of the bromophosphate.

[6] "The Mode of Action on Tetraethyl Lead as an Inhibitor of Combustion Processes," by G. H. N. Chamberlain, D. E. Hoare, and A. D. Walsh. Presented at discussion on "The Reactivity of Free Radicals," at Meeting of Faraday Society, Toronto, Sept. 8, 1952.

It thus appears that if the above compounds are not dissociated in the end gas during the appropriate time in the preflame reactions, then they must be equally effective as antiknocks to lead oxide itself. Since lead is thought to exert its antiknock action by means of a surface deactivation effect this would mean that each of these compounds contains something common in the surface crystal structure which enables them to be effective.

The work described in this paper is still incomplete and the conclusions that may be drawn are necessarily tentative, but sufficient information has already been acquired to illustrate the value of the technique described for the study of the mechanism of formation of various lead deposits.

Acknowledgments

The author wishes to thank the Shell Petroleum Co. for permission to publish this paper and to record his indebtedness to Ricardo & Co. (1927) Ltd., in whose laboratory the sampling technique was developed and the engine work carried out, and in particular to D. Downs and R. W. Wheeler who supervised the engine work and actively participated in the general planning of the program. Thanks are also due to Mrs. C. Orman, Dr. P. T. Davies, and Dr. C. G. S. Waghorn, who conducted the X-ray investigations.

DISCUSSION

Cites Importance of Lead Oxide Studies

– B. M. Sturgis

E. I. du Pont de Nemours & Co., Inc.

A KNOWLEDGE of the successive reactions of the lead introduced into an engine, from the time it exists as tetraethyl lead until it leaves the exhaust port or is deposited on the combustion-chamber walls, is of great importance in developing an understanding of the antiknock action of tetraethyl lead and of deposit formation and scavenging. Mr. Street has developed a novel method for the study of this problem and has obtained a considerable amount of new information. It is hoped that this work will be continued to such a point that the complete picture can be clarified.

A rather extensive study of the fate of the lead after the flame has passed – that is, during that part of the engine cycle in which deposit formation and scavenging take place – has been made by Newby and Dumont of the du Pont laboratories. Although this study supports many of Mr. Street's conclusions, it demonstrates that the role played by lead oxide is much greater than he has indicated. Results of this work were not quite in accord with our previous ideas. It is known that a number of different simple and complex lead compounds usually are found in combustion-chamber deposits. It was believed that the simple lead salts – lead chloride, lead bromide, and lead sulfate – were formed in the combustion space by vapor-phase reactions and deposited on the walls. Our investigations have demonstrated that this probably is not true, but that lead oxide is the only lead compound deposited on the walls under most conditions. After lead oxide has deposited in the solid state, then it can react with the acidic gases present in the combustion chamber – HCl, HBr, and SO_2 or SO_3 – to form the corresponding lead salts by means of gas-solid reactions.

When the flame passes through the combustion chamber very high temperatures result, and the lead present in the gas space, either in the form of tetraethyl lead or

some inorganic lead compound, is converted to lead oxide vapor. If an engine is operated on a fuel containing no sulfur or halogens, essentially the only lead compound found on combustion-chamber walls is lead oxide. Even when sulfur and halogens are present in the fuel, thermodynamic calculations show that lead oxide is the only lead compound which is stable at the gas temperatures existing during the expansion and exhaust strokes. The lead oxide, therefore, is deposited as such and later is converted to lead halides and sulfates.

That these calculations are correct in interpreting the reactions that actually take place in an engine was indicated by a series of experiments carried out in a single-cylinder engine so devised that the gas-solid reactions could be isolated and studied. The possibility of vapor-state reactions was eliminated by introducing the fuel additives into the combustion chamber separately. The fuel used was isooctane and the oil was an ester-type lubricant. These experiments consisted of operating the engine for 50 hr on two separate fuels, one containing tetraethyl lead without halogens at a concentration of 3.0 ml per gal, and the other containing one theory of chlorine as ethylene dichloride. These fuels were alternated every 2 min so that each fuel was used a total of 25 hr.

In these experiments, in which two different fuels were used, lead chloride could be formed only by a reaction between solid lead oxide on the combustion-chamber walls and the hydrogen halide in the gas, because at no time were the lead and halogen present together in the gas phase. For comparison, 25-hr tests were conducted in which the tetraethyl lead and ethylene dichloride were both present in the same fuel. The weight and composition of the deposits formed in the first test, in which only gas-solid reactions could take place, were compared with those in the second test in which there was opportunity for both gas-solid reactions and vapor-state reactions.

Similar experiments were conducted in which fuels were used containing ½-T of bromide as ethylene dibromide and 0.05 weight per cent sulfur as disulfide oil. From two to four experiments were conducted for each set of conditions.

A comparison of deposit weights for the two types of experiments showed that regardless of whether vapor-state reactions were possible or not, the amounts of deposits formed on both the cool and hot engine surfaces were identical within limits of experimental error. Moreover, the composition of deposits was not altered significantly by eliminating the possibility of vapor-state reactions, as shown in Figs. A and B.

The sequence of reactions of lead oxide in an engine is illustrated in Fig. C for most combustion-chamber surfaces, and in Fig. D for very hot surfaces such as exhaust valves. It is seen that the lead oxide first deposits on the surfaces and then reacts with the acidic gases to form the simple lead salts. These then can react further with lead oxide by solid-state reactions to form the complex lead salts. Seldom is any free lead oxide found in an engine deposit since both the acidic gases and the simple lead salts compete for it. On hot surfaces the lead oxyhalides are not stable and give volatile lead halides. The products found on such surfaces are largely lead oxysulfates.

These results indicate that some of the lead salts found in his sampling valve by Mr. Street, when sampling after the flame had passed, may have been formed on the surface of the valve by the reactions just described and were not present as such in the combustion gases. Further experimentation will be necessary to clarify this point.

Laboratory Data Correlated With Experimental Results

— Ellis B. Rifkin
Ethyl Corp.

THE author's work involves a novel technique which promises to further greatly the understanding of the mechanisms of lead deposition and removal in engines. In this particular respect some of the most interesting data presented relate to the molecular form in which the lead appears in the exhaust gases. In this laboratory certain comprehensive calculations have been carried out which will be published fully later, and which predict, for a given temperature, pressure, and fuel-air mixture, what compounds will be present in gaseous, liquid, and solid phases. In applying these calculations to the author's fuel and engine conditions, it has been found possible to obtain

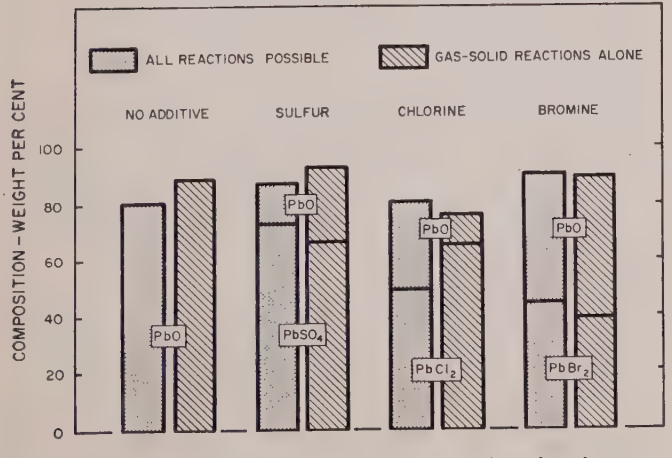


Fig. A – Composition of cool surface engine deposits

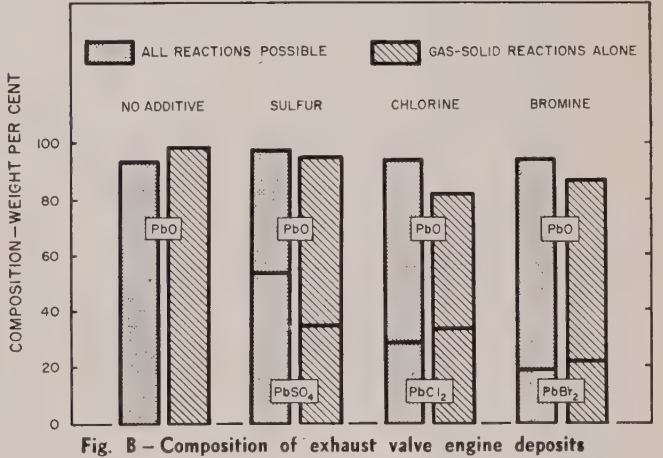


Fig. B – Composition of exhaust valve engine deposits

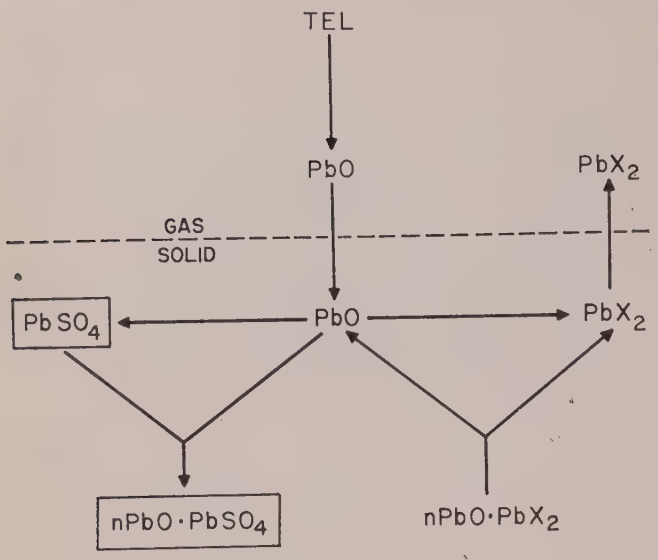


Fig. C – Normal deposit surface

a fair degree of correlation with the experimental results. Of particular interest is the prediction that elemental lead should appear in the exhaust gases from rich mixtures when equilibrium is reached at temperatures over about 300 C. This is indicated in the author's results, as shown by Tests 1a and 3 in Fig. 4, where the fuel contains 1-T and ½-T respectively of ethylene dibromide. In the absence of scavenger, where no elemental lead is obtained (Test 2 in Fig. 4), the calculations indicate that lead should appear in roughly the same amounts as in the presence of scavenger and in the same temperature range. The reason for this lack of agreement is not clear, unless it be due to the presence of sulfur in the experimental mixture, where it was not taken into consideration in the calculations.

The fact that an approach to a theoretical thermodynamic understanding of some of the author's results can be made, perhaps indicates that equilibrium at various temperatures rather than kinetic considerations are the important factors in determining the composition of the deposit on the foil. If this is true, it would seem that the technique might be quite difficult to apply to the problem of the mechanism of the antiknock action of tel, where kinetic factors are obviously of great importance. Nevertheless, the usefulness of the technique in studying the problem of the deposition and removal of lead compounds from the combustion chamber would seem to be extensive, and it is to be hoped that further work along the same lines will be accomplished.

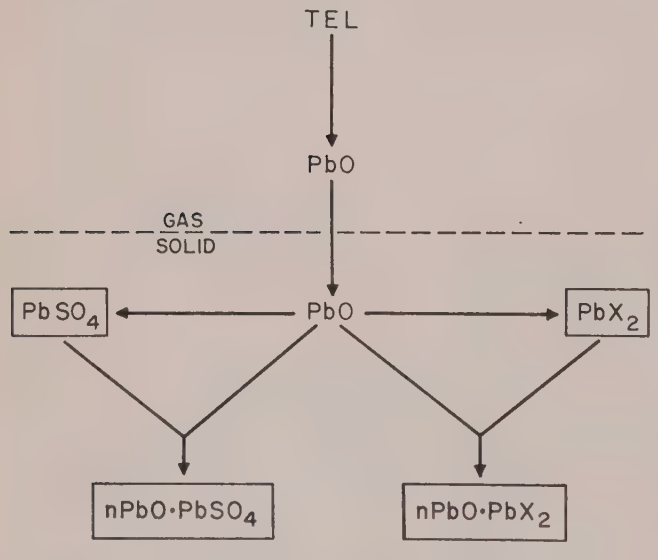


Fig. D – Very hot deposit and combustion-chamber surfaces

Author's Closure To Discussion

THE work described in this paper may broadly be divided into two parts, separated by the time of flame arrival at the sampling point. Before flame arrival, the end-gas temperature, although higher than that of the foil, is of the same order, and also no free bromine or hydrogen bromide exist in the gas phase before tdc, since the ethylene dibromide has not decomposed. In this region, therefore, it is probable that no large amount of reaction occurs on the way to the foil, though this point still has to be checked. It is also in this region that tel is exerting its antiknock effect.

The thermodynamic calculations referred to by the discussers are concerned with products after flame arrival and, for this reason, the fact as pointed out by Dr. Rifkin that they correlate to some extent with the present results obtained after flame arrival does not necessarily mean that the same conditions apply before the passage of the flame – in fact, it is extremely unlikely that they do so. The technique, therefore, should not be as difficult to apply to the problem of the mechanism of the antiknock action of tel as suggested.

The second phase of the work described refers to deposits obtained after flame arrival in the end gas. Here the temperature of the end gas is considerably higher than that of the foil, and Dr. Sturgis' comments are probably valid. In fact, a comparison of the results obtained with those obtained from analysis of combustion-chamber deposits had already led to the conclusion that reactions on the surface play a large part in determining the composition of the final deposits in the engine.

Having determined what the final products would be when collected on a surface at a low temperature, the next stage will be to increase the temperature of this surface in stages, and note the changes in the composition of the deposit. This technique should then yield important information on the nature of the reactions on the surface at various temperatures, thereby giving complementary information to that described by Dr. Sturgis.

A Method for Identifying Preignition

R. F. Winch and F. M. Mayes, Sun Oil Co.

This paper was presented at the SAE Annual Meeting, Detroit, Jan. 16, 1953.

CONTINUED increase in the compression ratio of automobile engines has given greater significance to the problem of preignition. While practically all investigators involved in the search for better fuels and engines have recognized the seriousness of the preignition problem, efforts to define, isolate, and control the many possible preignition sources have been hampered by the lack of a simple, accurate method for detecting and measuring it. This paper describes the development of a procedure and equipment which satisfactorily indicates the occurrence of preignition and which, it is believed, should be quite helpful to those working on the problem.

Preignition, as it applies here, can be defined as the initiation of a flame front at any point in the mixture before combustion initiated by the spark plug would normally reach that point. This definition has been purposely made all inclusive and encompasses all of those phenomena which have previously been described by such terms as surface ignition, hot-spot ignition, afterfiring, and wild ping. Detonation is eliminated from this definition only by the fact that detonating combustion does not conform to the conventional flame-front type of combustion.

Preignition, in its advanced stages, can result in extremely rough engine operation and is almost always accompanied by a loud and irritating knock. In its somewhat milder phases, it may not always

A NEW, simple method, applicable in the laboratory or on the road, for detecting even mild preignition, is described in this paper. The method requires a small amount of inexpensive equipment, easily installed without mechanical modifications to the engine. The procedure is likewise simple, accurate, and will yield reproducible results. Mild preignition can be detected, regardless of the location of the source, and the results also clearly indicate the difference between preignition and detonation.

An oscilloscope is used to identify preignition; detonation is determined by its audible knock. With increase in compression ratios and power output necessitating study of these phenomena, it is expected that proper instrumentation for differentiating between the two will be extremely useful.

The Authors

R. F. WINCH (J '52) has been a research engineer in the Automotive Laboratory of the Sun Oil Co. since 1947. He graduated from Swarthmore College in 1945 with a B.S. in mechanical engineering and served in the U. S. Navy.

F. M. MAYES has been a research physicist at the Physical Research and Development Laboratory of the Sun Oil Co. since 1945. He graduated from Eastern Kentucky State Teachers College with a B.S. in physics and received his M.S. in physics at the University of Kentucky.

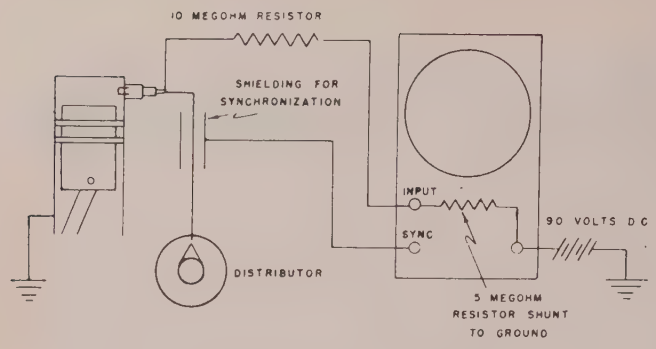


Fig. 1 – Schematic diagram of equipment

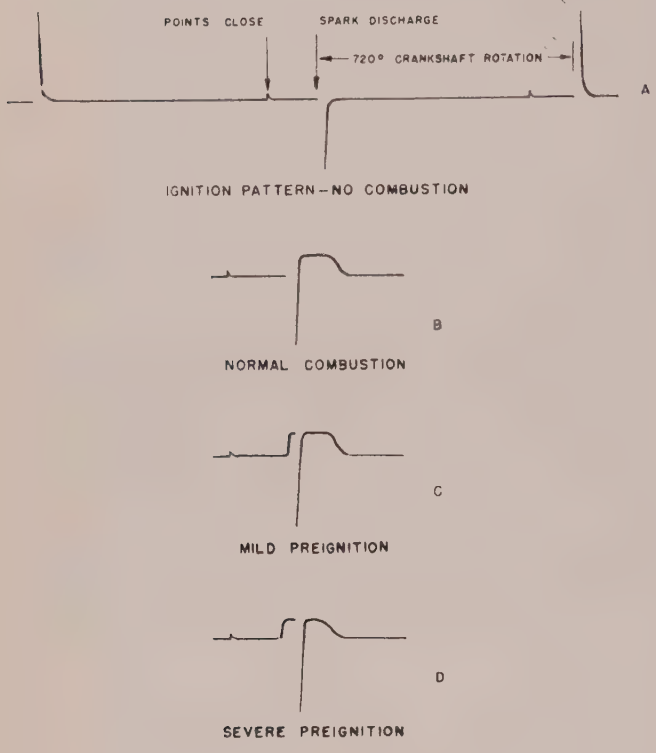


Fig. 2 – Preignition observed through spark plug. Hot spark plug is preignition source in single-cylinder engine

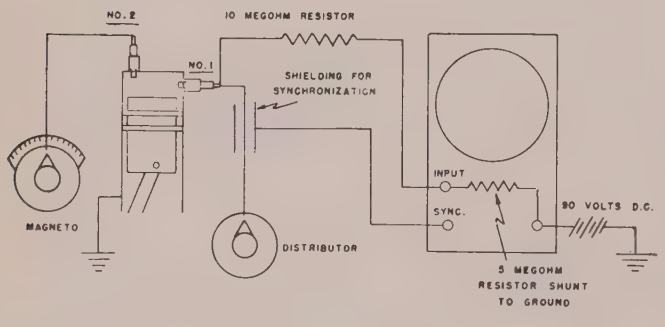


Fig. 3 – Schematic diagram for end-gas preignition evaluation

result in knock, but it always results in higher than normal temperature and pressure which, in turn, tends to increase engine octane requirement. In such an instance, the engine is not truly knock-limited but in reality is preignition-limited. This situation is apparently occurring with increasing frequency in current high compression-ratio engines, but without equipment which can differentiate between detonation and preignition, the two phenomena cannot be studied or controlled independently.

The various procedures used to detect preignition in the past have many disadvantages, particularly in road testing applications. On the road, the most commonly used methods of attempting to distinguish between detonation and preignition are based on differences in noise level and tendency of the car to run with the ignition turned off. Such methods are inaccurate, not too reproducible, and, in many instances, they are not capable of indicating preignition when it occurs. Laboratory methods, on the other hand, while somewhat more precise, frequently require elaborate, unwieldy, and expensive equipment not adaptable for road work. For example, observations of power loss, pressure indications, and photographic observations through quartz combustion-chamber windows are usually included in laboratory investigations.

The method which has been developed in the Sun laboratory and which is described herein, appears to have overcome many of the disadvantages of previous methods in relation to required equipment.

1. Equipment is simple, easy to construct and operate, and comparatively inexpensive.
2. It is applicable to single or multicylinder engines in the laboratory or on the road.
3. It is easy to install, with no mechanical modifications to the engine.
4. It is sensitive to mild preignition, regardless of location of the source.
5. The investigator is able to distinguish between preignition and detonation.
6. The method leads to accurate and reproducible results.

Description of Method

1. *Basic Operating Principle* – This method of indicating the occurrence of preignition is based on the fact that the electrical properties of the spark-plug gap will vary, depending on the time in the cycle at which combustion is initiated and depending on the rate at which the combustion process is completed; that is, whether or not secondary flame fronts are present. Specifically, changes occur in the conductivity of the gap which are reflected in variations in current flow. These variations in current flow, when observed on an oscilloscope, serve as an excellent means of detecting the presence of abnormal combustion. In order to utilize this principle to best advantage, the normal current flow associated with the spark dis-

charge has been supplemented by the use of a constant d-c voltage across the gap. The difference between the current-flow pattern produced on the oscilloscope during a preigniting cycle and a normal combustion cycle is sufficiently great that change from the normal pattern is easily seen. Detonating combustion, on the other hand, does not disturb the normal combustion pattern unless the knock becomes excessively severe.

2. *Description of Equipment and Circuit* – The equipment and circuit used in this method are shown in the schematic diagram of Fig. 1. The effects of the combustion process are impressed on the oscilloscope circuit as variations in the current flow across the spark-plug gap. The 10-megohm, 1-w resistor between the spark plug and the vertical input circuit of the oscilloscope is necessary to protect it from the very high voltages present in the ignition system secondary when the spark discharges.

While the engine ignition system itself provides the necessary circuit voltage during the relatively short period of the spark discharge, it is necessary to provide a constant external voltage across the spark gap during the remainder of the cycle. This voltage, which is supplied by a 90-v d-c source, causes the oscilloscope to act as an ammeter, thereby providing a measurable indication of the current flow across the gap during the entire cycle. The 5-megohm shunt shown between the input circuit and the voltage supply is added to complete the path for this direct-current flow.

In order to synchronize the horizontal sweep of the oscilloscope to the speed of the engine, a signal is picked up from the magnetic field of the secondary wire of the engine ignition system by a short piece of metallic shielding.

The same circuit and equipment can be used in cars on the road. The only additional equipment required is a 110-v a-c power supply which must be made available to operate the oscilloscope. A radio-type vibrator will invert the 6-v d-c available in the car to the necessary 110-v a-c.

Demonstration of Method

In order to simplify the discussion as much as possible, the method will first be demonstrated in use on a single-cylinder engine under normal and preigniting conditions, and then it will be demonstrated in use on a multicylinder engine. In both of these demonstrations, the preignition has been induced by the use of artificial preignition sources. By this means, it has been possible to provide a definite, easily located point of preignition origin, which greatly simplifies the demonstration of the capabilities of the method. Since the preignition was induced only to show its effect and the effect of its location on the spark-plug gap, how it was originated is not important.

1. *Single-Cylinder Engine Application* – In relation to the spark plug, the two extreme locations at which preignition could originate would be (a) at the spark plug or (b) at the farthest end of the combustion chamber from the spark plug. Satisfactory identification of preignition occurring at the two extreme locations provide an indication that the method will work for points of origin anywhere between the extremes.

A. Preignition originating at spark plug: The performance of the method will first be demonstrated with the preignition source located at the spark plug.

Before any attempt is made to describe differences between normal and abnormal combustion cycles, it is necessary to clarify the nature of the oscilloscope indications by examining the basic trace observed with a nonfiring engine. Fig. 2A shows the ignition current-flow pattern of the engine operating without fuel, as it is observed through the 10-megohm resistor on the oscilloscope. The engine was being motored by a dynamometer.

The parts of the ignition process shown in the picture are the closing of the ignition points and the spark discharge. When the points close, there is a current flow in the primary ignition circuit which causes a magnetic field about the secondary circuit. An induced signal is picked up by the oscilloscope from this field causing the small pip in the picture. For this particular engine, the point closure occurs 140 crank angle deg before the spark discharge, which occurs when the points are again opened, causing the magnetic field about the coil to collapse. The high, induced secondary voltage, through the action of the condenser in the primary circuit, oscillates at a frequency of about 3000 cps. These high-frequency oscillations are filtered out of the signal seen on the oscilloscope by the 10-megohm resistor in the input circuit; the blank space in the picture is this high-frequency period, the high spike being the last stages in the decay of the spark signal. As can be seen, this first indication can occur at either a positive or negative part of the discharge, depending upon the polarity at the time the oscilloscope is first capable of following the pattern.

Fig. 2B shows the change in the pattern which occurs as a result of combustion. The spark discharge is again indicated by the blank space, but the rest of the picture has been changed by the combustion process. Instead of a normal dissipation, as occurs in the absence of combustion, a hump or peak, as it will be called, has resulted. It will be shown later that this peak results from a combination of the potential generated in the spark gap by the combustion process, together with the increased conductivity of the air gap.

Bringing a preignition cycle into the picture for the first time, Fig. 2C shows the characteristic pattern when the engine is brought into a mild preignition condition with the preignition originating at the spark plug. This was done by using a hot-running spark plug and gradually increasing the output of the engine which, in turn, increased

the temperature of the spark plug up to the point where it served as a source of ignition. In contrast to the normal combustion pattern, it will be noted that a part of the combustion portion of the picture now precedes the spark discharge. This is an indication of the flame front that was formed before the spark occurred. In order for preignition to originate at the normal ignition source, it would have to precede the spark impulse, as the picture indicates, or the combustible in that area would have been consumed by the flame emanating from the spark discharge.

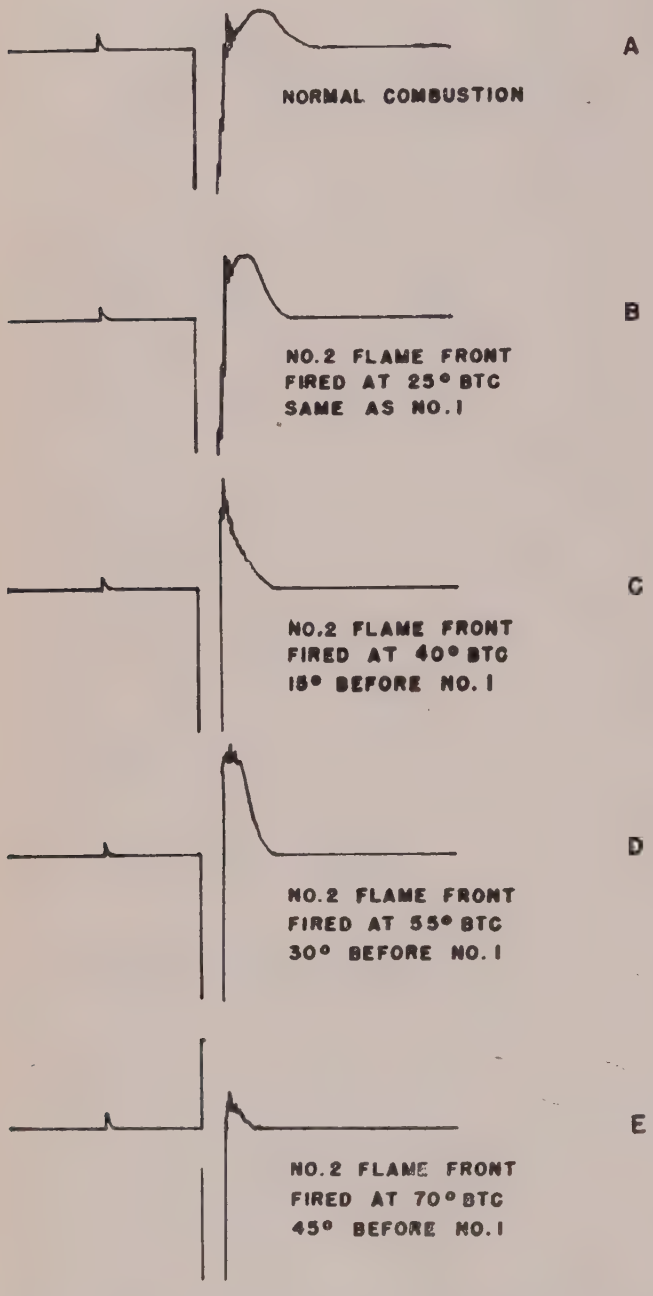


Fig. 4 – Effectivenes of spark plug at indicating preignition originating in end gas. No. 1 spark plug fired from coil at 25 deg btc. No. 2 spark plug fired from magneto as imitation preignition source

Fig. 2D shows the preignition in a more severe condition; the portion preceding the spark has become larger. As the severity increases, this portion continues to grow larger, moving more and more to the left, and indicating the occurrence of preignition earlier and earlier in the compression stroke.

From the foregoing discussion, it is apparent that, when preignition occurs at or near the spark plug, it can be observed quite readily through observation of the current flow across the spark plug.

B. Preignition originating in end-gas region: A preignition flame front starting in the end-gas region, as far from the spark plug as possible, can also be identified. Fig. 3 shows the schematic arrangement of the engine as it was operated to demonstrate this condition. A second spark plug, fired by the magneto, has been used to simulate the preignition source. By changing the timing of this magneto, the severity of the preignition can be altered and controlled.

Fig. 4 shows a series of pictures demonstrating the effect of preignition initiated in the end-gas zone at various times in the engine cycle. In this series, Fig. 4A, the normal combustion condition, is slightly different from the normal picture in Fig. 2. The difference is the result of the horsepower output level at which the engine was operated for the two series of pictures. The power level was lower in the Fig. 4 series, because heat was not necessary to produce the preignition for this demonstration.

Fig. 4B shows preignition originating in the end-gas area at the same time as normal combustion occurs. The peak of this picture is slightly higher than the peak of the normal combustion picture. Fig. 4C shows preignition occurring 15 deg before the normal flame. The peak is even higher and is moving to the left. The movement of the peak to the left can best be seen by comparing its position in each picture to the peak position during normal combustion. In Fig. 4D, the preignition starts 30 deg before the spark discharge and, finally in Fig. 4E, with preignition starting 45 deg before normal combustion, the preignition flame has crossed the combustion chamber and caused a signal on the scope before the normal spark discharge.

End-gas preignition is, thus, different in its effect on the spark-plug gap from the preignition occurring in an area close to the spark plug, although its detection is just as easy and just as precise. End-gas preignition increases the height of the combustion peak and moves its location to the left as the severity increases, while preignition at the spark plug is identified by the signal produced prior to the occurrence of the spark discharge.

2. *Multicylinder Engine Application* – The application of this method to a multicylinder engine is exactly the same as was described for the single-cylinder engine, with one exception. Ten-megohm resistors are connected to each spark plug and then to a common lead to the oscilloscope. The synchronizing signal is taken from the No. 1 spark-plug wire.

Fig. 5 shows the indications obtained from a multicylinder engine during normal and preignit-ing operation. To demonstrate the effect of pre-ignition occurring both in the end gas and at the spark plug, it was again introduced by artificial means. Two cylinders of the engine (3 and 4) were equipped with hot spark plugs, and cylinder 6 had an end-gas exciter. The first series of pictures, Fig. 5A, shows the normal combustion pattern for the engine. The variation between pictures from the different cylinders is typical of the variation that occurs between combustion cycles during normal operation. These differences are not critical, how-ever, since preignition itself causes changes in these pictures that are considerably greater.

Fig. 5B shows the effects of mild preignition in cylinders 3 and 6. Preignition in cylinder 3 is occur-ring at the hot spark plug, just before the spark discharge. In cylinder 6, the preignition originating in the end-gas zone results in a peak higher than normal and closer to the spark discharge. Cylinder 4, despite the presence of a hot spark plug, is still operating in a normal manner.

Fig. 5C shows a condition where cylinders 3, 4, and 6 are all preigniting. Cylinders 3 and 6 are about the same as in the previous series, but 4 is preigniting severely. A large portion of the com-bustion signal now precedes the spark impulse, indicating that a flame front was started during the compression stroke. The next series, Fig. 5D, shows severe preignition in all three cylinders. A signal appears for cylinder 4, for instance, at about 40 deg of crankshaft rotation before the spark impulse. In cylinder 6 where end-gas preignition was occurring, the peak is higher than in either of the previous cycles.

The preignition in each of the cases just dis-cussed has been artificially induced, since hot spots were added to the engine. However, these pictures are representative of the indications that are ob-tained when the preignition source is some condi-tion natural to the engine, such as combustion-chamber deposits.

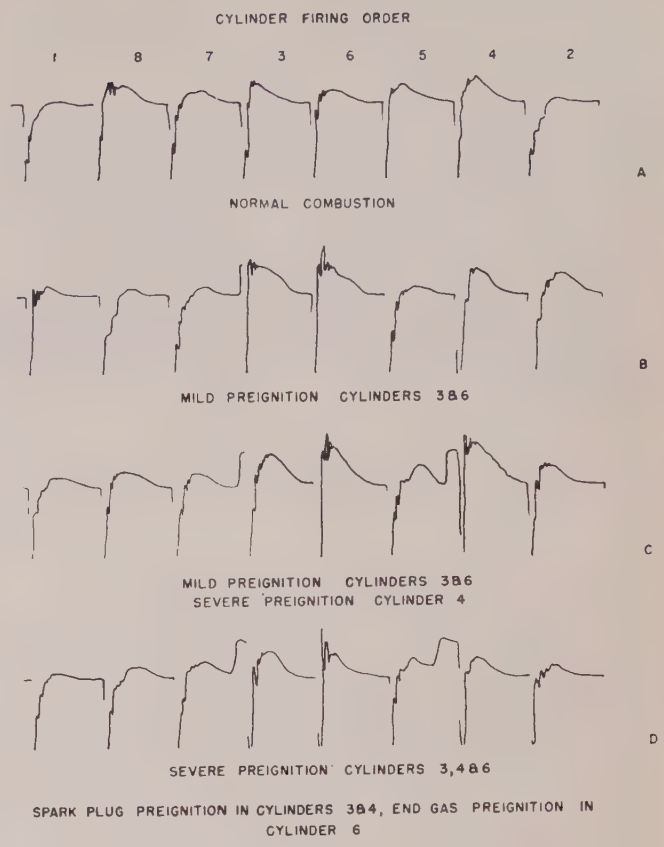


Fig. 5 – Preignition occurring in a multicylinder engine

Effect of Detonation

Detonation has no effect on the oscilloscope pat-tern, unless it becomes excessively severe. This is true regardless of whether the engine is preignit-ing or not, the preignition pattern being essentially the same with or without detonation present. Like-wise, the pattern associated with normal combus-tion is not changed when detonation is present in the engine.

However, wild ping – the wild, erratic, loud, sharp rap frequently heard in engines these days and usually associated with preignition *can* be seen on the oscilloscope. Wild ping is believed to be the result of detonation which has been induced by severe preignition. The process probably occurs something like this: a hot spot, perhaps a piece of incandescent carbon, starts a flame front early in the compression stroke, long before the spark dis-charge. This leads to detonation of a very large quantity of the mixture, a much larger quantity than is usually involved in more normal detonation. The source of the ignition is of a transient nature, and in consequence the severe knock occurs for only a few cycles at a time.

The picture produced by this wild ping is unique. Fig. 6 shows the characteristic indication of a wild ping on the oscilloscope. It will be noted that the peak involved is very high and that there is a very large area included under the peak. Also, the occurrence of the noise and the oscilloscope indi-cation is simultaneous.

Practical Consideration of Method

It has been demonstrated in the previous dis-cussion that this method works; but it is also of interest to know what is actually happening in the

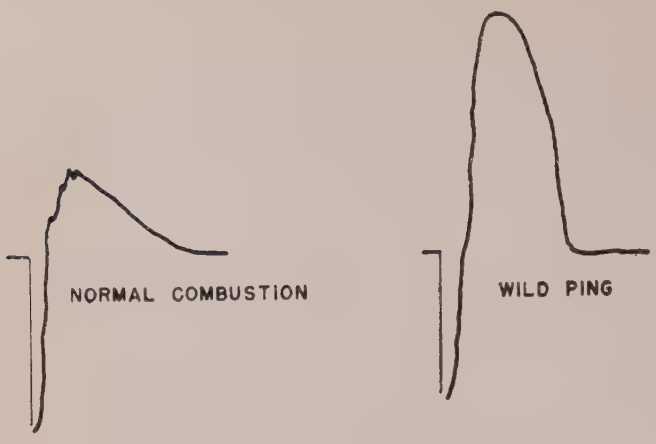


Fig. 6 – Effect of wild ping on spark-plug voltage as viewed on oscilloscope

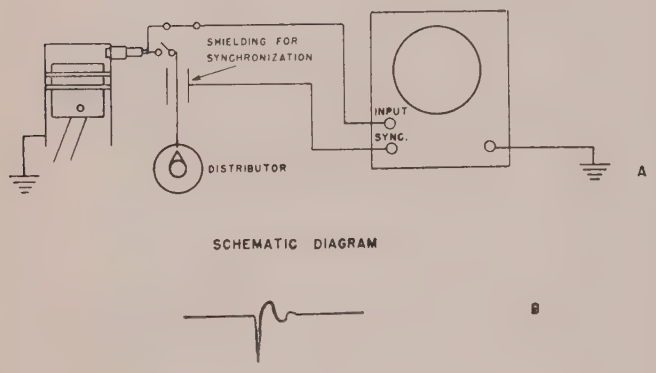


Fig. 7 – Current flow generated in spark gap by combustion

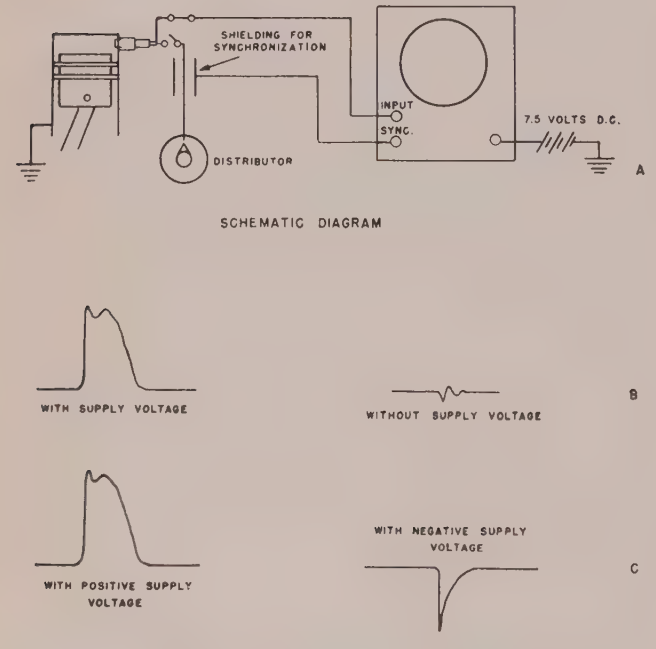


Fig. 8 – Effectiveness of an ionization gap at indicating combustion

spark gap that makes it work. In attempting to answer this question, it is necessary to consider two things:

1. What are the changes that occur in the spark gap that produce the observed indications?

2. What property or properties of the combustion process bring about these changes?

1. *Changes that Occur in the Spark Gap* – The combustion process has been shown to change the electrical properties of the spark gap. There are two ways in which these properties are changed:

A. A voltage is generated in the gap.

B. The electrical conductivity of the gap is increased.

There is some evidence that both of the above effects combine to produce the observed oscilloscope patterns. The first, however, is relatively minor; it is the second condition which is most useful to this test procedure. The relative effect of these two factors will be demonstrated in the following discussion.

A. Voltage generated by the combustion process: A current flow is generated in the spark gap by thermoelectric or thermionic effects as a result of combustion or the heat generated by it. However, observation of the combustion process by means of a spark plug does not permit isolation of this generated effect, unless the spark plug discharge potentials are removed from the spark plug. To observe these generated potentials by themselves, the engine has been fired without the aid of a spark discharge. A high-temperature spark plug was used as a source of ignition, and the engine was operated with the spark-plug porcelain serving as the ignition source. There was no electrical energy supplied externally to the spark plug.

To observe the generated potential, the oscilloscope was connected directly to the spark plug without any resistor, shunt, or external voltage, as shown in Fig. 7A. It was possible in this way to observe on the oscilloscope the current flow generated in the spark gap. Fig. 7B shows this generated current flow and demonstrates the fact that there is electrical activity in the spark gap brought about by combustion.

However, this signal is so weak it is of little value when combined with the extremely high voltages of the spark discharge. The signal shown in Fig. 7B was obtained with the amplification of the oscilloscope increased 10 times over that used in the previous pictures. A signal as weak as this is insignificant in comparison to the intensity of the spark discharge signal.

B. Effect of combustion on electrical conductivity of spark gap: The generated potential, as described above, is of no value for indicating combustion in a cylinder, because of its relative weakness compared to the spark discharge with which it would be combined. However, combustion, in addition to generating a potential across the spark-plug gap, also gives rise to a considerable change in the conductivity of the gases between these

points. It is this fact that enables the spark gap to be used as a means of indicating the presence of preignition.

In order to demonstrate the changes that take place in the conductivity of the gases between the spark-plug electrodes, a constant d-c voltage supply has been added in the circuit so a current will flow across the gap when combustion occurs. This current flow can then be observed on the oscilloscope; and, if a high enough voltage is used, it will produce a significant signal when combined with the spark discharge. Here again, the engine has been operated by means of preignition with the ignition system turned off. This time a 7.5-v dry cell has been added in series with the spark plug and oscilloscope, as shown in Fig. 8A. What happens now? Combustion starts, and the resulting chemical activity generates the electrical potential previously discussed. The gap is ionized and current from the outside voltage supply, in addition to that which was generated, flows across the gap, permitted and controlled by the degree of ionization of the gap. If the polarity of the outside voltage agrees with that of the generated voltage, the current flows will be additive. If the two voltages are not of the same polarity, the current flows will oppose each other.

When an outside voltage supply is used, the current flow is sufficiently high so that the amplification of the oscilloscope can again be reduced while still producing a usable picture. Fig. 8B shows the current flow pattern produced by combustion and its relative size in comparison with the picture obtained with no d-c voltage on the spark plug. It can be seen that combustion can be observed much better by noting its effect on the electrical conductivity of the air gap, rather than by just observing the potential it generates in that air gap. Fig. 8C shows the effect of changing the polarity of the supply voltage. When the spark plug is 7.5 v above ground, the generated and supplied current flow are additive in the gap; but when the spark plug is 7.5 v below ground, the two currents oppose each other, reducing both the height and area of the picture. The upside-down picture would be expected in the latter case as a natural consequence of polarity reversal.

An additional effect of the ionization of the spark gap by combustion is the reduction that results in the breakdown voltage of the spark discharge. If combustion is already present in the spark gap at the time the spark discharge occurs, the gap is already ionized by this combustion and the coil voltage does not have to build up as high for the spark to jump the gap. This reduction in peak voltage and the greater conductivity of the gap also tends to dampen out the oscillations immediately following the discharge and to hasten the decay of the spark. However, since these effects are slight compared to the voltage of the spark discharge and the input circuit restricts this portion of the picture, changes occurring during this portion of the combustion cycle are not apparent on the oscilloscope.

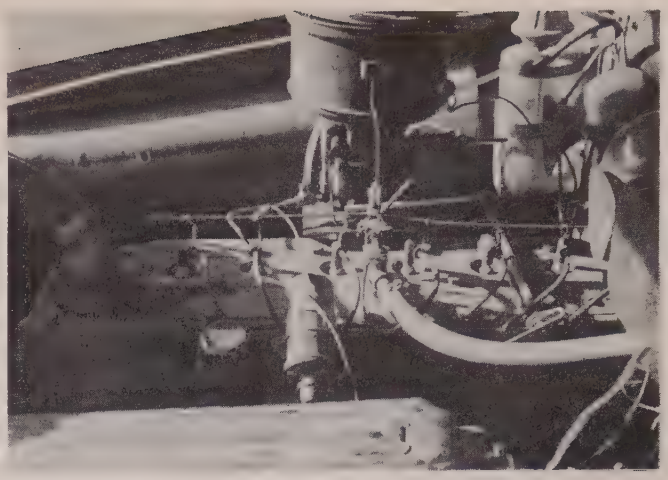


Fig. 9 – Resistor pickups and shielding for synchronizing signal

2. *Effect of Combustion Process on Electrical Conductivity* – The method by which the combustion process changes the spark-gap conductivity is not clearly understood, but a certain amount of experimental evidence, which will be discussed later, has led to the following theory:

A high degree of ionization of the spark gap is initially brought about by the chemical activity associated with the flame front when it passes through the spark gap. Once the gap has been ionized by this flame front, it is electrically conductive, thereby permitting current to flow between its two electrodes. Pressure and/or temperature changes occurring in the combustion chamber can alter the chemical activity of the burned gases that exist in the area behind the flame front so that the degree of ionization of the gap is altered by such changes. This change of ionization thus alters

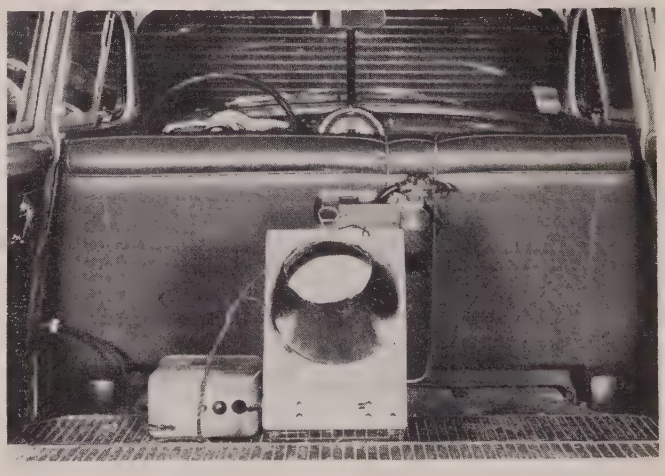


Fig. 10 – Vibrator and oscilloscope installed in car for road testing

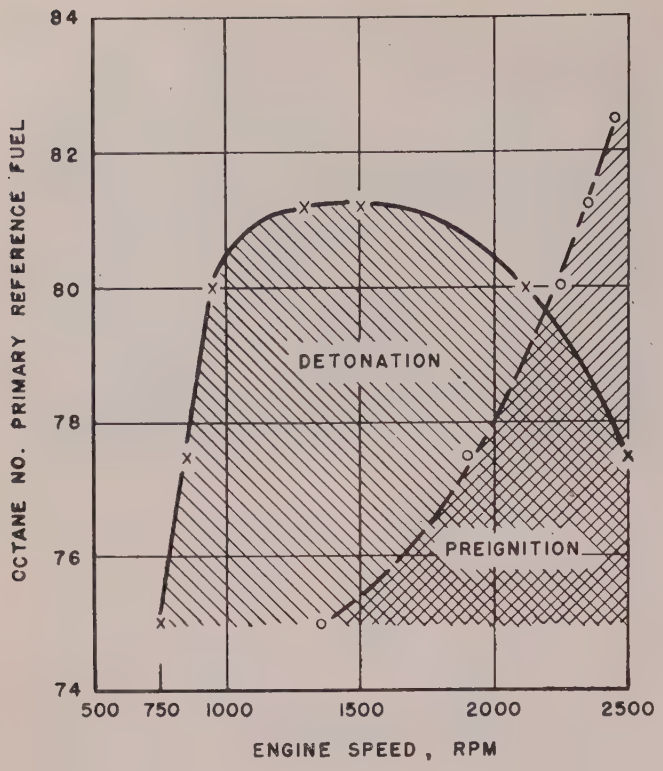


Fig. 11 – Octane and preignition requirements of clean, high-compression ratio engine

the conductivity of the spark gap and, in turn, the current flow across the gap. However, if the gap is not first ionized by the flame front, pressure and/or temperature will be ineffective in causing any change in the essentially nonconductive spark gap.

In an effort to support this theory, reference is again made to Fig. 4. As was previously described, two spark plugs were present in the engine at opposite ends of the combustion chamber. No. 1 was the normal ignition source through which end-gas preignition was observed. The end-gas preignition source was the No. 2 spark plug, which was fired by a magneto at various degrees of advance.

The well-known fact that a spark gap is ionized by the flame front when it passes through is illustrated in Fig. 4E. Preignition had to be started 45 deg before normal ignition occurred in order for combustion from the preignition to cross the chamber and ionize the spark plug before the normal spark discharge. This 45 deg is equivalent to a speed of 55 fps across the 3-in. combustion chamber of this engine which was operating at 1500 rpm. This is in quite close agreement with flame front speeds of about 65 fps which have been quoted for comparable combustion chambers. In computing the velocity in this engine, no allowance could be made for ignition lag at the preignition source. If the time of this ignition lag were known and considered, these two figures would probably be in even closer agreement. In any event, there seems to be little doubt that the indication in Fig. 4E is a result of ionization by the flame front as it passed through the spark gap.

At the same time, Fig. 4E indicates that temperature and pressure changes had no effect on the ionization gap before the flame reached the gap. Such changes were occurring in the spark gap while the flame front was crossing the combustion chamber, but they did not permit any current to flow. The gap remained un-ionized until the flame front arrived. Therefore, as previously stated, if the gap is not already ionized by combustion, temperature and/or pressure will not cause any significant change in the electrical conductivity of the gap.

However, Fig. 4B indicates that temperature and/or pressure do have some effect during that portion of the cycle which follows the normal spark discharge. In this figure, combustion was started at both the spark plug and the preignition source at the same time. The preignition flame front itself could not have caused the observed change in the spark gap, since it could not have reached it in time. Nevertheless, a change did occur, and the gap was more conductive in the presence of two flame fronts, as can be seen by comparing Fig. 4B with Fig. 4A. The increased temperature and/or pressure associated with the multiple flame-front combustion must have produced the difference.

From the foregoing discussion, it seems reasonable to conclude that variations in temperature or pressure within the combustion chamber will change the conductivity of the ionization gap if the gap has been previously ionized by a flame front. If it is not ionized, these two conditions will cause no significant change in the gap.

Sample Application of Method

The purpose of this paper, as was pointed out in the introduction, is to describe the equipment and procedure that have been developed by the Sun laboratory for the detection of preignition. It has been extremely useful to Sun in its experimental work on the preignition problem, and the thought behind the presentation of the paper is that it might also prove to be useful to others working in this same field. A detailed discussion of the test data that have been obtained up to the present is, thus, not suitable to the purpose of the paper, although it was felt that it would be of interest to illustrate the kind of information that can be obtained by use of the method.

Figs. 9 and 10 show the equipment installed in a car for operation on the road. Fig. 9 shows the resistors on the spark plug and the piece of shielding on the No. 1 spark-plug wire for synchronizing the oscilloscope. Fig. 10 shows the oscilloscope in the car and the vibrator for supplying the 110-v a-c power for the oscilloscope.

Fig. 11 shows data obtained on a recent model

high-compression engine in connection with a study of the effect of combustion-chamber deposits on detonation and preignition. The combustion chambers of the engine were perfectly clean in this instance, and all of the data were obtained under the full throttle, level road, accelerating conditions normally used in octane requirement work. The octane requirement curve was obtained in a conventional manner by observing aurally the appearance and disappearance of knock. At the same time, preignition was observed by means of the method described in this paper. A number of interesting things are apparent from the figure.

First of all, even though the engine was perfectly clean, there was considerable preignition present. This preignition would have been even worse with commercial fuels which are lower in their resistance to preignition than are primary reference fuels. Thus, the well-laid plans of this particular engine builder for control of combustion and such related functions as rate of pressure rise, were pretty well confiscated by preignition.

The figure also illustrates the fact that preignition can occur without producing audible knock. Following across the 78 octane line, as an example, it is seen that knock appears at about 850 rpm. Going on up through the acceleration, knock continues without the presence of preignition up to 2000 rpm when the engine begins to preignite. Both preignition and knock continue up to about 2450 where the knock dies out. The preignition, however, continues and actually becomes more and more severe as the speed is increased.

The chart also serves to illustrate the difficulty of trying to study the phenomena of knock and preignition simultaneously without proper instrumentation for differentiating between the two. At octane levels above 81.5, for example, no knock can be found, although preignition is apparently going to be present at speeds above 2500 rpm, more or less independent of the octane quality of the fuel used.

Summary

In conclusion, it is hoped that the foregoing discussion has adequately described the equipment and procedure which have been developed. It is hoped that it will be useful to others who, like ourselves, have arrived at the conclusion that preignition is a problem that is going to have to be wrestled with if progress is to continue in the field of mutual adaptation of fuels and engines. One big advantage of the technique is that preignition and detonation can be separately identified, preignition by its effect on the oscilloscope and detonation by its audible knock. Its other big advantage is its simplicity. The method can be used with any type of automotive engines, requiring no mechanical changes to or linkage with the engine. It, thereby, provides a means of studying preignition on single or multicylinder engines, either on the road or in the laboratory.

DISCUSSION

Presents Results of Autoignition Tests

— F. W. Bowditch
Research Laboratories, GMC

THE authors are to be congratulated for having adapted the ionized gap principle to the standard automotive ignition system. We believe this principle has definite possibilities for the testing of autoignition tendency of cars on the road. And it is on this account that General Motors research laboratories has also conducted work on the possibilities of this system.

Our first difficulty with the instrumentation was found to be the correct selection of 10-megohm, 1-w resistor in the test circuit. The high voltages produced by the ignition system evidently render some resistors unsuitable for this purpose. Also it was found that only certain oscilloscopes would give the type of ionization trace shown by the authors. This was found to be due principally to the variations in the input circuits and the return times of the various oscilloscopes.

We then decided to investigate the validity of the interpretations of the ionization traces made by the authors. In order to do this the same oscilloscope model and a resistor of the same manufacture as the authors used were set up in the indicator circuit on the General Motors Research window-engine. This engine was fitted with 2 spark plugs, 2 in. apart, whose timing could be independently varied. The spark plug to which the author's preignition circuit was attached was fired by a standard Chevrolet ignition system.

Fig. A shows the flame photographs and the ionization trace of simulated advanced preignition. Time increases from left to right and from top to bottom in the photographs. The time of each photograph in crank angle deg is shown below each picture, the negative numbers indicating deg btc. The plug from which the ionization trace was taken, hereafter called the No. 1 plug, was fired at 10 deg btc and is situated at the lower left-hand corner of the combustion chamber in the photographs. The other plug, situated in the upper left-hand corner of the combustion chamber, hereafter called the No. 2 plug, was fired 30 deg before the No. 1 plug or at 40 deg btc. As can be seen in the flame photographs, the flame front from the second plug, the upper left corner, reached the indicating or No. 1 plug before it fired, probably reaching the No. 1 plug about 8 deg before it fired. Hence, the sharp upward deflection of the ionization trace before the No. 1 plug fired. The low-amplitude fluctuations in the first part of the trace are due to the spark pickup from the second plug.

Fig. B shows 6 ionization traces in which all engine operating conditions except spark-timing were constant. The ionization indication was again taken from the Chevrolet ignition plug situated in the lower left-hand corner of the photographs, again denoted as the No. 1 plug. The other plug in this case is mounted 3½ in. from the first plug in the center top of the flame photographs. The first plug, the one in the lower left-hand corner, was fired at 10 deg btc in all of the traces shown. The second plug was not fired in the first trace, therefore a case of normal combustion, but was fired at 20 deg after the first plug in the second trace; 10 deg after the first plug in the third; at the same time as

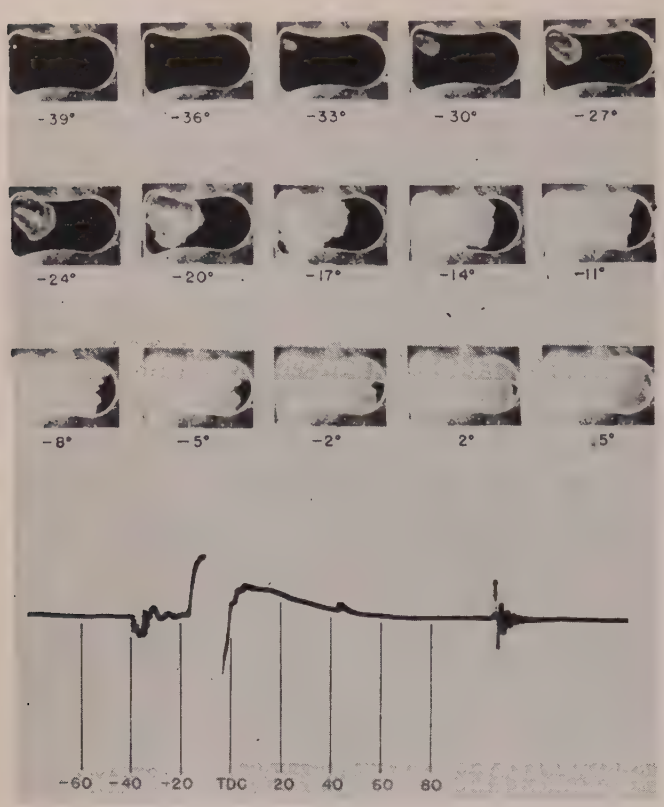


Fig. A – A simulated case of advanced preignition

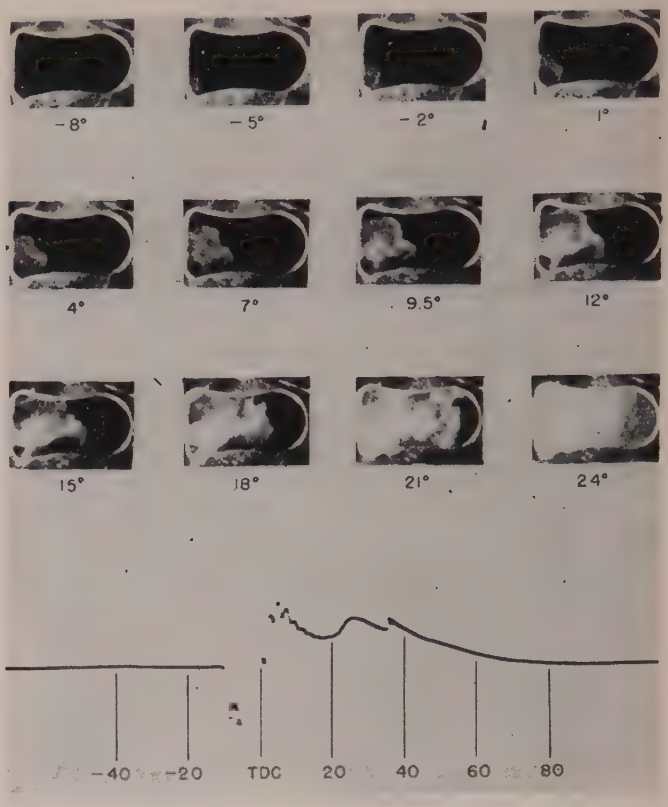


Fig. C – A normal detonating explosion in presence of carbon-lead deposits

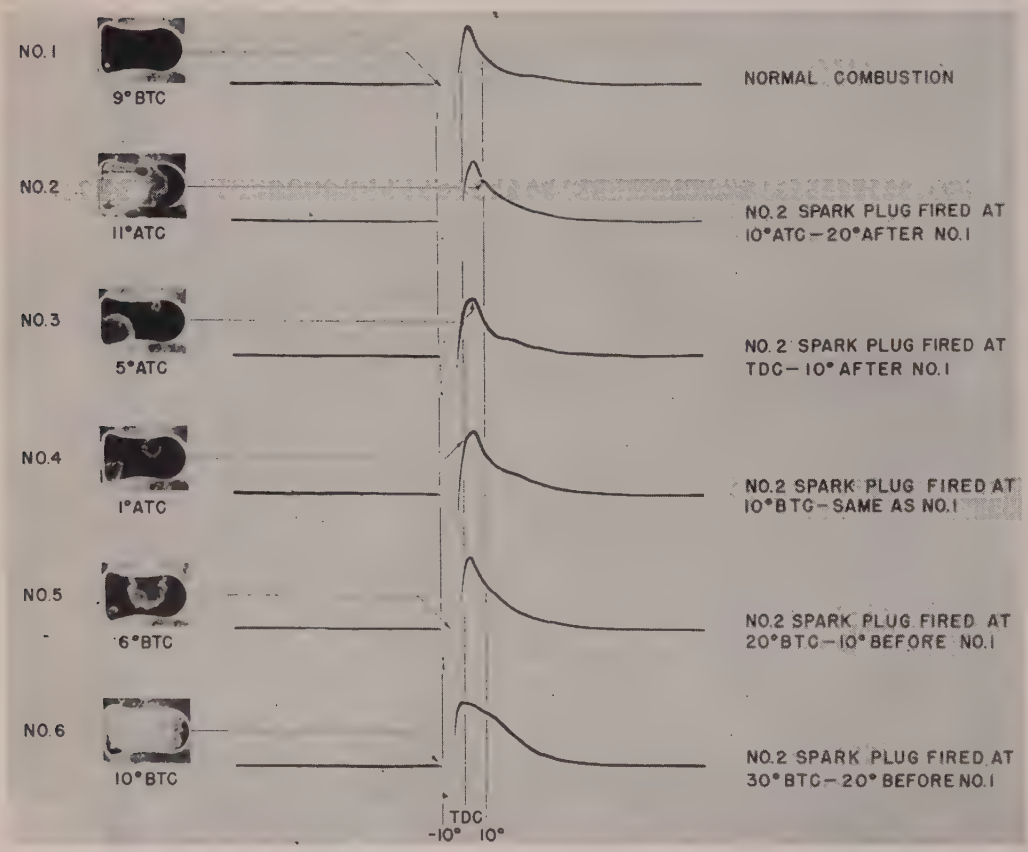


Fig. B – A comparison of ionization traces obtained in presence of various degrees of simulated autoignition

the No. 1 plug in the fourth; 10 deg before the No. 1 plug in the fifth; and 20 deg before in the sixth, the simulated autoignition becoming more severe as we move from top to bottom. Flame photographs were taken as before and a flame photograph showing the 2 flame fronts is shown at the appropriate position on each ionization record.

Even very close observation of the traces show very little difference in the time of occurrence of the ionization peak whether or not a second flame front was initiated at the same time, or after the first or indicating plug fired (the first 4 traces). However, when the second flame front was started before the first or indicating plug fired, but did not reach the indicating plug, the occurrence of the peak of the ionization trace appeared noticeably earlier in the cycle than in the previous cases (the last 2 traces).

I would, therefore, agree with the authors' conclusions if they would alter their definition of preignition from "the initiation of a flame front at any point in the mixture before combustion initiated by the spark plug would normally reach that point" to the following definition: Preignition is the initiation of a flame front at any point in the mixture before the normal spark fires. However, in order to be able to recognize even mild cases of preignition as I have just defined it, some experience with the equipment is necessary.

I would like to ask the authors if these results are in agreement with their data.

One other interesting observation has been made in our study thus far. Fig. C shows this case. This is an ionization record and flame photographs of a combustion process as it occurred in the presence of carbon-lead deposits. As can be seen by the flame photographs, only one spark plug was used, it being fired at 8 to 9 deg btc. It can also be seen that autoignition did not occur. Knock, however, did occur between 21 and 24 deg atc. The difference in the ionization trace between the clean normal combustion case and this case is quite evident, the second deflection of the ionization trace starting at about 22 deg atc coinciding with the occurrence of knock. The initial portion of the diagram is not as well defined as those obtained from the clean engine shown previously. Detonation in the presence of carbon-lead deposits and a change from 100- to 75-octane-number primary reference fuels constitute the only differences in the engine operating conditions from those of the normal combustion traces shown as the first trace on the preceding figure. This seems to indicate that only the first portion of the ionization diagram can be used as an autoignition indicator.

We are planning to continue the investigation of the operation of this method of determining the presence of preignition. So long as the limitations are realized, this method of identifying preignition may be a very decided aid to our present means of identifying autoignition.

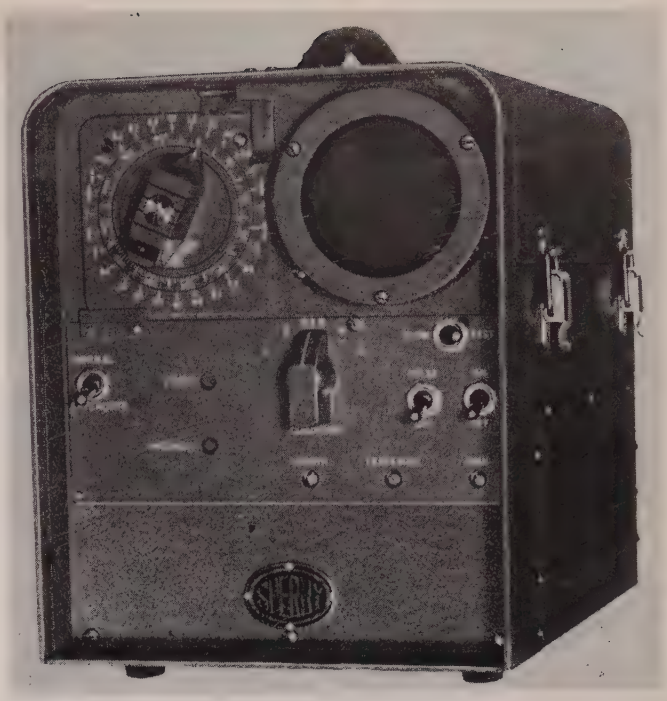


Fig. D – Engine analyzer used for preignition in aircooled engines with magneto ignition

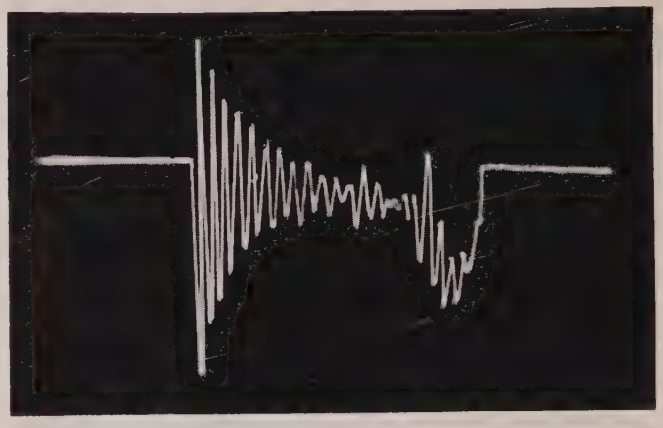


Fig. E – Normal ignition pattern for high-tension magneto ignition

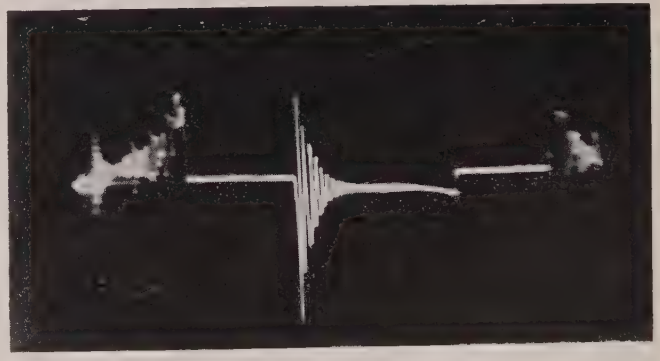


Fig. F – Shorted secondary pattern for high-tension magneto ignition, similar to pattern resulting from severe preignition

Describes Limitations Of Ionization Method

– J. W. Wheeler
Sperry Gyroscope Co.

THE authors have offered an extremely clear presentation of a simple yet effective method for determining preignition in an automotive engine.

The use of ionization methods to detect flame fronts in engine cylinders is well known. It is interesting that one of the most complete discussions of ionization methods the writer was able to find in the literature was a paper by Dr. Kurt Schnauffer, published in 1934.[a] Although adequate knowledge of the method was thus presented to SAE

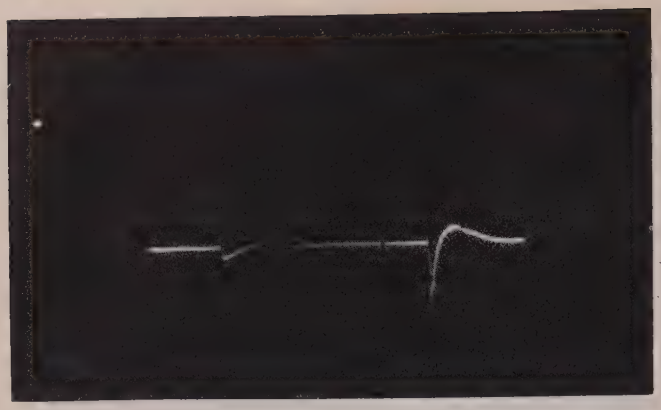


Fig. G – Pattern showing no preignition or autoignition, using Winch-Mayes method, with 120-v battery, and signal applied directly to cathode ray tube vertical deflection plates

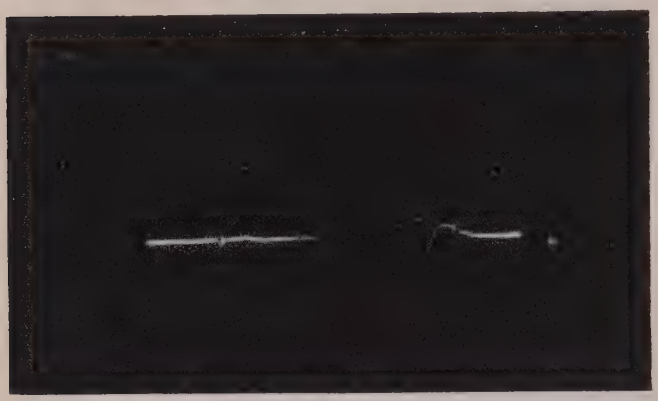


Fig. H – Pattern showing preignition 66 deg crankshaft before normal ignition, using Winch-Mayes method, with 120-v battery, and signal applied directly to cathode ray tube vertical deflection plates

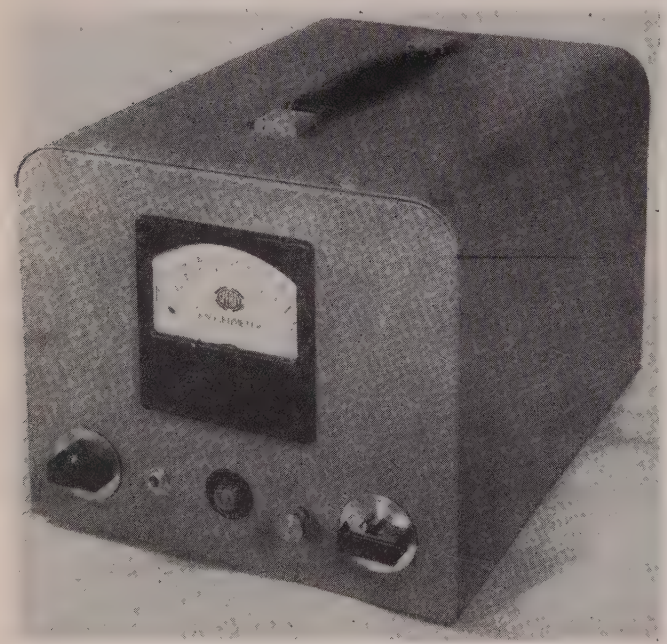


Fig. I – K-3 Knockometer for automotive fuel rating. This meter does not respond to preignition or autoignition, but only to detonation

members nearly 20 years ago, it has remained for Messrs. Winch and Mayes to apply the principles to automotive fuel testing.

Limitations of the ionization method lie chiefly in locating the gap near to the point of flame-front origination. Schnauffer, Hastings, and others solved this problem by using a number of gaps spaced throughout the combustion-chamber area. This is not possible to accomplish in present automobile engines without mechanical modification; and modification changes the engine-fuel characteristics. Using the spark plug as a single gap in the cylinder means that the exact crankshaft angle at which preignition occurs and the point in the cylinder where preignition originates cannot be determined exactly.

The equipment illustrated by Winch and Mayes has additional detailed limitations. First, applying a voltage in excess of their rating to the isolating resistors is highly undesirable from the standpoint of reliability. Second, the 10-megohm isolating resistor in combination with the oscilloscope input cylinder capacitance constitutes a low-pass filter circuit of the single-lag RC type. In this application the filter is used to pass the low-frequency phenomena related to preignition and to attenuate the high-voltage, high-frequency signal from the ignition circuit. Determination of the frequency response desired for optimum separation of preignition phenomena is not presented in the paper. As it is an important part of the method proposed, it would appear desirable to calculate the optimum filter characteristics. It would be interesting to know if this has been done by the authors. Third, the frequency response of the oscilloscope vertical amplifiers affect the results obtained. It would appear desirable to use higher d-c voltages so that it would not be necessary to use any vertical amplifier. Fourth, while the use of a standard oscilloscope synchronized by means of an ignition voltage pickup is possible, it is not nearly as convenient as other methods available, when engine operating conditions involve large variations in rpm.

The Sperry engine analyzer, Fig. D, has been used for the detection of autoignition or preignition. Full details of its use in this manner are discussed in a paper by J. E. Lindberg, Jr.[b] Briefly, the method involved watching for a change from the normal ignition pattern, as in Fig. E, to the shorted secondary pattern, as in Fig. F. The ionization causes an effective reduction in the resistance of the spark-plug gap, and the effect of this reduction in resistance is observed to progress from the latter portion of the ignition pattern to the beginning as autoignition progresses into preignition. Its use is limited primarily to magneto ignition systems, particularly in aircraft operations where modifications to the ignition system may decrease safety.

For our own information in reviewing the method disclosed by the authors, we found it of interest to run a short test in an aircooled single-cylinder engine. The test was made with a modified engine analyzer using a higher d-c ionizing voltage so that no vertical amplifier was required. We found the method to work very satisfactorily, as indicated in Fig. G, for normal combustion, and Fig. H, for preignition occurring 66 deg in advance of normal ignition.

The writer is not an engine expert and cannot tell from audible indications whether a particular knock results from detonation alone or from detonation resulting from preignition. In addition, with a limited knowledge of fuel technology, it is difficult to predict if economic penalties are incurred by the addition of low preignition rating com-

[a] See *SAE Journal* (Transactions), Vol. 34, January, 1934, pp. 17-24: "Engine-Cylinder Flame-Propagation Studied by New Methods," by Dr. Kurt Schnauffer.

[b] See Report of Aircraft Spark Plug and Ignition Conference, 1950. Published by Champion Spark Plug Co., Toledo, O.

ponents in fuel blending. A slight trend toward preignition rating of some aromatics having high octane rating would seem to make it extremely desirable to separate fuel detonation performance from fuel preignition performance, however. A knockometer (Fig. I) constructed for the automotive method of fuel rating and arranged to sense only detonation, was not well received by industry in 1946. This makes it evident that if low preignition rating of fuel components may cause an economic penalty, it was not felt to be a problem at that time. From operation of some of the later model engines, however, it appears that the knock-free performance is determined more by the fuel preignition limit than by its detonation characteristics.

Regardless of economic or marketing considerations, from a purely scientific point of view it is highly desirable that each separate phenomena be separately measured. Winch and Mayes are to be particularly congratulated for having the foresight and ingenuity to provide a simple tool for identifying preignition in automotive vehicles, thus making it possible to run adequate fuel requirement investigations.

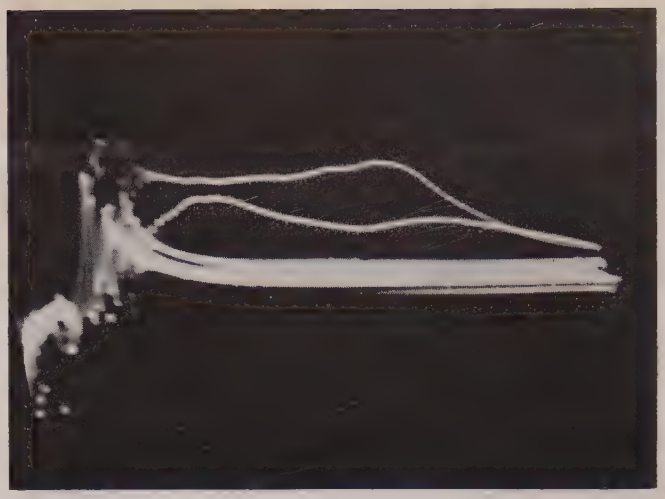


Fig. J – Pattern for two cycles of passenger car operating on low-octane fuel

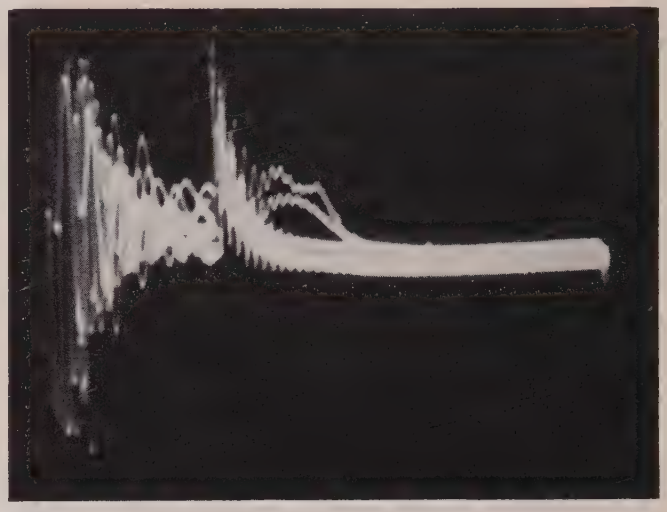


Fig. K – Pattern of car (in Fig. J) operating on commercial fuel and with retarded spark

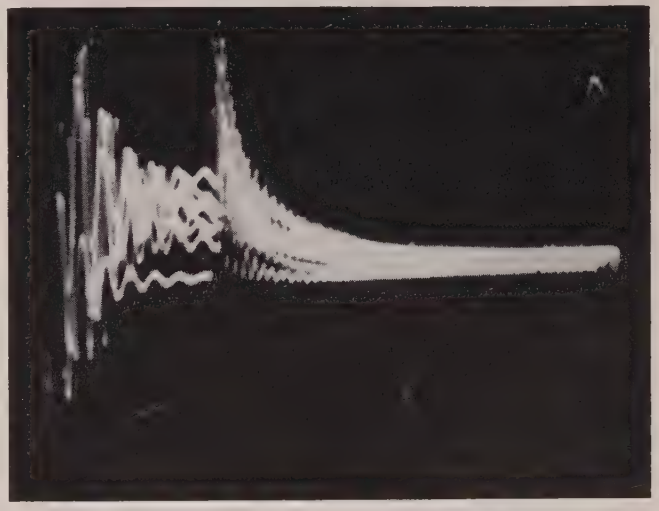


Fig. L – Typical preignition pattern

Theories on Preignition Modified by New Methods

– J. H. Goffe
Socony-Vacuum Oil Co

THE paper presented by these authors is especially valuable because of the wonderful progress it represents in this important field of preignition.

We witnessed a demonstration of the detecting system described in the paper, and as a result of the promise it illustrated, we attempted to duplicate the results. Since an oscilloscope of the same type as used by the authors was not available in our laboratories, a unit of a different type was substituted. The initial tests failed to reproduce the patterns presented in the authors' paper. The 10-megohm resistor failures which were encountered were overcome by a change in brand of resistors and the use of two 5-megohm resistors in series.

To analyze the reasons for our poor results, a study of Fig. 1 of the paper will show that a series circuit was formed consisting of the spark-plug gap, the 10-megohm resistor, the oscilloscope, and battery. Thus, the internal electrical characteristics of the oscilloscope are of prime importance in governing the current flow in the series circuit so established.

It was concluded, therefore, that the characteristics of the oscilloscope selected were responsible for the difficulties encountered, and a new approach was initiated. This involved an attempt to utilize the thermoelectric, or thermionic, voltage generated by flame. An instrument was selected having a high gain, and incorporating a wide band amplifier, and in order to more clearly see the preignition we hoped to observe, the horizontal sweep of the oscilloscope was triggered on each and every ignition pulse. Thus the horizontal length of the picture represented 90 deg of crank angle on the 8-cyl car.

Using such an oscilloscope, it was anticipated that the patterns would include considerable interference from the ignition proper. However, since our primary interest was in that period of crank angle just prior to ignition – a time when the distributor points are closed, and when no secondary voltage is present – this interference was considered to be a relatively minor objection.

From this type of instrumentation, which is really a circuit similar to that of Fig. 7 of the subject paper, re-

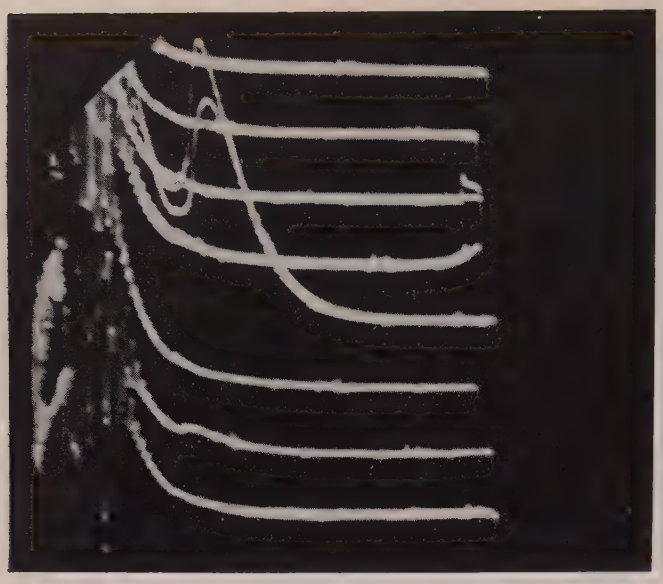


Fig. M – Pulse before ignition and two cycles of afterflame

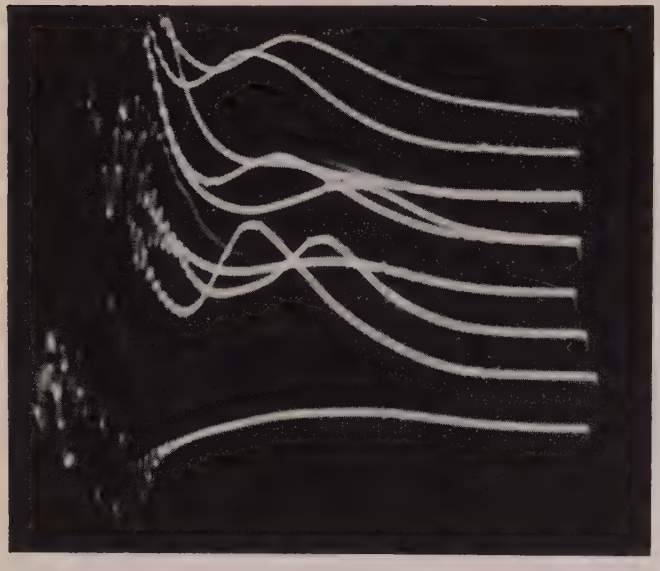


Fig. N – Normal pattern obtained from every cylinder simultaneously

Fig. O – Results without vertical separation of severe preignition induced by severe detonation

sults surpassing initial expectation were achieved, as shown in several typical patterns which follow.

Fig. J shows the pattern for two cycles of a 1952 V-8 passenger car operating on a low-octane-number fuel, resulting in medium knock. The signal was obtained from one cylinder only. The long-lasting afterflame seems typical of such fuels. Fig. K shows a similar condition except with a commercial fuel and a retarded spark. In this figure, the presence of late flames can be clearly seen, yet a definite difference between this and the previous condition is noted. Fig. L shows a typical preignition pattern. By estimating the percentage of cycles that produce such a pulse, the relative frequency of the preignition may be determined, and its advance in timing with respect to the normal electrical ignition can be measured.

To be certain that the pulse observed in Fig. L was actually before ignition, use was made of a special oscilloscope developed by the Socony-Vacuum laboratories, wherein the base line for each cylinder can be separated vertically. Fig. M shows the pulse to occur before ignition (right-hand end of fourth line) and two cycles of afterflame (fifth line).

This system was then tried on the same vehicle with all eight of the 10-megohm resistors connected together to obtain information from every cylinder simultaneously. The normal pattern for this condition is shown in Fig. N. Wide differences in normal flames can be readily noticed. Incidentally, the bottom trace on this pattern shows the effect of a fouled plug. Fig. O shows the results without vertical separation of severe preignition induced by severe detonation.

The work carried out by our laboratories thus far has demonstrated that there is much to be learned in this relatively new instrumentation field, and many previous theories on preignition may have to be modified under the scrutiny of these modern methods of detection. For those interested in the preignition problem, we commend these new analytical methods, as offering hope of more rapid advances in this field.

Instrumentation Detects Abnormal Combustion

– R. Meagher
E. I. du Pont de Nemours & Co.

THE method for identifying preignition described by Winch and Mayes should prove to be an extremely valuable tool for working on the preignition problem. We are interested in this technique at the du Pont petroleum laboratory and have conducted a preliminary evaluation to determine its suitability for detecting preignition. Our evaluation involved single-cylinder engine studies of oscilloscope traces showing the spark-plug ionization pattern and the rate of change of cylinder pressure observed simultaneously on the screen of a dual-beam oscilloscope. The time rate of change of pressure, or dp/dt, trace was used, since it is very sensitive to changes in the combustion process and affords a clear-cut and unmistakable identification of preignition. Photographic records of the ionization pattern and the dp/dt trace clearly demonstrated the ability of the ionization instrumentation to detect preignition although there was a time lag between the detection of preignition by the dp/dt trace and its manifestation on the ionization trace.

Our investigation was conducted in two CFR engines, one a standard research method engine, and one having an L-head with 7.0/1 compression ratio. The standard ignition systems of these engines were each replaced by a 6-v battery ignition system.

The effect of changes in compression ratio on the ionization pattern was evaluated in the standard valve-in-head engine. Fig. P shows these results obtained during normal combustion using isooctane containing 3.0 ml tel per gal. The main effect of increasing the compression ratio from 7 to 10/1 was to increase the size of the ionization pattern, particularly the peak height of the trace. These changes do not cause difficulty in interpreting the patterns, but it is important to recognize that the normal combustion ionization pattern can be affected by the engine compression ratio. The data shown in Fig. P were obtained at a constant air/fuel ratio of approximately 14/1. It has been observed that the ionization method of preignition detection works best at nearly normal air/fuel ratios. Slight departures from normal air/fuel ratios which might be encountered in ordinary operation of engines do not affect the pattern significantly, but radical departures from normal values may make preignition detection quite difficult with this instrumentation.

Changes in fuel type and in tel concentration also affect the appearance of the ionization pattern during normal combustion. Fig. Q shows the ionization pattern with isooctane and the change produced when 3.0 ml tel is added to this fuel. These data were obtained in the L-head engine and the patterns represent normal combustion with no preignition. The addition of tel to isooctane resulted in a broader pattern having a somewhat higher peak. Changing the fuel type from isooctane to diisobutylene increased the height and breadth of the trace. These changes also do not introduce difficulty in the interpretation of the patterns but should be recognized, particularly in rating engines with various reference fuel systems. The general appearance of the patterns in Fig. Q is not the same as that in Fig. P. This can be accounted for by two factors. First, the shape of the combustion chamber of the two engines was not the same, the patterns in Fig. P being obtained in the overhead-valve engine and those in Fig. Q in the L-head engine. Second, there was a difference in spark timing. In the course of studies with this instrumentation it has been observed that changes in engine type produce substantial changes in the ionization trace; furthermore, there can be slight differences in the pattern between engines presumed to be identical. Since fuel and engine variables affect the ionization pattern, training and experience are necessary to obtain accurate results.

In the L-head engine, combustion was observed by means of a rate of pressure change pickup in the cylinder head as well as by the ionization instrumentation. Fig. R shows a typical simultaneous record of the two traces on the screen of a dual-beam oscilloscope. This is an actual photograph of the oscilloscope traces obtained during deposit-induced preignition. The upper part of the figure shows the ionization pattern during three successive cycles with normal combustion and with preignition; the lower portion gives the simultaneous dp/dt traces. There is some distortion produced by electrical interference and overlapping of the traces caused by the size of the oscilloscope screen. To separate these effects and to distinguish between the traces more clearly, Fig. S has been drawn from this photograph.

The information in Fig. S was obtained using an equal volume mixture of benzene and methyl alcohol as fuel. A large number of photographs were taken at several engine operating conditions, and Fig. S was selected from the group since it showed a normal combustion cycle together with two cycles in which preignition was caused by the presence of combustion-chamber deposits. The solid lines show a normal combustion cycle in which some knocking is evident and in which spark ignition occurred 5 deg atdc. The cycle indicated by the dashed line shows preignition occurring approximately 14 deg before spark on the dp/dt trace; however, it will be noted that there is no indication of preignition on the ionization trace before spark ignition. There is, however, an appreciable increase in the height of the ionization pattern after the passage of the spark, and as Mr. Winch has pointed out, this is a manifestation of preignition. The dotted line shows a cycle in which more advanced preignition occurred approximately 22 deg before spark ignition according to the dp/dt trace, but was not indicated until 12 deg before spark ignition on the ioniza-

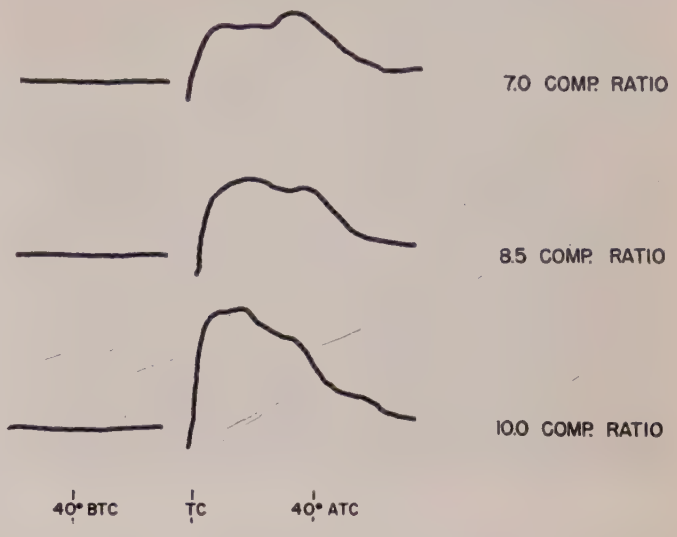


Fig. P – Effect of compression ratio on ionization pattern produced during normal combustion of isooctane containing 3.0 ml tel per gal in CFR valve-in-head engine

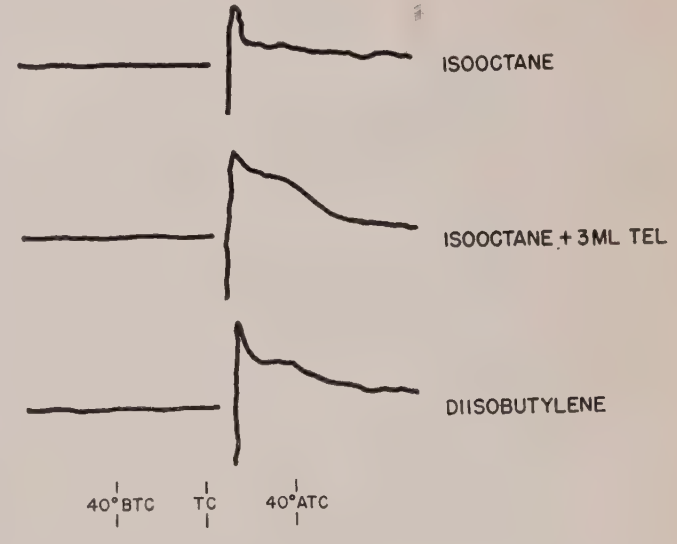


Fig. Q – Effect of fuel type and tel on ionization pattern produced during normal combustion in CFR 7.0/1 compression ratio L-head engine

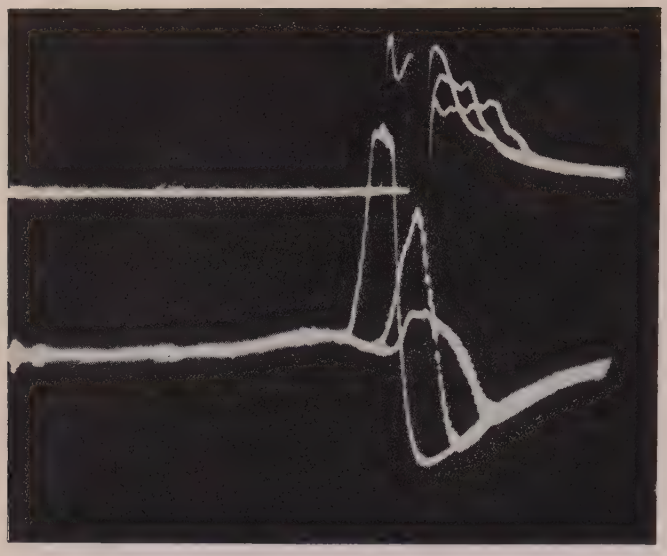


Fig. R – Actual photograph of simultaneous ionization patterns and dp/dt traces of normal combustion and deposit-induced preignition. CFR 7.0/1 compression ratio L-head engine operated on equal volume mixture of benzene and methyl alcohol

tion pattern. There is usually a time lag between the indication of preignition on the rate of pressure change trace and the ionization trace as illustrated by the fact that the ionization pattern did not show preignition until 10 deg after its indication on the dp/dt trace. This differential is the time required for the flame to proceed from its point of initiation in the combustion chamber to the ionization gap at the spark plug.

As a result of the comparison between the simultaneous records showing the ionization pattern changes and the rate of pressure change patterns during deposit-induced preignition, it is quite evident that this instrumentation can detect abnormal combustion in the engine cylinder. There may be additional problems in adapting this instrumentation to multicylinder engines and to cars on the road, but it appears likely that this method will be a practical and economical means for the detection of preignition.

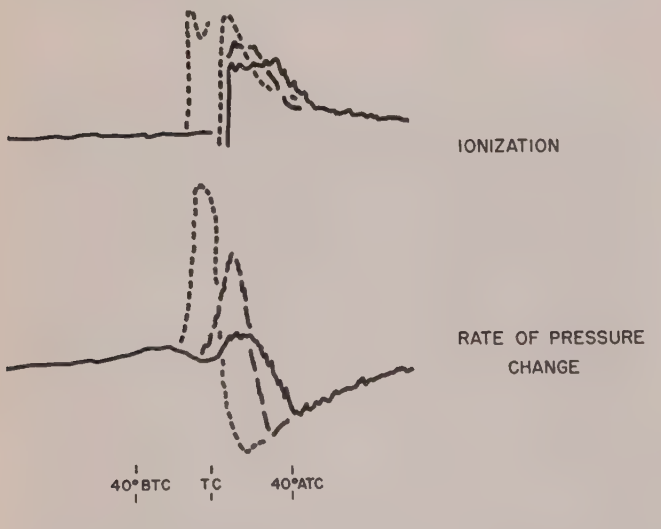


Fig. S – Reproductions of traces shown in Fig. R

Authors' Closure To Discussion

MESSRS. Bowditch, Wheeler, and Goffe pointed out that the type of 10-megohm, 1-w resistor was a critical component in the test circuit. We agree that this is the case, although it was discovered by us only a short time before this paper was presented. Therefore, we have not been able to determine what resistor qualities are essential for satisfactory performance. We have tried two makes of resistors – one works fine – one does not. The type of oscilloscope has also been mentioned as a critical item. Actually, we have had no difficulty either with the make of resistor or with the make of oscilloscope originally selected for our test circuit. Again, we have tried two different makes of oscilloscopes and found both to perform satisfactorily; therefore, we spent no time trying to identify essential oscilloscope characteristics. Since we are not interested in manufacturing instrumentation and had a technique which was satisfactory for our work, development was stopped before all limiting specifications could be determined for the equipment.

Our definition of preignition has been questioned, and as was pointed out in the paper, is a definition designed to include all types of secondary flame fronts under one name, for the purpose of this paper. It is not suggested as a definition for the industry to adopt.

Mr. Bowditch points out that the technique will not indicate flame fronts occurring after the spark has occurred. We have found that to be true, and agree that the limit of sensitivity is probably where normal and secondary flames occur simultaneously with the technique and procedure described in this paper. Present-day knock rating techniques using the human ear as the rating tool accept a similar limit of sensitivity. Just because knock cannot be heard does not mean it is not present. And because all the knock cannot be heard does not mean the technique cannot be useful.

We agree with Mr. Diggs that the ionization traces on the oscilloscope are affected by changes in compression ratio, spark advance, engine speed, the type of fuel used, and fuel/air ratio. We do not consider this to be particularly serious, but rather something that must be kept in mind if the procedure is to be applied satisfactorily and the results interpreted properly. For each condition, there is a normal pattern; and for each condition, the normal pattern must be observed so that changes resulting from preignition can be observed.

Mr. Wheeler and Mr. Goffe have both pointed out that the synchronizing technique has limitations which are recognized. This system was used as opposed to more positive methods, since there was no mechanical linkage with the engine required, as is necessary with the use of potentiometers or signal generators.

Because the sensitivity of the test technique depends upon so many factors, additional experience and exchange of ideas by different laboratories should result in improved sensitivity and even more advantageous applications of the equipment and procedure.

New Test for Drawability of Sheet Steel

C. B. Buker and J. R. Speer, Jones & Laughlin Steel Corp.

This paper was presented at the SAE Annual Meeting, Detroit, Jan. 12, 1953.

THE two most desired characteristics of sheet and strip steels are good drawability and freedom from stretcher strains. A considerable amount of valuable information concerning these qualities has been accumulated by the analysis of regular and special test data.

The usual means for estimating drawability in a broad sense is by hardness and cup ductility tests. Evaluation of stretcher straining has required a tensile specimen from which yield behavior is studied.[1] Such procedures leave a great deal to be desired in production line testing where volume, convenience, and speed are most essential. They are generally time-consuming, destructive in character, and extremely localized as to test result. Due to aging, the test result is often not indicative of

[1] See "Mechanical Testing of Sheet Metals," by John R. Low, Jr., pp. 128-131, Metals Handbook. Published by American Society for Metals, Cleveland, Ohio, 1948.

A MECHANICAL corner-bend test is presented in this paper as a quantitative measure of drawing quality in sheet and strip steel.

The load required to bend the corner, when corrected for sheet thickness, is representative of ductility, and is associated with the yield point and, to some degree, with the Rockwell hardness of the material. The curvature of the corner bend is related to yield elongation, and can be used as a measure of stretcher-strain susceptibility.

In stamping plants this flex-test has filled the need for a simple nondestructive test immediately applicable at the operation. It requires no specimen preparation and provides evaluations that can be supported by actual press performance of the steel sheet.

The usefulness of the test has been extended to laboratory studies of the mechanics of strain aging in strip steels.

C. B. BUKER (M '52) is a metallurgical engineer on the general technical staff of Jones & Laughlin Steel Corp. He attended Western Reserve University prior to serving for several years in a supervisory capacity with Fisher Body Division of GMC. He joined the Jones & Laughlin organization in 1937, where his activities have been primarily concerned with engineering in the field of flat rolled products.

J. R. SPEER is in charge of physical testing for the metallurgical research division of Jones & Laughlin Steel Corp. He is a graduate of Princeton University. He has been associated with his present organization since 1933, and has held a number of positions in the metallurgical and research departments.

Fig. 1 – With this instrument, the Flex-Tester, a corner of a metal sheet is bent through a given arc

the quality of the material at the time of its use.

Over the years it has been common practice for press and die men to bend the corner of a sheet to "feel" its resistance. Usually the greater the effort required to bend the corner, the less its possibility of success in extra deep drawing. Some operators became very adept at predicting press performance by such means. However, the method is at best only roughly qualitative, convenience being its real asset.

H. H. Stanley[2] has stated, "The radius of the bend, under standard conditions, should be closely related to the yield-point extension, and this radius might form a rapid means of measuring this value." It is well known that the tendency to stretcher strain is directly proportional to the amount of yield-point elongation observed during a tension test.[1] Thus, curvature of bend should be a measure of the tendency to stretcher strain.

These principles form the basis of a test developed in the Technical and Research Divisions of Jones & Laughlin Steel Corp. Instruments have been devised to accurately measure the resistance of a material to a given bending force, and to determine the degree of curvature of the bend produced by that force. This paper discusses this test as a quantitative measure of a material's probable success or failure in drawing.

Test Instruments

The prime instrument, called a Flex-Tester, is illustrated in Fig. 1. With this instrument the corner of a sheet is bent through a given arc. The resistance of the material to the bending force causes a tongue portion, acting as a cantilever spring, to deflect. This deflection is recorded on a dial gage in what is arbitrarily termed *F* units.

In order to obtain comparative *F* values, a correction for sheet thickness is required. An investigation of the mechanics of the bend indicated that this correction should be in proportion to the square of the thickness. Such correction was found to be satisfactory empirically. As 0.035-in. thickness is in most common use, this was made the standard for which no correction was required, thus,

$$F \text{ value} = \left(\frac{0.035 \text{ in.}}{\text{Actual thickness}}\right)^2 \times \left(F \text{ units from dial}\right)$$

The conversion to *F* values is readily made using a table of values, a slide rule or a nomogram, rather than by arithmetic substitution in the above formula.

It should be noted here that stamping performance can be affected by certain mechanics of material thickness which may involve die clearance. This is not a reflection of the material's inherent drawing quality as indicated by *F* values.

The maximum curvature of the flexed corner is measured by a specially designed spherometer shown in Fig. 2. The highest dial unit obtained is the deflection of the material in 0.001 in. across the span of the two outside instrument points. The dial units have been arbitrarily termed *R* values.

Preliminary Evaluation of Test

It was thought that quantitative values applied to the corner flex-test would greatly increase its usefulness and scope of application. Can varying degrees of resistance to bend determined instrumentally be correlated with stamping performance? Will the flex-test show changes due to strain aging and how will it compare with existing tests? These are some of the questions which arose in assessing the flex-test. Thus, much of the prelimi-

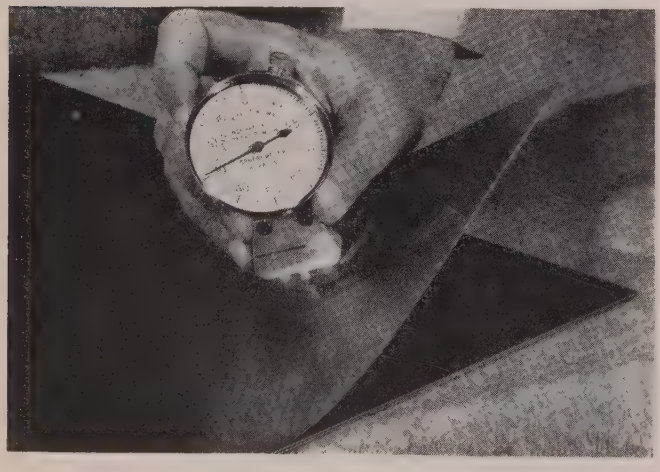


Fig. 2 – Spherometer measures maximum curvature of flexed corner shown in Fig. 1

[2] See Proceedings, Institution of Automobile Engineers, Vol. 29, 1934-1935, pp. 616-620: Discussion by H. H. Stanley of "Cold Pressing and Drawing."

nary work consisted of collecting data on material of known history by present test and performance results, and comparing these results, using the new test.

At the outset it was thought advisable to determine what changes in test determinations would be brought about by strain aging. Skin rolled steels, all 0.035 in. thick, were available for investigation. Fig. 3 shows typical results obtained. The rimmed steels increased in both *F* and *R* values with aging, the rate of change being at first rapid, then diminishing with longer aging times. The aluminum-killed steels, on the other hand, changed very little. These results indicated that the aging characteristics of strip steels could be determined by the changes in flex-test and spherometer values with time.

No performance results were available on these steels to afford any correlation with *F* values. Tension tests on the aluminum-killed group showed no return of the yield, while the rimmed steel showed increasing yield elongation with aging, thereby supporting the spherometer method. Skin rolled nonaging steels, which are usually not troublesome with respect to stretcher straining, has spherometer or *R* values of 30 or less. Rimmed steels which may be assumed to be free of stretcher straining immediately after tempering had *R* values of approximately 30. It seemed reasonable to assume then that a steel with an *R* value of 30 or less will not show stretcher strains if fabricated soon after testing.

Next, samples of four satisfactory and four unsatisfactory steels from an average deep-drawn, exterior auto-body stamping operation were obtained for tests. The unsatisfactory steels gave *F* values of 60 or higher while the satisfactory steels were 60 or under. Spherometer tests were of little value due to an unknown time lapse between procurement and tests. *R* values were in the neighborhood of 45, or well above the 30 which may be assumed to be required for exposed parts where stretcher strains would be a hazard. The apparent breakoff point of 60 *F*, dividing satisfactory and unsatisfactory steel for this particular part, pointed to a possibility of establishing a maximum *F* value to satisfy other parts.

The investigation was extended to include a variety of steels of known performance history, attempting some correlation between flex-test results and die performance. Many variables were present, such as severity of stamping, strain aging between skin rolling, stamping, and test time, and the yet unknown effect of roller leveling on certain samples. Also, it was found that the rimmed steels in particular gave somewhat higher load readings at the center than at the edge. (From much accumulated succeeding data it was found that for most material at low *F* values the difference was slight, 0 to 3 points. In a few instances of higher loads the variation was as much as 6 to 7 points.) Despite these variables, the trend of results again

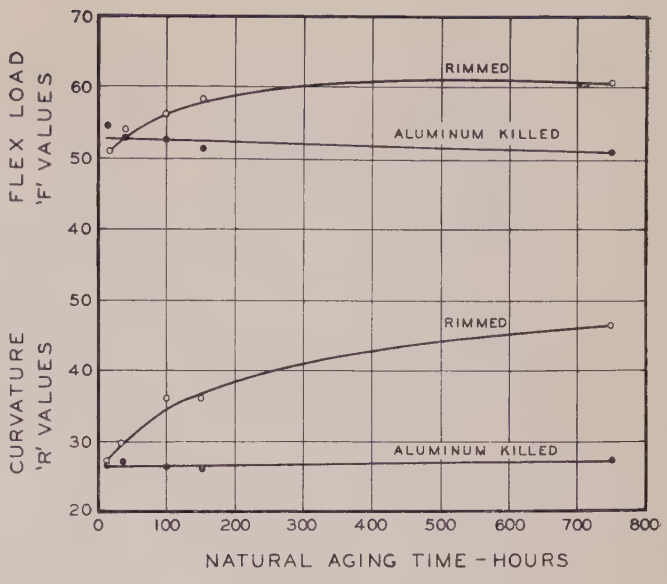


Fig. 3 – Flex-test results showing effect of strain aging

indicated an approximate dividing line in *F* values between satisfactory and unsatisfactory steels. This dividing line seemed to vary with severity of the stamping operation. *R* values were of no importance because of unknown changes due to aging between the stamping operation and testing. Certain samples with *R* values of 30 or less, regardless of time, were found to be stabilized steels. Samples from some parts were untempered and, whether aging or nonaging, gave *R* values greater than 60.

Study of Strain-Aging Behavior

The test to this point appeared to be a reasonably quantitative measure of stamping quality at the time of test and it was thought advisable to put it to investigational use. The effect of small amounts of cold work and aging on yield behavior was studied. As skin rolling and roller leveling are the mechanical processes affecting strain aging in strip steels, they became the subjects for investigation.

Skin Rolling – For the study of skin rolling a 93-in. temper mill was used. Material rolled consisted of 0.035-in. thick, box-annealed sheets from an aluminum-killed and a rimmed steel. Sufficient material from each was passed at various degrees of temper in a range from 0% to 4% for ample tests at intervals up to 2000 hours natural aging time.

Fig. 4 shows the effect of degree of temper, as well as aging, for a rimmed steel. The upper group of curves indicates the progressively higher movement of the *F* value with both age and degree of temper. Using the approximation of *F* 60 maximum as satisfactory deep-drawing material, any temper up to nearly 2% would be satisfactory if stamped within a few hours after temper rolling.

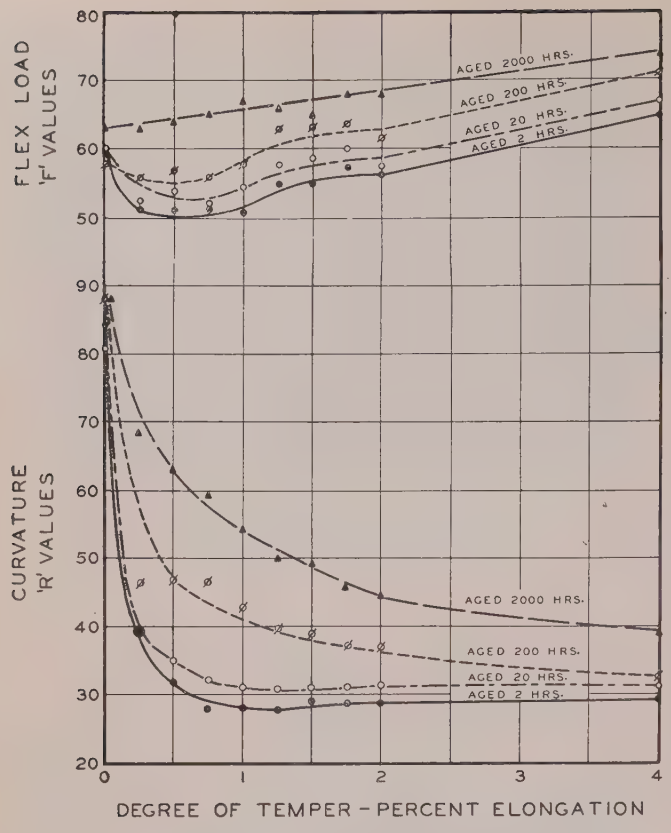


Fig. 4 – Combined effect of aging and temper – rimmed steel

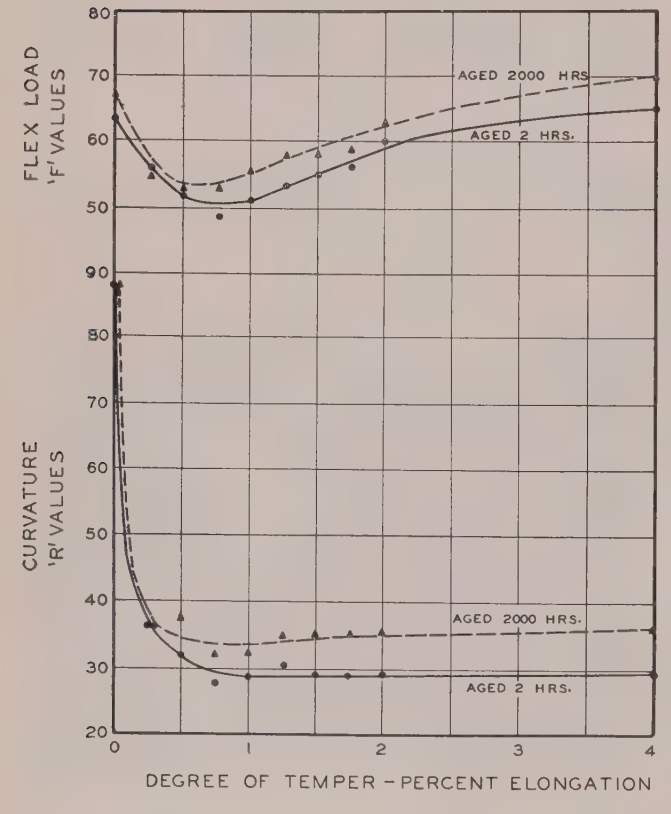


Fig. 5 – Combined effect of aging and temper – aluminum-killed steel

With longer aging the degree of temper producing F 60 maximum was less until, at 2000 hours, the material was unsatisfactory (above 60) throughout the range. For shorter aging times a temper under 1% – in the range of usual practice – seemed best. The lower group of curves shows that a satisfactory curvature value approaching 30 R was lost within 20 hr. It should be noted that an increase in temper was of benefit in retarding the increase in R values, but was accompanied by less desirable F values.

Fig. 5 shows the effect of degree of temper, as well as aging, for an aluminum-killed steel. The spread between 2 and 2000 hours, as expected, was not great, and so the intermediate curves are not shown. The F values were lowest in the vicinity of ¾% temper, as indicated in the upper curves of the figure. The lower curves indicate that at least ¾% temper was required for minimum R values. Further increase of temper had little or no effect on the curvature of the bend. Thus, ¾% temper would be best for this particular steel considering both load and bend diameter. Even though stabilized, the killed steel showed values above 30 R with long aging time.

Roller Leveling – Due to the time interval between skin rolling of strip steel and its use, a fabricator generally resorts to roller leveling for the relief of stretcher strains induced and aggravated by strain aging. In order to investigate this behavior a Budd-McKay sheet processor in a large stamping plant was placed at our disposal.

Table 1 shows the effect of roller leveling on the two previously investigated materials tempered ¾%. For the rimmed steel the flex load was little affected by leveling. For the killed steel the load was increased somewhat. Since roller leveling is performed to relieve stretcher straining, curvature is the more important measurement, particularly for the rimmed steel. With 11 days and also 4 weeks elapsed time between skin rolling and roller leveling, leveling restored the curvature to values approaching 30 R. At 10 weeks R 59 was reduced to only 35, indicating that this material was apparently beyond restoration except perhaps by a double pass. This reduction in R values, however, was not permanent, and an increase was found in tests made several hours after leveling. In the case of the aluminum-killed steel no benefit came from leveling since the curvature was satisfactory before the operation.

Attempts to eliminate stretcher straining in box-annealed untempered steels by roller leveling have never been completely successful. Experiments with such leveling produced only 34 R using pressures much greater than general practice. The increased pressure also heavily distorted the material. It is conceivable that a yield elongation in box-annealed material would be sufficiently short to be eliminated by severe leveling. However, such a short yield is probably associated with a coarse

grain size which would, when stamped, result in an "orange peel" surface.

Correlation with Other Tests

Evaluation of the flex-test would not be complete without correlations with properties determined from the usual tensile, hardness, and Olsen cup tests. A comparison with these tests should give some insight to the qualities measured by the *F* and *R* values in terms of established fundamental properties.

Hardness – Using test results from the foregoing investigations, Rockwell B hardness was plotted against *F* values as shown in Fig. 6. While an exact relationship does not exist, there is, nevertheless, a trend indicated.

Tensile Properties – Fig. 7 shows the relationship between *F* values and tensile behavior. A straight-line relationship appears between yield point and *F* value. Some of the spread in results is probably due to the variant nature of the yield point itself as well as differences in test location.

A correlation also seems to exist between tensile strength and *F* values above approximately 65. The *F* values continue to decrease below this level while the tensile strength remains at approximately 43,000 psi.

A very close correlation was obtained between *R* values and yield elongation as shown in Fig. 8. The dearth of points between 0 and 0.030-in. yield elongation was due to the change in this property caused by aging within the time required to make tensile specimens. There were, of course, many more points than shown, especially at 0 elongation for aluminum-killed steels which did not age.

It is of interest that the relationship between yield elongation and curvature expressed in inches of bend diameter, rather than *R* values, is a straight line as shown in Fig. 9.

A comparison between *F* values and Olsen cup depths was attempted. The scatter points permitted only a very general relationship between the two. The poor correlation was probably due to the fact that the Olsen test provides no correction for the effect of material thickness.

Experiments in Fabricating Plants

Of utmost importance in the development of this test method was the actual experience obtained at the scene of operations. The GMC laboratories and plants were cooperative in this phase of the work and a great deal of information has become available.

Fig. 10 shows data developed by one GMC laboratory. Using as a standard of drawability the per cent uniform elongation as well as the yield point, ranges were established for 122 samples of various steels as to best drawing quality, borderline, and poor drawing quality, as indicated in the lower chart in Fig. 10. Uniform elongation is here considered as the total elongation to the point where the load drops off after reaching the ultimate strength. Steels were classed as having best drawing quality when uniform elongation was 28% or over with a yield of 32,000 psi or less. A uniform elongation less than 26½% or a yield point over 34,000 psi rated the material as having poor drawing quality. In between these specifications is a range of borderline drawing quality. *F* values were determined for each of the samples, and fell as shown in the bar graph of Fig. 11. Using arbitrary ranges of *F* values as shown, these values seem to be as reasonably dependable in evaluating the materials as the more involved and destructive tensile determinations. In no case did a steel of best drawing quality have an *F* value over 64, and

Table 1 – Effect of Roller Leveling – Material Tempered ¾%

Elapsed Time Between Skin Rolling and Leveling		Immediately Before Leveling	After Leveling			
			2½ Hr	6 Hr	24 Hr	48 Hr
		Rimmed Steel				
Flex load (F values)	11 days	57	57	—	—	—
	4 weeks	59	60	61	61	63
	10 weeks	61	61	60	59	58
Curvature (R values)	11 days	46.5	30	—	—	—
	4 weeks	50.5	30	32.5	30.5	32
	10 weeks	59	35	36	35.5	36
		Aluminum-Killed Steel				
Flex load (F values)	11 days	49	56	—	—	—
	4 weeks	49.5	58	59	55	56
	10 weeks	50.5	56	55	58	57
Curvature (R values)	11 days	28	30	—	—	—
	4 weeks	29	30	31.5	30	31
	10 weeks	30	30	31	30.5	30.5

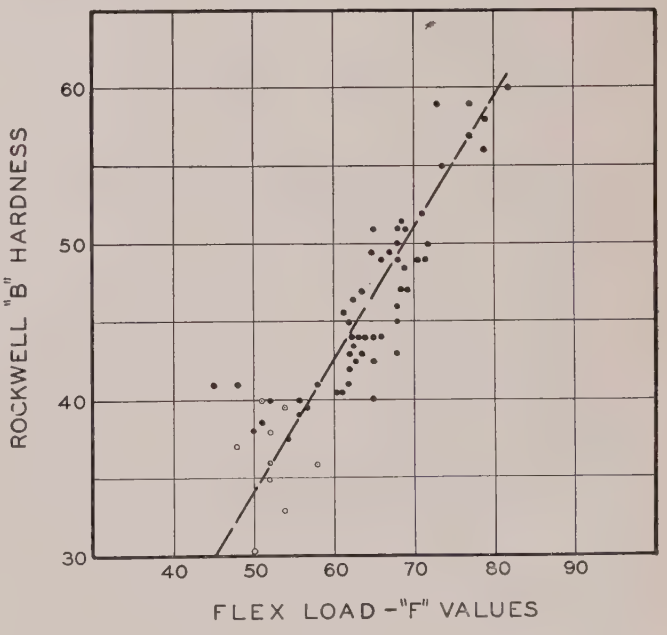


Fig. 6 – Relationship between hardness and flex load

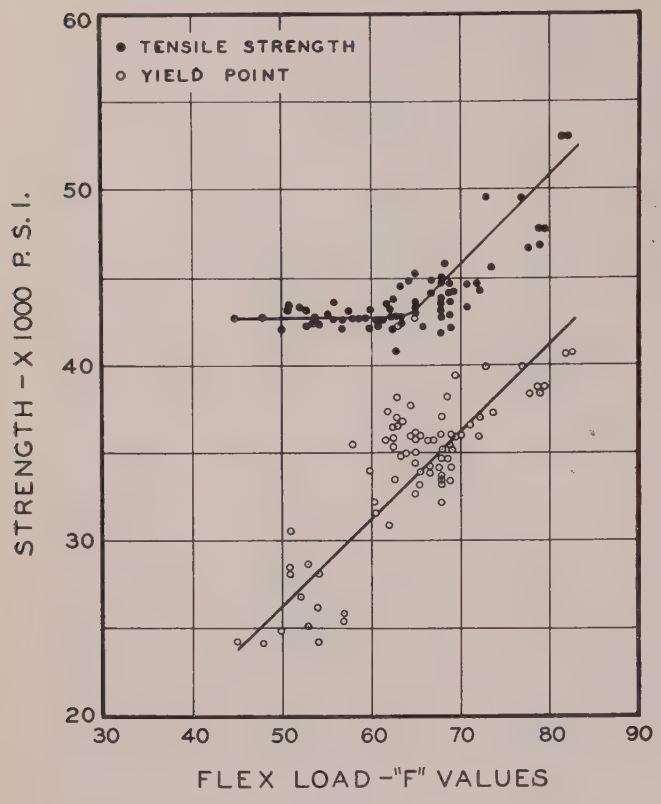


Fig. 7 – Relationship between tensile properties and flex load

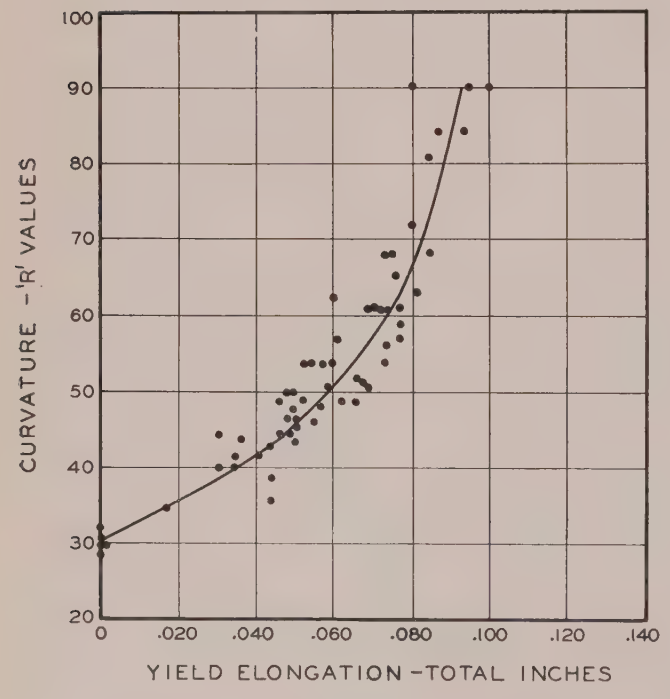


Fig. 8 – Relationship between yield elongation and curvature expressed in R values

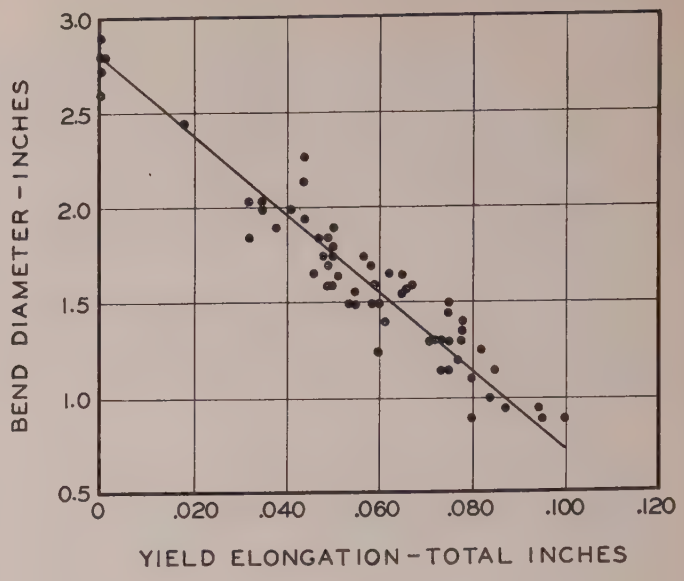
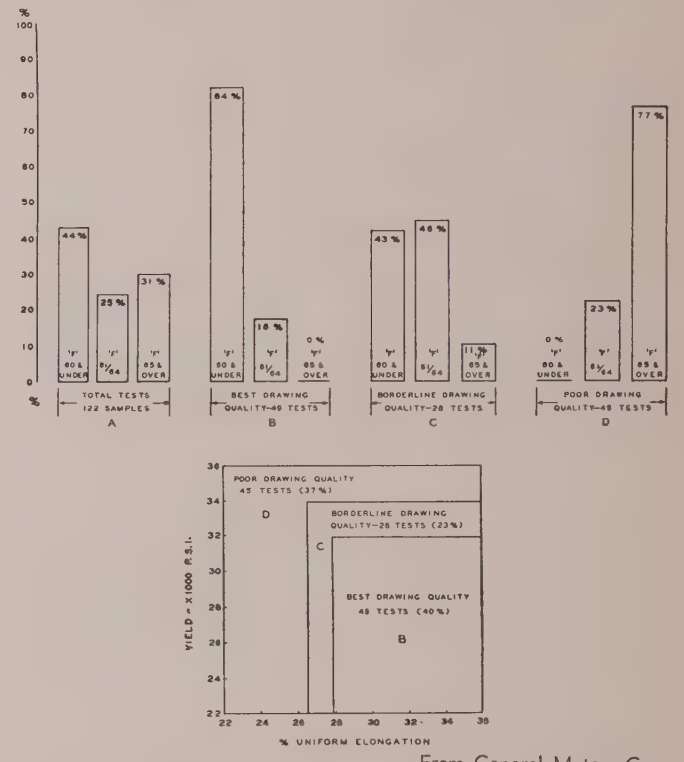


Fig. 9 – Relationship between yield elongation and curvature expressed as bend diameter



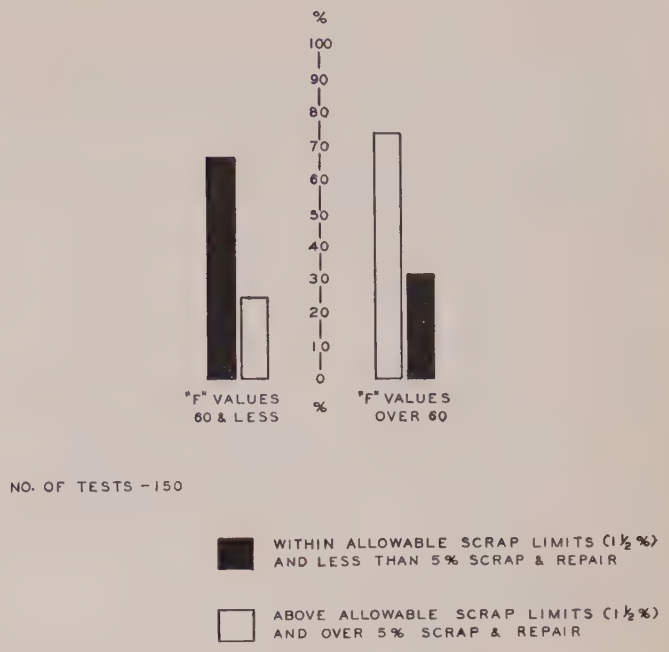
From General Motors Corp.

Fig. 10 – Percentage correlation – drawing quality expressed in terms of combined yield and per cent uniform elongation compared with F values

in no case did a steel of poor drawing quality have an F value under 61.

Fig. 11, a bar graph from Fisher Body laboratory, is shown as a matter of interest. Tests for F values were made on 150 different samples from numerous parts in various stamping divisions. The performance of quantities of material was known but not classified; thus, performance results as shown include all defects causing scrap and repairs such as surface, operator's fault, and so forth. Even so, 68% of the material, which ran within 1½% scrap limits and less than 5% total scrap and repairs, was less than F 60. With elimination of faults other than drawing quality the ratio would, no doubt, be considerably greater.

Fig. 12 shows early tests at stamping plants made on different types of draws. Each represents the range of tests obtained during one day's die performance. Again an approximate dividing line at F 60 was found. The rear quarter-panels, being the most severe draw, produced a higher percentage of scrap and repairs and required many die adjustments when tests went much beyond F 60. The door panel and one trunk lid were mild draws so that steels approaching F 70 gave fairly good results although better stampings were obtained and fewer die adjustments were necessary with material around F 60 or less.



From General Motors Corp.

Fig. 11 – Percentage correlation – press performance compared with F values

Conclusions and Recommendations

Enough experience has been gained using the Flex-Tester to warrant certain conclusions as to its value and recommendations as to its use.

The advantages of the test are many. Requiring no specimen preparation it can be applied to the raw material. It is simple, convenient, and nondestructive. R values provide the mills and stamping plants with the first production test to determine stretcher-strain characteristics.

Flex values will indicate most suitable stock for direction to a given application. This is particularly advantageous where stock sizes are used for more than one part.

Estimates of quality can be made at receiving time at a stamping plant. A consideration of aging changes then permits the immediate use of borderline material and storing of better material for later use.

Flex values can be used to show the changes in rate and degree of aging in strip steels. For example, materials with low flex values a few weeks subsequent to mill tempering are probably of nonaging type.

R tests will show whether a cold reduced material can be directly applied to smooth surface parts, or whether a roller level pass is required prior to stamping. The jump-roll of the sheet processor should be used only when R values indicate it to be necessary.

During various series of tests in stamping plants it was found that results with deep-drawn parts could be improved by delivery to operations of materials having F values within a reasonably close range. Attempting to stamp material of scattered F values caused delays for die adjustments with unnecessary scrap and salvage. Material for die spotting should have an F value in the range required for production purposes. Dies spotted

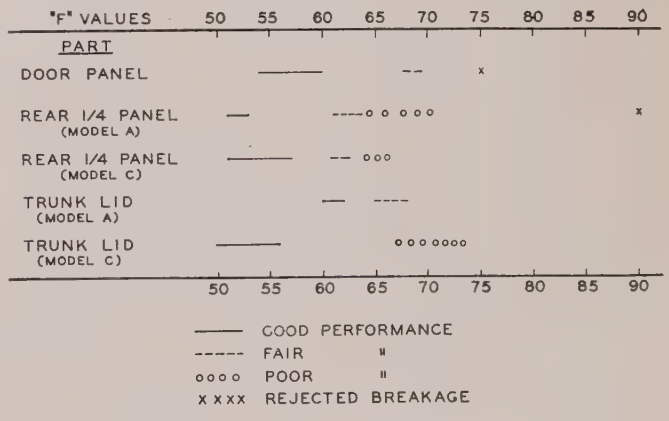


Fig. 12 – Performance ranges from stamping operations

with exceptionally low F values may not be suitable for production materials, while excessive die-spotting time could be eliminated by selecting steels with F and R values in the desired bracket.

The future for the test appears very promising in both strip mills and stamping plants, and should provide a tool for research investigation as well as a welcome addition to control methods.

Acknowledgments

The authors gratefully acknowledge the support and help of their associates within Jones & Laughlin Steel Corp. in the development of this instrument, and particularly wish to thank General Motors personnel, both in the Pittsburgh and Detroit areas, for their continued interest and active cooperation in the plant trials.

DISCUSSION

Drawability Test Is Regarded as Yardstick

– R. S. Burns
Armco Steel Corp.

THE authors have developed a new and ingenious method of testing sheet steel for drawability or drawing quality. This test is very easily performed in both the steel mill and the stamping plant, and could be a very useful tool for both parties.

Our comments are necessarily based on our experience with the old standard test methods as they compare with this new test procedure. The old standard test methods, on which we have based our experiences during the past few years, recognized 22,000 to 28,000 yield strength, and 32 to 43 Rockwell B, as being of excellent drawing quality. In Fig. 6, this Rockwell range would be equivalent to 47 to 60 F, and the yield strength range would be equivalent to F 40 to 55 as judged by Fig. 7. It is noted that individual F test values at constant Rockwell B or yield strength vary considerably; that is, at 24,000 psi yield strength, F may be as low as 44, or as high as 54. This is indicative of other unknowns, or variance in testing procedure and sampling, or perhaps that more values per plotted point are necessary.

This leads to the question of determining the variance or standard deviation in any testing procedure. This is characteristic of all testing, and it is difficult to determine how much testing is necessary (even though a perfect correlation exists) to determine variations in breakage and scrap of, say, $\pm 2\%$.

This test is limited by its ability to test only the corners of a sheet. Corners are ordinarily the softest part of any sheet as manufactured, because in box-annealing they become hotter than the balance of the charge and are kept at this temperature for longer times than the balance of the charge. Furthermore, the testing of corners and comparing these results with the remainder of the sheet could be very misleading in testing material from top portions of rimmed steel ingots.

Like other tests also, it appears as though it would be necessary to know the mill routing before the test value becomes significant. To be more explicit, normalized, normalized-annealed, and box-annealed materials do not have the same relation between yield point and yield-point elongation. This relationship also varies with chemistry and amount of cold reduction before box-annealing. Fig. 8 shows the relationship between R value and yield-point elongation. It has been our experience that we are interested principally in yield-point elongation between 1 and 2%, which we would assume would be betweeen 0.020 and 0.040 yield elongation total in Fig. 8. This yield-point elongation range is covered by R values from 36 to 42 according to the curve, and we note considerable variation in R value for a given yield-point elongation or for constant R value of 35. Yield-point elongation may vary from 0.018 to 0.042. We say we are interested in yield-point elongation values of 1 to 2% because it takes over 1% yield-point elongation before stretcher strains can be seen on Pangborn surface, and it is not difficult to keep yield-point elongations below 2% by proper mill processing.

As we interpret Fig. 11, we would say that for F values of 60 and less, drawing performance differing between 1½% and over 5% breakage, can be predicted in only 1 out of 4 cases. This degree of correlation leaves much to be desired.

In its present state of development, it would appear that this test, as is true of other tests such as the Erichsen or Olsen cup, gives results that will be useful to users of sheet steel as a "yardstick," rather than as a "micrometer," for specification purposes.

Authors' Closure To Discussion

MR. BURNS states that, of necessity, he must compare "old standard" tests with the new method for conclusions. Were the old standard test procedures an infallible measure as reflected by actual performance with test results, there would of course be no incentive to find a better way.

With the recognition of 22,000 to 28,000 psi yield strength, and 32 to 43 Rockwell B as indicative of excellent drawing quality, Figs. 6 and 7 set forth a similar spread in F values, that is, 40 to 60, or in the range purported by the authors to be of best drawing quality. (Certain material characteristics producing tests at the bottom of the range of all of these methods prompts our conclusion that the middle area of each is the optimum.)

The notation that individual F values vary considerably at constant Rockwell B or yield strength appears true if the two latter tests were reproduceable constants of 100% accurate drawability predictions themselves. That such is the case only in some reasonably good percentage of cases is evident in a multitude of laboratory results at the scene of material fabrication. We might, therefore, reverse the

observation, considering the possibility of varying Rockwell B and yield strength at constant flex value.

Realistically, the comparison either way is inconclusive to the degree that common practice has been to consider results of several tests (yield strength, Rockwell B, cup, and so forth). As the flex test measures a mechanical property other than these, (resistance to a standard bend force) it would appear logical also to consider the test in addition to, rather than compared to "old standards."

We agree with Mr. Burns' statements regarding the testing of the corner area; however, extensive testing indicates little or no difference, edge to center, when tests are in the low range. Such differences appear exaggerated in a small percentage of cases with high tests. Too, this variation in the low to middle range is usually quite relative, so that tests always taken at the edge will be satisfactory when a required range for a given part is determined.

With regard to knowledge of mill routing being necessary, our paper deals only with the one product, cold-reduced, box-annealed, and while chemistry and degree of cold-reduction does have a bearing on drawability, their effect appears measured by the test. Other heat-treating processes or special steels may not necessarily respond to the test in a comparative range, requiring consideration for the specific material and process.

Our experience with hundreds of on-the-job tests indicate an almost infallible prediction of stretcher strain in material producing values of 30 or higher. The fluctuation is slight, only a point or two, depending of course on the elongation exerted by the draw operation. We find that material at 36 R, for example, or about 1% yield elongation, will stretcher-strain in most all cases unless suitably roller-leveled immediately prior to stamping. The variation of R values above 30 is of no consequence, as they simply represent the degree of stretcher-strain prominence, indicating to the fabricator the amount of cold processing required prior to stamping.

Fig. 11, as noted, includes total scrap and repairs from all causes. Subsequent data indicates rather conclusively that F values can predict performance in nine of ten cases when considering only drawability.

Again, we point out that the Flex-Tester is admittedly a "yardstick." However, "old standard" tests appear to fall in this category themselves, and to our knowledge no real "micrometer" test measure of drawability prediction has yet been found.

ORAL DISCUSSION

Reported by Walter Mincho
Great Lakes Steel Corp.

Homer Pratt, Fisher Body Division, GMC: Fisher Body is very interested in a test which is nondestructive. Through the GMC Sheet Committee we were able to obtain one of the original instruments. Our efforts in using the instrument in the laboratory and shop are as the author has described them. We concur 100% with his findings as presented in the paper. The *R* value radius readings are accurate. We found by channeling sheet steel with *F* readings of 60 maximum to presses that we were able to run with minimum down-time for die adjustment. Through laboratory tests we were able to correlate *F* readings against uniform elongation and yield point. Efforts to correlate Rockwell hardness and *F* readings were not successful, as Mr. Buker reported. We have testers in all our fabricating plants and are making an effort to accumulate a lot of data to determine how we can best use them.

E. A. Hahn, Lyon, Inc.: Have you done any work in stainless with this test?

Mr. Buker: We have made no extended efforts to test anything beyond normal low-carbon steel in the flat rolled field. Our company is not so diversified. I know of one or two instances where tests have been performed on 18-8 and 430 stainless in light gages. Results appeared to be satisfactory, but did not correlate in the same range as low-carbon steel. The instrument is a function of yield point. We can test any flat rolled product with a fairly low yield point. We could use it on dead soft stainless.

H. C. Smith, Great Lakes Steel Corp.: Does the rate of speed in bending have any effect on the test results?

Mr. Buker: We have not found that speed with which the test is forced back has any apparent influence on test results. Moderate pressure and steady increase of load is required for consistent readings. Care should be exercised so that no shock is absorbed by the instrument when the base is forced against the test sheet; this would kick maximum hand of the dial and cause a false reading.

Mr. Smith: Is it possible to test very light thicknesses and materials such as tin plate?

Mr. Buker: We have tested some light thicknesses, down to 0.019-in. gage aluminum. We can and have measured light gage. The instrument as designed now will not take tin plate past its yield point. We are working on a similar instrument of different design to take tin plate past yield point to obtain measurement of stiffness of such plate.

Mr. Smith: What is maximum capacity of instrument; how heavy a thickness can you test?

Mr. Buker: We can measure anything with the instrument up to clasp thickness. The opening is 0.100 in. We have tested 0.060-in. and 0.062-in. in cold-rolled annealed material satisfactorily, and it would depend upon yield point, beyond that, how much further we could go. For safety factor of the instrument calibration, no tests should be carried much beyond 1½ revolutions of the dial.

P. N. Stafford, South Dakota School of Mines and Technology: Could you reverse the process and estimate yield point and hardness of material by using instrument, without actual testing to yield point?

Mr. Buker: I would have to say no. It is not a method or substitute to determine yield point. The nature of the test is such, however, that yield point could be estimated from readings taken.

Mr. Smith: Is it possible to establish a drawability range so as to say material up to a certain test number will be satisfactory and those tests above that number unsatisfactory?

Mr. Buker: The purpose of the test is to establish drawability range. It takes a study of tests against a particular part. The test itself has limits to its accuracy. We have some spread on sheets, as from one corner to another. We have taken certain parts, made a large number of tests, and found on material up to, say *F* 65, that they worked satisfactorily; between 65 and 70, some would work and some would not, while nothing above *F* 70 would be found workable. Stamping-plant variables enter into the picture. We can establish a maximum and then use a borderline area where trials should be taken. The borderline range would be narrow on extremely severe parts, and as severity decreases the borderline range becomes more spread out.

NEW BUICK V-8 ENGINE

- Durability
- Fuel Economy
- Easily Serviced
- Light and Compact

BUICK has had a long and very successful history in the production of overhead-valve, in-line engines. Pioneering in the development of the overhead-valve principle, we have produced cars from 1903 to the present time powered by engines utilizing this design feature. Starting with a 2-cyl opposed engine, Buick has produced overhead-valve 4's, 6's, and 8's from 1903 to the present time.

The question naturally arises as to the reasons behind the decision to change to V-type engines in

AIMS and objectives behind the decision to adopt the V-type design for the large Buick engine in 1953 models are set forth in this paper. From 1903 until the present time Buick produced cars powered by engines utilizing the overhead-valve principle. The authors cite four main reasons for departing from this long tradition:

1. Newer styling, based on extremely low lines, demands an engine proportioned to fit within allotted space under the hood.

2. A compact V-engine is inherently light in weight, especially in larger sizes. Weight reduction improves car balance, handling, and performance.

3. Improved combustion chambers and fuels permit higher compression ratios. Resulting higher explosion pressures call for a more rigid engine structure, which can be achieved on the V-type.

4. Great strides in engine manufacturing made old tooling obsolete. Since new tooling was needed on the larger engine, the advantage of a type change was indicated.

In conclusion, the authors state that the new engine offers durability, fuel economy improvement, low production costs, light weight, and is easy to service.

The Authors

V. P. MATHEWS, (M '20) chief engineer of Buick Motor Division, GMC, first went to work for Buick in 1926 as a chassis engineer. In 1942 he left Buick to become general manager of the Andover Motors Corp. In 1945 he returned to General Motors as chief engineer of its Moraine Products Division. He was appointed chief engineer of Buick in 1951. Mr. Mathews graduated from Case Institute of Technology in 1917.

J. D. TURLAY is a staff engineer on special projects at Buick Motor Division, GMC. For the last eight years he has been working on the design and development of V-8 engines. He joined Buick in 1929, shortly after his graduation from Oregon State College, where he received his degree in mechanical engineering.

OFFERS MANY ADVANTAGES

V. P. Mathews and J. D. Turlay,
Buick Motor Division, GMC

This paper was presented at the SAE Annual Meeting, Detroit, Jan. 14, 1953.

the Series 50 and 70 Buicks at this time. In the first place, it should be emphasized that we do not believe that the V-engine has any inherent advantages over the in-line engine in regard to power or economy. As the head of our engine test department often says, "The cylinders don't seem to care whether they stand up straight or lean over." Increased performance in some of the newer cars has been accomplished with large displacement engines and by newer refinements which can be (and in many cases have been) adapted to the in-line engine.

Why then has the change been made, and why only on the large engine? Primarily for four reasons:

1. The proportions of the V-engine are more suitable for installation in cars with the newer styling and particularly in cars with the styling which General Motors believes will be standard several years from now. The cars of the future, the XP-300 and LeSabre, have extremely low lines and the 90 deg V-type engine was chosen for these cars because it was the only design which we could fit within the allotted space under the hood.

2. The V-engine, being more compact, is inherently lighter in weight than the equivalent in-line engine, especially in the larger sizes.

The many advantages of reduced weight and size are obvious. By reducing the engine weight, steering effort is reduced, while car balance, handling, and performance are improved. Weight is also a major factor in production cost, but to realize the maximum cost saving, any reduction in weight must be obtained by good commercial design, and not by the use of more expensive materials, or by the adoption of designs which are more complicated and difficult to manufacture.

3. The V-engine also is more rigid and is structurally more suitable to withstand the higher explosion pressures resulting from the higher permissible compression ratios obtained with improved combustion chambers and improved fuels.

4. The basic tooling of the Buick Roadmaster engine has been unchanged since the introduction of the justly famous Buick Century in 1936. New tooling was indicated at this time because of the great strides which have been made in engine manufacture in the last few years. The Buick 263 cu in. in-line engine on the other hand was retooled in 1950.

As has been mentioned, Buick has never before had a V-type engine in production, but extensive research work has been done dating from a 1931 experimental twin-6. Since 1944 a continuous program of high-compression V-engine development has been in progress. A special department was set up for the purpose of investigating the possibilities of this type of engine and many cylinder arrangements, V-angles, and combustion-chamber designs were tried. During this period 10 different types of engines were built and tested, and in all over 100 experimental models were made.

One promising line of investigation of 35-deg V-engines was abandoned, although these engines had many desirable characteristics, because the carburetor height could not be reduced sufficiently to

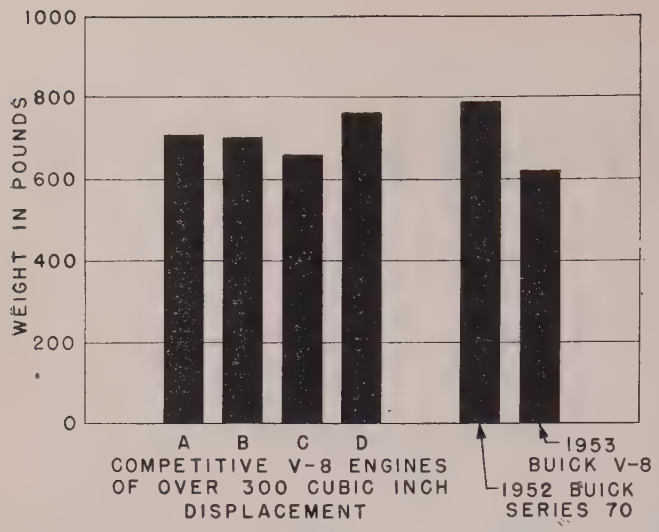


Fig. 1 – Engine total weight comparison

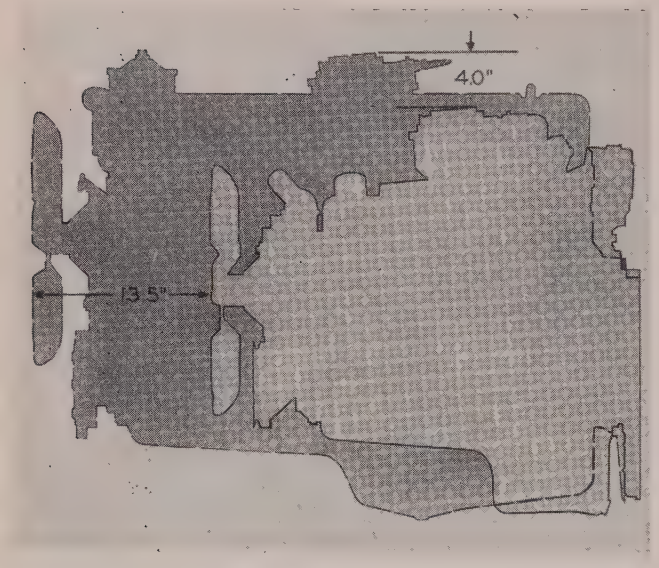


Fig. 2 – Engine size comparison. Black section indicates 1953 V-8 engine measured against gray background representing 1952 Series 70

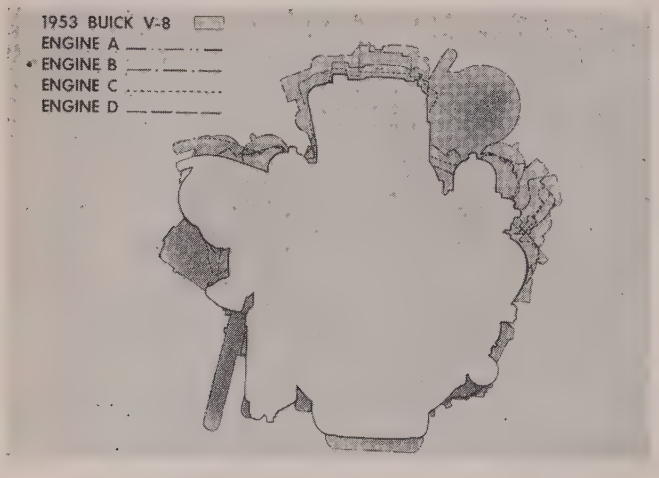


Fig. 3 – Engine size comparison

clear contemplated future hood levels. In fact, space requirements as dictated by the body stylist, are one of the most important, if not *the* most important factor, in determining the future trend of engine design.

The greatly reduced height and length of the 90-deg V-engine is a great help to the chassis designer and the body stylist, but the increased width is indeed a problem. The clearance condition is especially acute on the steering-gear side. The space between the extreme steering-lock position of the left front wheel and the left engine bank in many of the new designs leaves insufficient room for the frame side rail and steering gear, without very undesirable compromises in side rail section shape, steering-post angle and position, or turning radius; compromises which the Buick management chose not to accept. The Buick design, incorporating a stroke/bore ratio of 0.8 and a vertical, in-line positioning of the valves served to minimize the overall engine width. These engine design features accompanied with a 1-in. increase in front tread width permitted the continuation of all chassis improvements developed with the former in-line engine.

The new Buick V-8 engine has a 3.2-in. stroke and a 4-in. bore, resulting in the aforementioned stroke/bore ratio of 0.8 which is, we believe, the lowest used in any American production engine at the present time. The short-stroke, big-bore adopted in all the new V-8 engines has many real advantages, but the stroke/bore ratio of the new Buick engine was chosen chiefly because, from our investigation, it produced the minimum external engine size. The 322 cu in. displacement of the new engine as compared to the 320 cu in. displacement of the straight-8 Roadmaster, is considered ample in view of the power gains obtained from some of the newer design features to be discussed later.

The Fireball V-8 engine program was started in March, 1950, and, because of the difficult machine-tool situation, production was delayed until this year. The tool and building program was enormous, but it is now completed, and the new engines are in production. Needless to say, we are very happy with them.

Objectives

Probably every designer on attacking a new project is fired with a desire to produce a finished product so perfect as to excel in every detail anything that has been done before. As the design progresses, invariably some balancing of objectives must be done, and the individual emphasis on the many design factors will determine the trend of the final design.

In planning the new Buick engine no compromise could be accepted in the requirements of a good powerplant; that it should be powerful, efficient, smooth, quiet, durable, and easily serviced.

However, these objectives were to be realized with the lightest and most compact package which we could produce, consistent with minimum manufacturing cost.

Weight and Size

The greatest weight saving in the new Buick engine was obtained by adopting the inherently lighter 90-deg V-8 principle. But we went further. Every design detail was investigated for possible further weight reduction. Many of the old rule-of-thumb drafting room design standards were found to result in overdesigned and overweight parts. The reduction in size of lightly loaded flanges and bolt bosses, for example, provided worthwhile savings when applied to the entire engine.

In designing the stressed parts of the engine for minimum weight, consistent with adequate fatigue life, full advantage was taken of the fact that an automobile engine operates most of the time at part throttle. In any of the short stroke-large bore engines the calculated gas pressure loads and stresses are extremely high, and the inertia loads are correspondingly low. Luckily, experience has shown that even under severe road testing, fatigue life of the parts is more dependent on the inertia loading than on the gas pressure loading. This result is largely explained, of course, by the relative number of cycles of each type of loading which the engine undergoes in road operation.

In establishing the endurance standards for the new Buick engine we took the position that while the engine must be capable of withstanding our standard full-throttle high-speed dynamometer test, and not less than 200 hours of full-throttle power development running, the true gage of engine life was to be the behavior on the road.

The production-built Buick V-8 engines are 170 lb lighter than the 1952 Roadmaster engines (Fig. 1), and are lighter than any of the competitive V-8 engines of more than 300 cu in. displacement. The weights shown are based on the dry engine weight with all standard accessories, without transmission or clutch.

In addition to the compact design features discussed previously, the old standard design proportions and running clearances were re-examined, and where practicable, revised to save space. For instance, a very small length/diameter ratio of the piston, and its 1/16-in. running clearance with the counterweights aided in reducing the height and width of the crankcase.

The Buick V-8 engine is 4 in. lower and 13½ in. shorter than the former Roadmaster engine (Fig. 2), but the width is considerably greater. However, it compares very favorably with that of competitive V-8 engines as shown in Fig. 3.

Power Output and Efficiency

As has been stated, in obtaining a light, compact unit, no sacrifice in engine output and efficiency

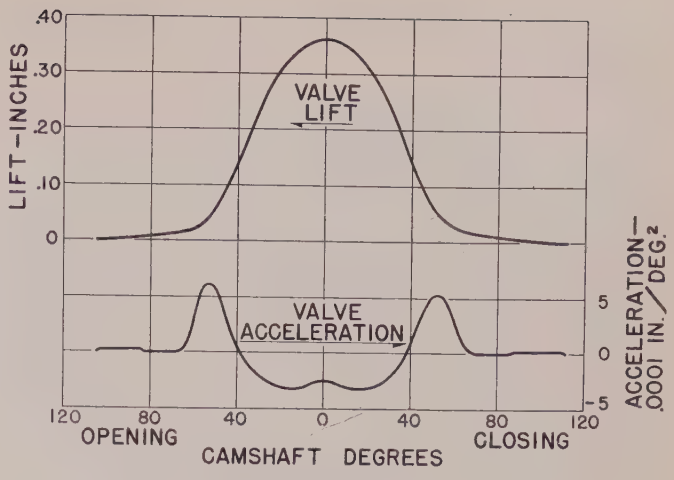


Fig. 4 – 1936 cam design

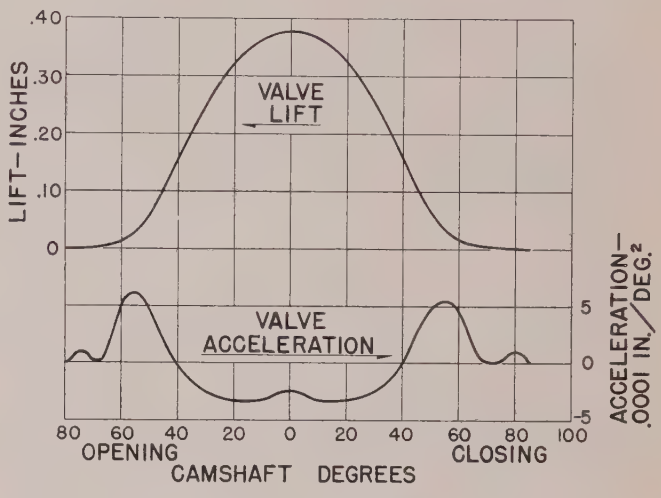


Fig. 5 – 1953 cam design

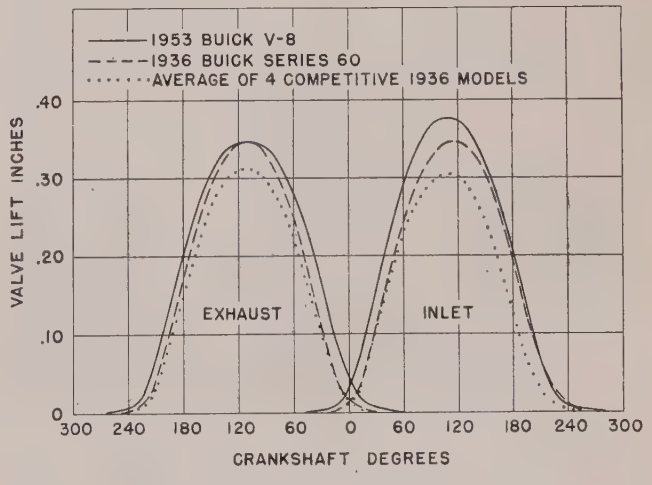


Fig. 6 – Valve lift comparison

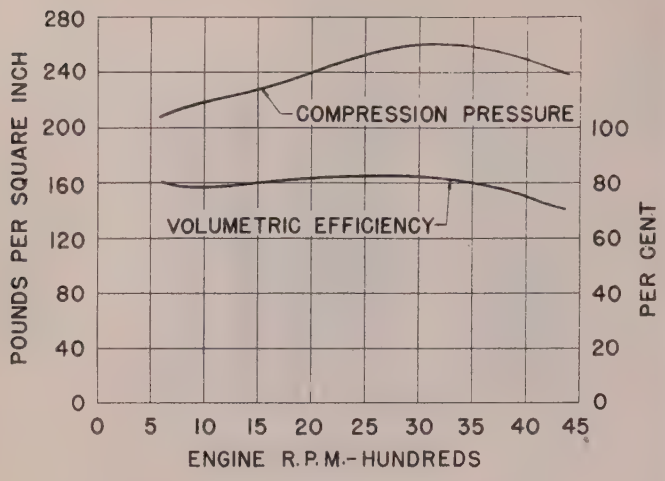


Fig. 7 – Compression pressure and volumetric efficiency

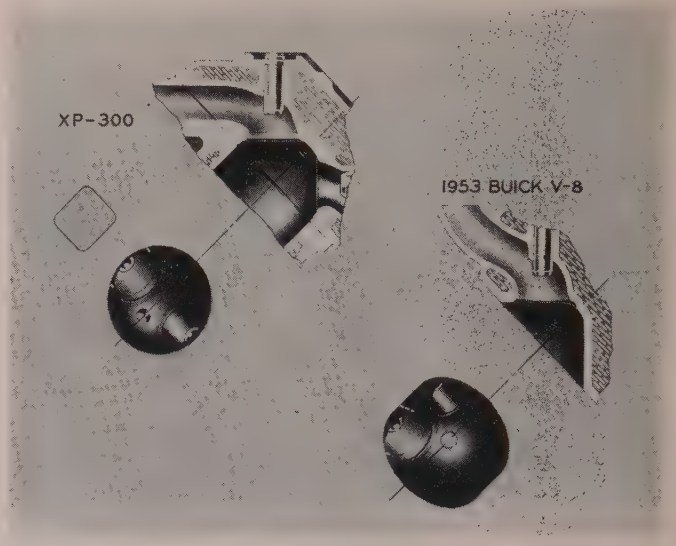


Fig. 10 – Inlet valve port comparison

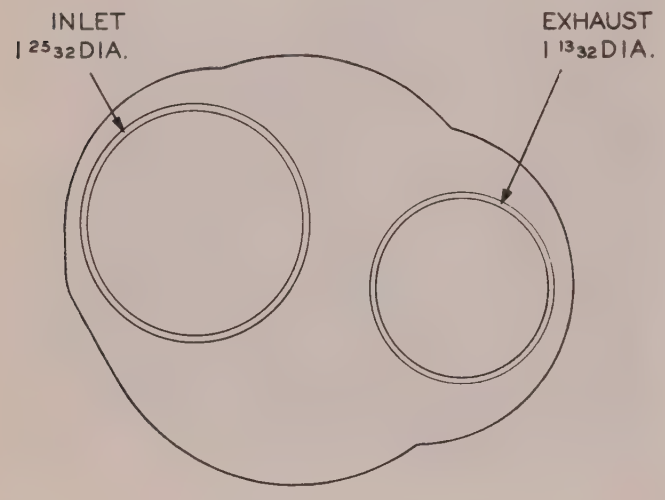


Fig. 8 – 1918 valve size comparison

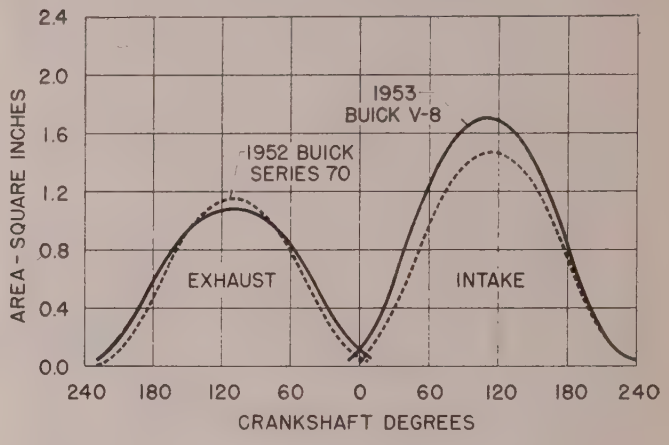


Fig. 11 – Effective valve opening area comparison

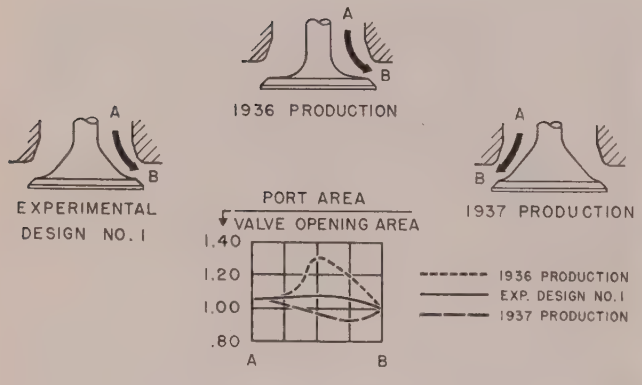


Fig. 9 – Streamlined inlet valve development

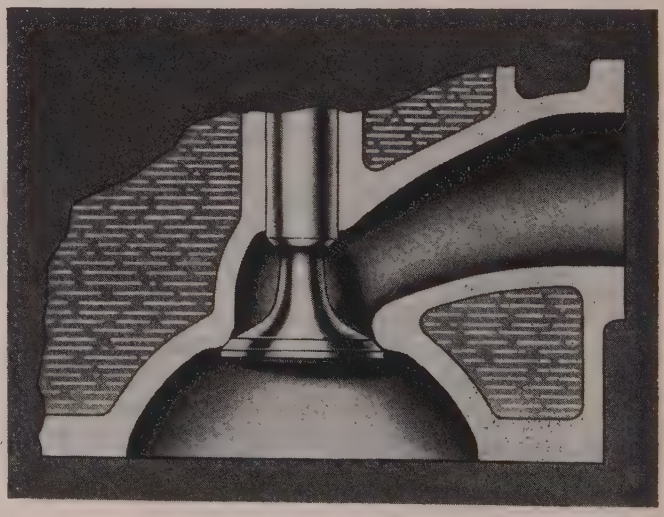


Fig. 12 – 1936 streamlined exhaust port

could be accepted. There has been much discussion on the relative merits of designs which emphasize high volumetric efficiency and those which improve the mechanical octane rating and permit the use of higher compression ratios, with consequent higher power and economy. It is difficult to see why there should be anything antagonistic in these two approaches to higher engine power and efficiency. Buick has carried out a continuous development program on improving the overhead-valve engine along both lines, and has made many important engineering contributions. Some of these improvements have been claimed as new discoveries by companies which have recently adopted the overhead-valve principle, notwithstanding the fact that many of these same features have been in production at Buick for many years.

Some of the earlier Buick developments which improved volumetric and combustion efficiency and which influenced the Buick V-8 design will be discussed briefly. Later the engine will be described in more detail with some discussion of the engineering problems encountered.

Cam Design and Valve Timing

Push-rod-operated overhead-valve engines require much more highly developed cams than do L-head engines operating in comparable speed ranges, because of the greater deflection of the valve mechanism parts under operating conditions. Buick has always been confronted with this problem and, as engine operating speeds increased, cam and valve mechanism design had to be improved to meet the requirements.

By 1936 Buick had developed a new type of cam profile (Fig. 4) which has been used in production since that time. All recently developed overhead-valve V-engines have adopted cams of similar contour. Briefly, the Buick cams are designed with an acceleration curve of generally sinusoidal form with all changes in acceleration, and therefore in load, as smooth and gradual as possible.

The cam profiles of the Buick 90-deg V-engine (Fig. 5) have been carefully tailored to suit the elastic characteristics of the valve mechanism and to minimize spring surge. The pump-up speed of the hydraulic lifters is 5500 rpm for the exhaust valves and 5300 rpm for the inlet valves. These speeds are high enough above the normal top operating speeds to provide an ample factor of safety, and are quite gratifying considering the high inlet valve lift of 0.378 in. and the relatively low valve spring loads of 62 lb closed and 144 lb open.

Buick developed a high-lift high-speed timing for the 1936 Century models (Fig. 6), and the basic principles of this timing are still used. The Buick V-8 has an even wider timing with the higher inlet valve lift already mentioned. The resultant volumetric efficiency is excellent as is shown by the volumetric efficiency and compression curves (Fig. 7). Note that the volumetric efficiency is above 75% at 4000 rpm.

Inlet Valve and Port Design

As early as 1909 Buick recognized the value of an inlet valve of much larger size than the exhaust valve and the Buick-Bug race car which exceeded 105 mph in 1910 utilized this feature. Buick adapted this principle to its 1918 production engines (Fig. 8) with a 1 25/32-in. diameter inlet valve and 1 13/32-in. diameter exhaust valve which is very close to the size and proportion used in most of the new engines today.

Throughout the middle '30's the Buick engineering department was very active in investigating the effect of manifold, port, and valve configuration on flow resistance over the full range of valve openings. Best results were obtained by maintaining a nearly uniform velocity in the whole induction system from the carburetor to the exit side of the inlet valve seat. The valve throat area is very important, and a considerable reduction in flow resistance has been obtained by reducing the inlet throat area to approximately the maximum valve opening area. The Buick streamlined inlet valve (Fig. 9), adopted in 1937, with its conical section at the junction of the head and stem was designed to avoid a sudden enlargement in cross-sectional area at this point with consequent energy loss.

The in-line valve arrangement was retained in the 1953 Buick engine chiefly to make the engine more compact, to save weight, and to facilitate manufacture.

Surprisingly enough, the flow restriction of the Buick V-8 inlet port and valve was no greater than that of the experimental engine used in the XP-300 and LeSabre cars, under the same conditions (Fig. 10) with the same size valve and valve openings. This experimental engine was designed for high output and had both valves on the transverse centerline of the combustion chamber. Upon investigation, the curving inlet port of the Buick V-8, which was so shaped because of clearance conditions in the head, was shown to have a slight advantage over the straight-in port of the XP-300, because of the effect on flow of valve stem and guide interference.

Exhaust Valve and Port Design

The size of the Buick V-8 exhaust valve was based on earlier investigations, but its adequacy has been thoroughly proved by further testing of different valve sizes in the V-8 engine. The most critical part of the exhaust event is the blowdown period, that part of the cycle from the time the valve opens to a little past bottom dead center. A valve opening and port cross-section area sufficiently large to handle this portion of the cycle has been shown by test to be adequate for the remainder of the exhaust stroke.

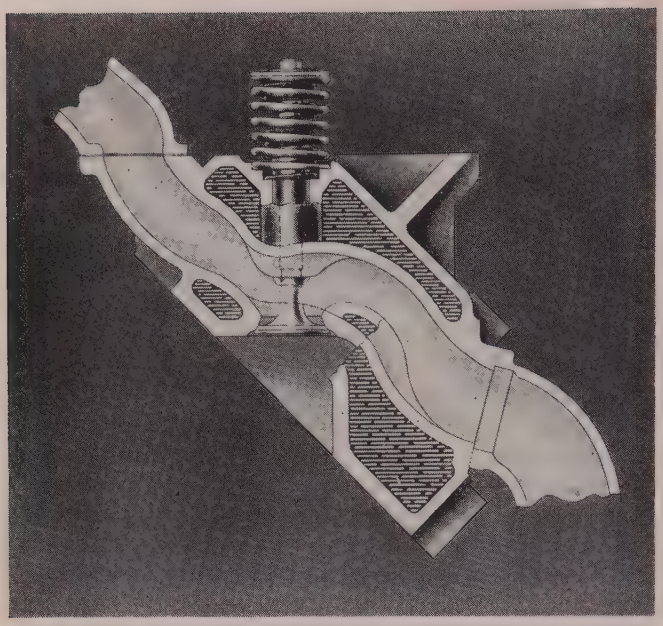


Fig. 13 – Exhaust port at cross-over

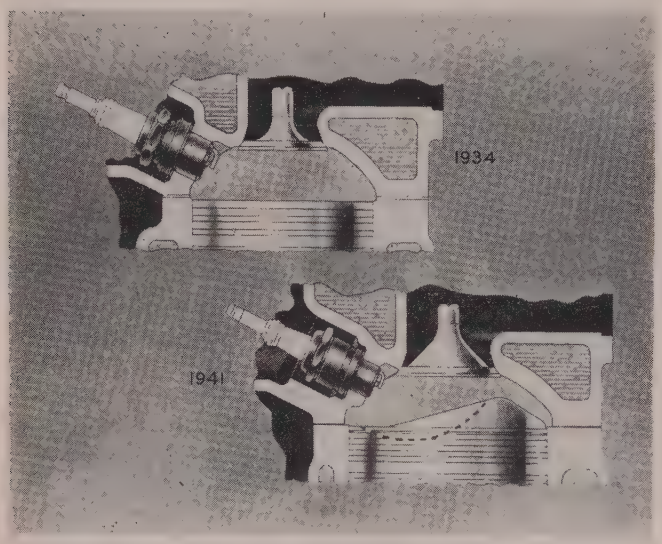


Fig. 14 – Earlier Buick combustion chambers

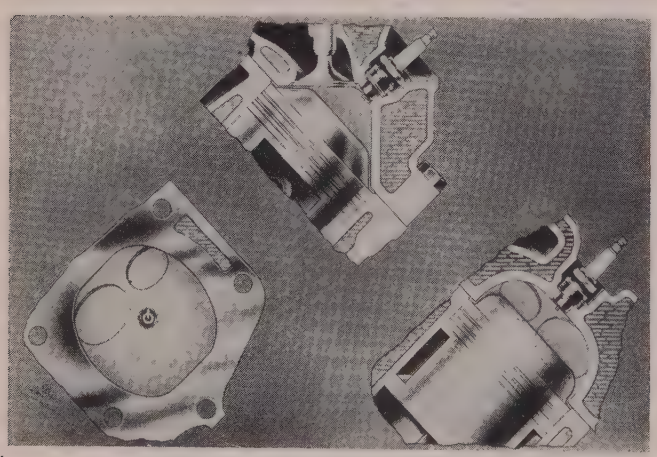


Fig. 15 – Buick V-8 combustion chamber

At bdc the exhaust valve in the average engine has opened less than half of its maximum lift. Therefore, a smaller valve opening earlier and higher can duplicate, through the critical range, the opening area of a larger valve (Fig. 11) with the added advantage that flow efficiency is maintained by avoiding the sudden expansion into a large port.

The use of this relatively small exhaust valve made a more compact combustion chamber possible, improved valve cooling, and reduced cost, without entailing any loss in power. The reduced valve weight brought the exhaust pump-up speed well above that of the inlet valve, thus providing insurance against exhaust valve breakage.

As shown by the 1936 port design (Fig. 12), Buick engineers have long recognized the benefits of streamlined exhaust ports. Considered as the exit side of a venturi, they are designed with a gradually increasing cross-sectional area from the valve throat into the main section of the manifold to obtain all the pressure recovery possible.

Because of the valve arrangement adopted for the Buick V-8, the exhaust ports (Fig. 13) must cross the combustion chamber, but the area of the port exposed to water is minimized by the reduced width of the head, the angle at which the exhaust flanges are placed, and by the relatively small diameter of the ports.

Combustion-Chamber Design

Although many valve arrangements and head shapes had been tested previously, the best combustion chamber developed at Buick by 1934 (Fig. 14) had a semihemispherical shape in the transverse section but was necessarily elongated in the longitudinal section to accommodate the valves. The piston top was flat and no quench area was provided. A further improvement in the combustion chamber was obtained with the turbulator piston adopted in 1938, modified in 1941, and used in production up to the present time. This design provided a considerable increase in turbulence and brought the bulk of the charge closer to the spark plug. All the combustion-chamber investigations made at Buick have shown the great importance of short flame travel and high turbulence in reducing octane requirements, although increased engine roughness in past models has prevented full benefit from being obtained from these factors.

In planning the new large bore engine a centrally located spark plug was considered mandatory. In addition to the flame travel considerations, there is considerable evidence that the centrally located spark plug provides improved part-throttle ignition characteristics.

In the production design (Fig. 15), the Buick combustion chamber is of symmetrical inverted-V form with the spark plug at the apex and the valves in line longitudinally at a 45 deg angle to the cylinder axis. The spark plug is more nearly cen-

trally located than in any other American automobile engine.

The combustion chamber is fully machined for more accurate volume control. The piston crown is raised and is shaped to conform closely to the head shape, with a flat top surface. The resultant combustion chamber is very compact with minimum flame travel from the spark plug to the extreme edge of the effective portion of the combustion space. The close clearance space provided around the lower part of the piston crown results in high turbulence in the combustion chamber during the latter part of the compression stroke. The excellent combustion-chamber characteristics of the Buick V-8 have made possible the use of the very high compression ratio of 8.5/1, using premium fuel, with remarkable freedom from combustion harshness.

General Motors Research collaborated with Buick in developing the new Buick combustion chamber and piston-top shape, and tested many cylinder-head and piston-top designs for the new Buick engine, including two without quench areas, before the final design was approved. The General Motors research experience and new techniques in combustion-chamber testing have been extremely valuable. Their work on the combustion process has been most thorough for many years, and recently they have undertaken a very comprehensive testing program in which practically every production combustion chamber as well as a great many which are still experimental have been evaluated.

Covers and Exhaust System

The compact design of the 1953 V-8 engine is immediately apparent (Fig. 16). The horizontally mounted rocker-arm covers and vertically mounted spark-plug covers carry on the traditional Buick arrangement. The spark-plug covers protect the ignition wires from moisture and excess radiant heat from the exhaust manifold. These covers are also effective in reducing television interference.

The exhaust manifolds are of the 4-port type, to avoid the exhaust overlap periods resulting from siamesed ports, with the left-hand manifold outlet directed forward to shorten the crossover exhaust pipe which passes under the front of the oil pan, and to move this source of radiant heat away from the starter. The heat control valve is located in the left-hand manifold at the outlet flange. The right manifold outlet (Fig. 17) is directed backward and outward so that the exhaust pipe will clear the full-flow oil filter at the rear of the engine.

Generator

The generator is mounted on the right exhaust manifold, in the position adopted by other General Motors engines. The relative location of the generator and spark plugs is such that all plugs can be serviced without disturbing the generator (Fig. 18).

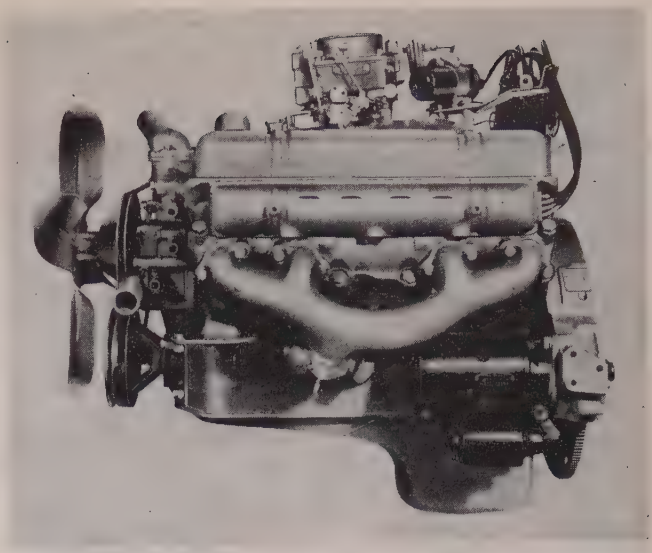


Fig. 16 – Left side of Buick V-8

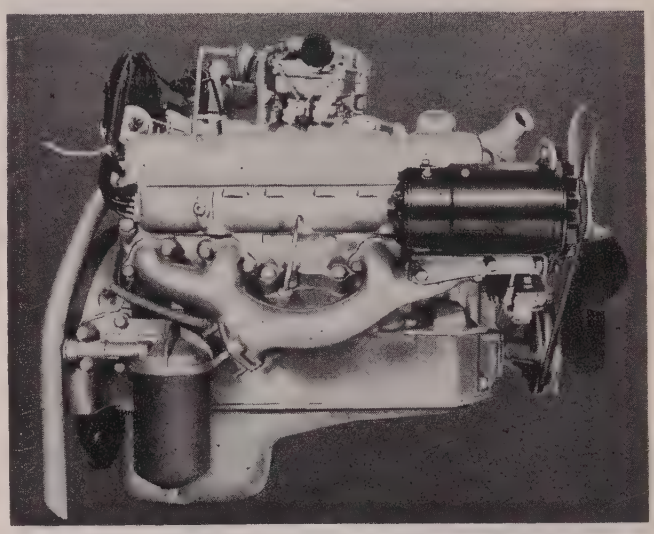


Fig. 17 – Right side of Buick V-8

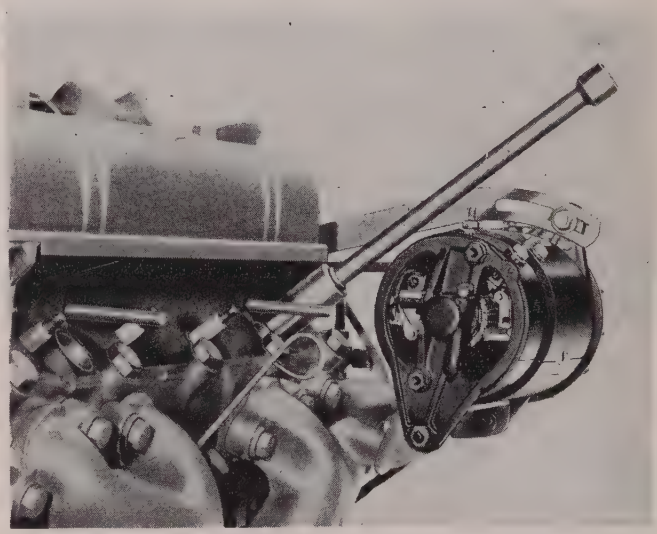


Fig. 18 – Spark-plug removal clearance with generator

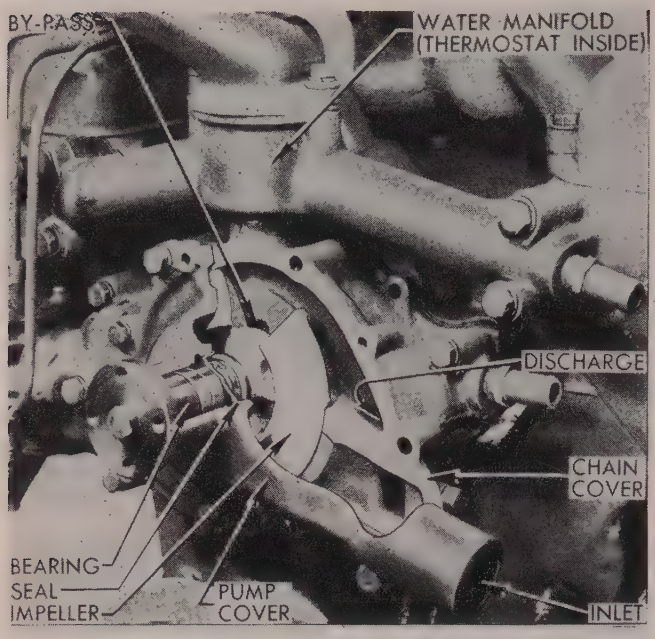


Fig. 19 – Water pump section

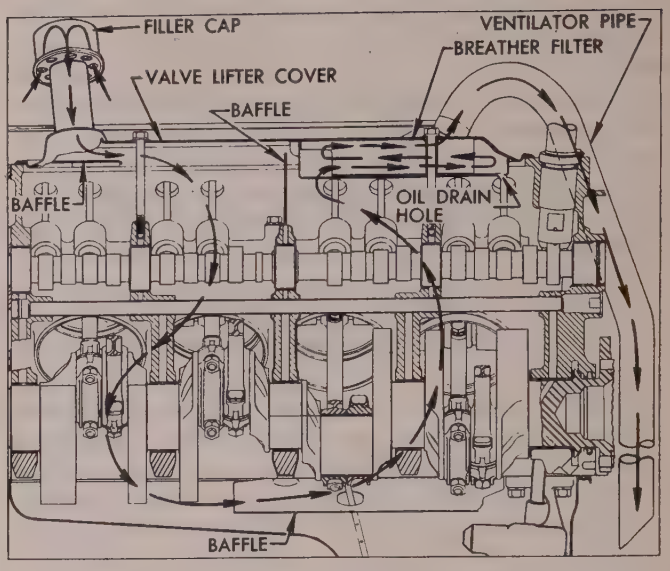


Fig. 20 – Crankcase ventilating system

Fig. 21 – Inlet manifold section

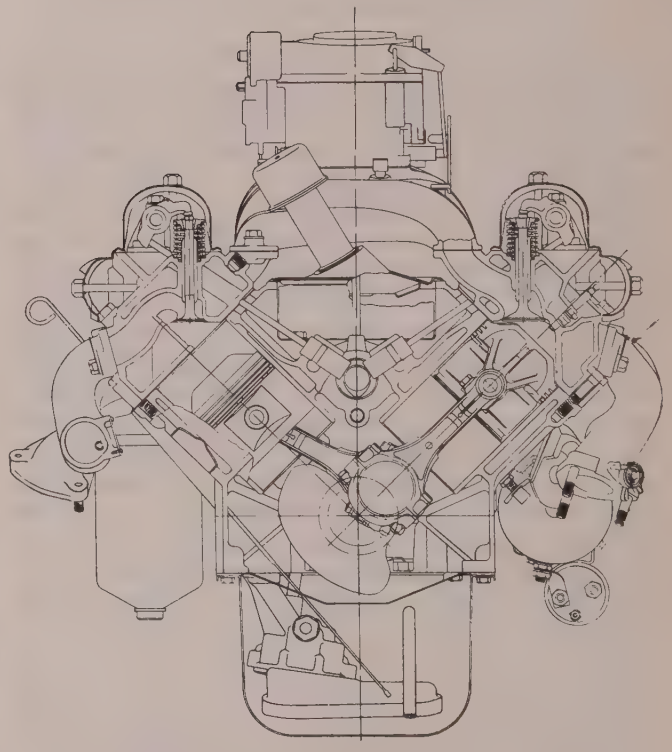


Fig. 22 – Transverse cross-section of Buick V-8

Cooling System

A cast timing-chain cover is used, with the water-pump impeller housing and discharge passages recessed in the front face (Fig. 19). The water-pump cover forms the front half of the pump housing and carries the inlet pipe and the impeller bearings and seal, with one side of the inlet passage formed by the water-pump cover and the other by the timing-chain cover and the impeller disk. The water flows from the inlet side through six holes in the hub to the vane side of the impeller. This construction saves considerable space and opens up the passages so that these parts can be die cast in aluminum alloy. At present they are being cast in grey iron because of economic considerations.

Discharge passages from the water pump are provided to supply cooling water to each bank of the cylinder block. The standard Buick straight-through cooling system is used with connecting water holes at the rear of each cylinder head and cylinder block. A water manifold carrying the thermostat housing is bolted to the water outlet passages at the front of the cylinder heads. To provide a bypass connection, the water manifold is spigotted into a hole in the timing-chain cover leading to the pump inlet, with a rubber O-ring seal. The use of a separate water outlet manifold simplifies servicing as the intake manifold may be removed without breaking any water connections, and the heads may be removed without disturbing the water pump.

Fuel Pump

The fuel pump is mounted on the timing-chain cover. This low position, in line with the fan blast, is helpful in avoiding vapor-lock troubles.

Crankcase Ventilation

The combined crankcase ventilator inlet and oil filler cap is located in the front (Fig. 20), and the multiple pass outlet is located in the rear of the lifter cover. A transverse baffle separates the front and rear sections of the lifter compartment and directs the flow of air in the ventilating system.

Carburetor and Manifold

A 2-barrel carburetor is provided in the 1953 Buick Series 50 line, and a 4-barrel carburetor for added power in the Series 70. Except for the carburetor flange the two manifolds are identical. The Buick inlet manifolds (Fig. 21) provide 90-deg T-shaped sections at all branching points. The individual exhaust porting is carried to the heat crossovers, providing direct hot spots at each of the 4 horizontal T-sections of the manifold, and extensions of these hot spots surround the carburetor riser section.

Piston

Fig. 22 shows a transverse section through the engine. The piston, although much larger in diam-

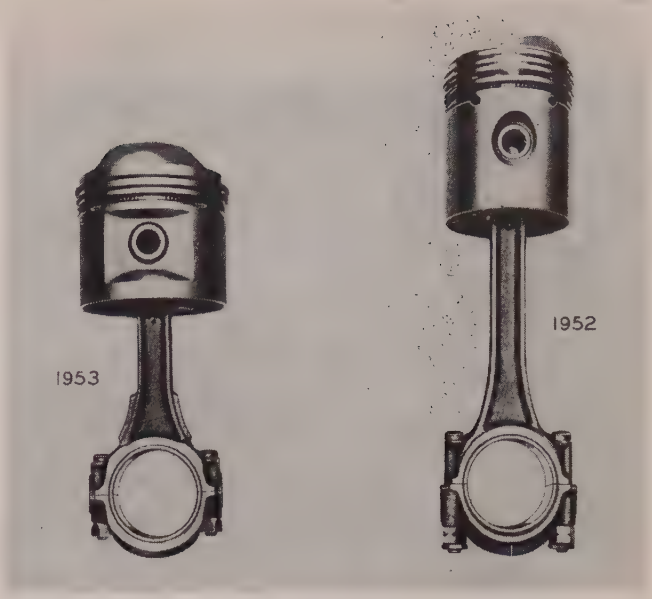


Fig. 23 – Piston and rod assembly comparison

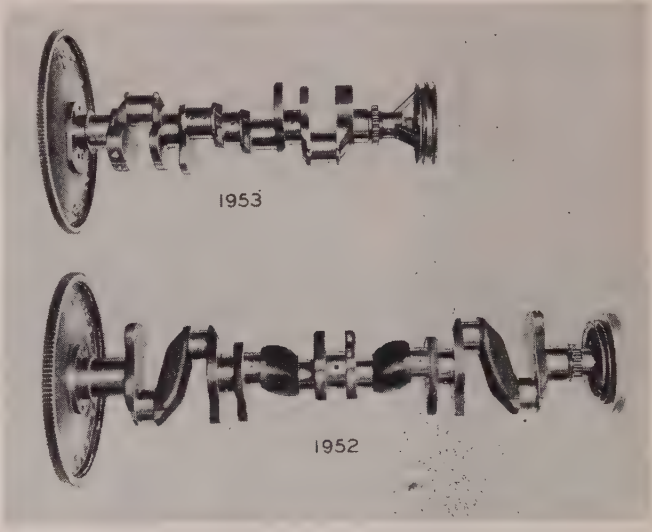


Fig. 24 – Crankshaft comparison

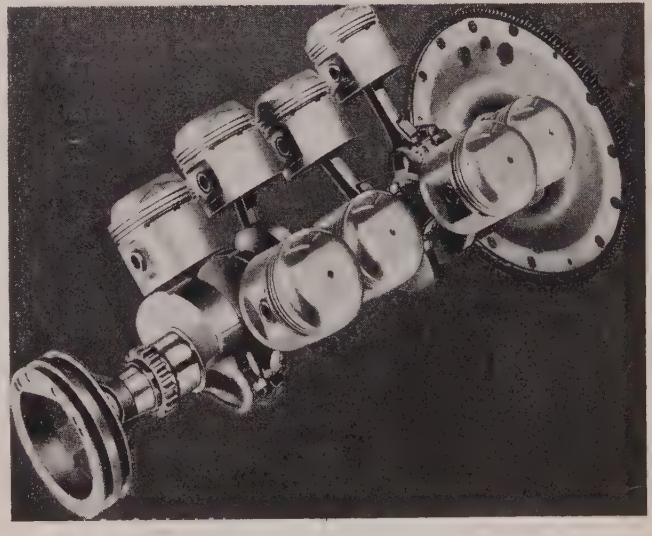


Fig. 25 – Power train assembly

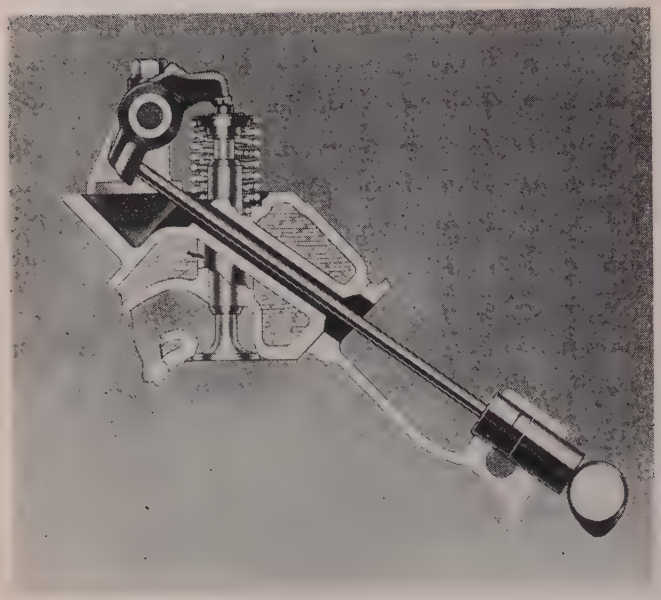


Fig. 26 – Valve-actuating mechanism

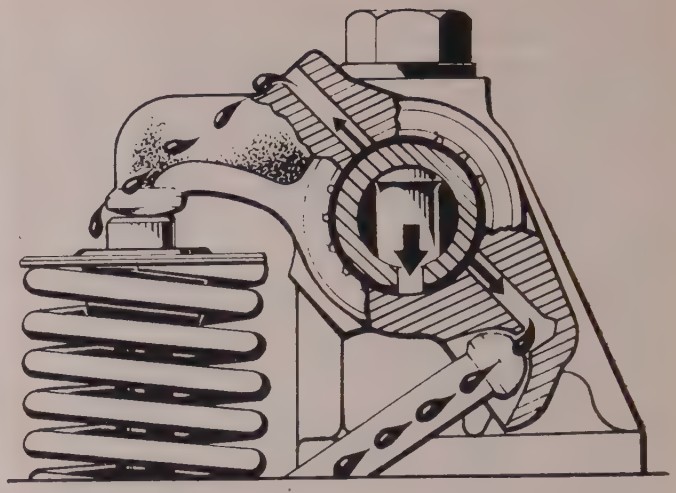


Fig. 28 – Rocker-arm lubrication

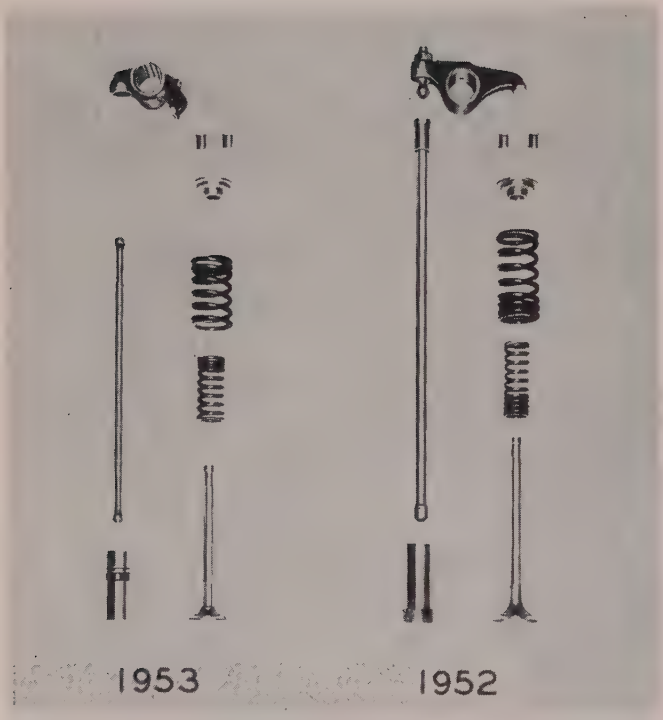


Fig. 27 – Valve train comparison

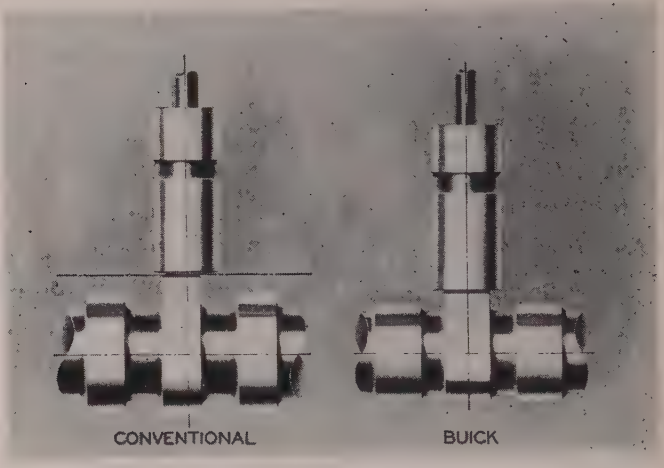


Fig. 29 – Vave lifter and camshaft alignment

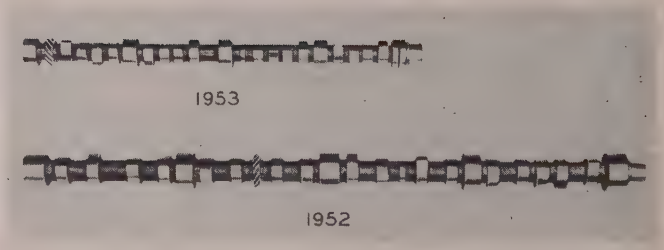


Fig. 30 – Camshaft comparison

eter than in previous models, follows time-tested Buick practice with 1-piece aluminum alloy construction and full skirt trans-slot design. Two conventional 5/64-in. compression rings and one 3/16-in. flexible steel oil ring are provided. The piston pin is not offset.

Connecting Rod

The connecting rod is very short (Fig. 23), only 6 in. between centers, which in conjunction with the short stroke gives a rod length/stroke ratio of 1.875, a very conservative figure. The rods are brought to uniform weight by machining off the required amount of stock from lugs located near the center of gravity of the rod. Our computations show that commercial tolerances in balancing the connecting rods will be less detrimental to overall balance in the new engine than in the former in-line engine with its balance lugs at each end of the rod. The production Buick-method of clamping the piston pin in the connecting rod is retained, because we have found nothing superior in quietness, durability, and replacement cost.

Crankshaft and Bearings

The crankshaft (Fig. 24) has five main bearings with the rear bearing flanged to carry the thrust load. The main and connecting-rod bearings are of the replaceable liner type of Durex 100A material as used in previous Buick engines. The exceptionally great bearing journal overlap, of over ¾ in., contributes to the outstanding rigidity of the crankshaft, which from our tests is greater than that of any other American automobile engine now in production. The natural frequency of the crankshaft coincides with the engine firing frequency at 5300 rpm, well above the maximum engine operating speed. The weight of the crankshaft is 56 lb which is lighter than that of any of the new engines of more than 300 cu in. displacement, and less than half the weight of our 1952 Roadmaster crankshaft.

Due to the exceptionally short stroke and short connecting rod the space left for counterweighting is severely limited (Fig. 25). The outer surface of the counterweights is cam-turned to maintain maximum useful counterweighting with a constant minimum clearance with the bottom of the piston skirt. The radius at the tips of the counterweights is limited by the clearance with the cylinder barrels, and this barrel clearance radius is blended into the piston clearance contour, with the final shape so calculated as to permit high-speed cam-turning in production. A small amount of counterweighting is carried in the crankshaft pulley and in the flywheel. This method of completing the counterweighting is very effective from a weight standpoint because of the long span from the pulley to the flywheel. It also makes possible a correction for any future piston weight changes without affecting the crankshaft counterweight tooling or balance.

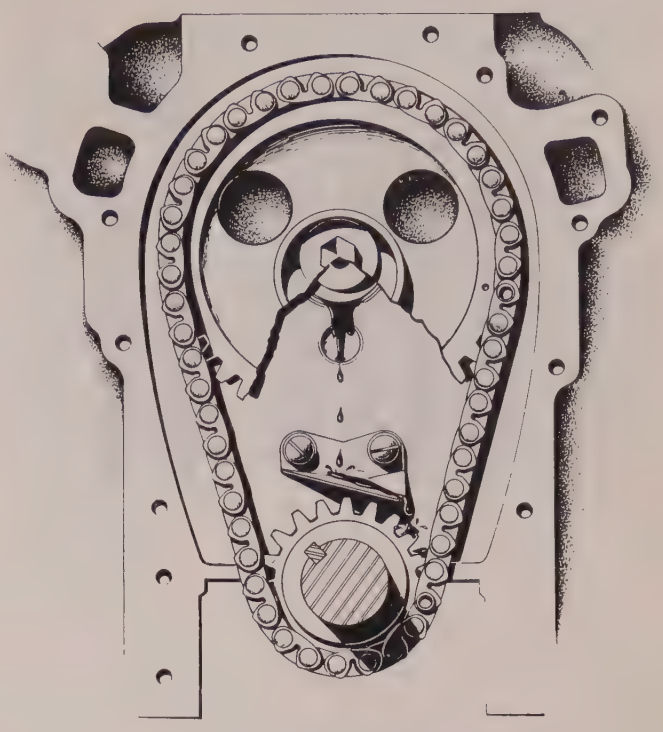


Fig 31 – Timing-chain lubrication

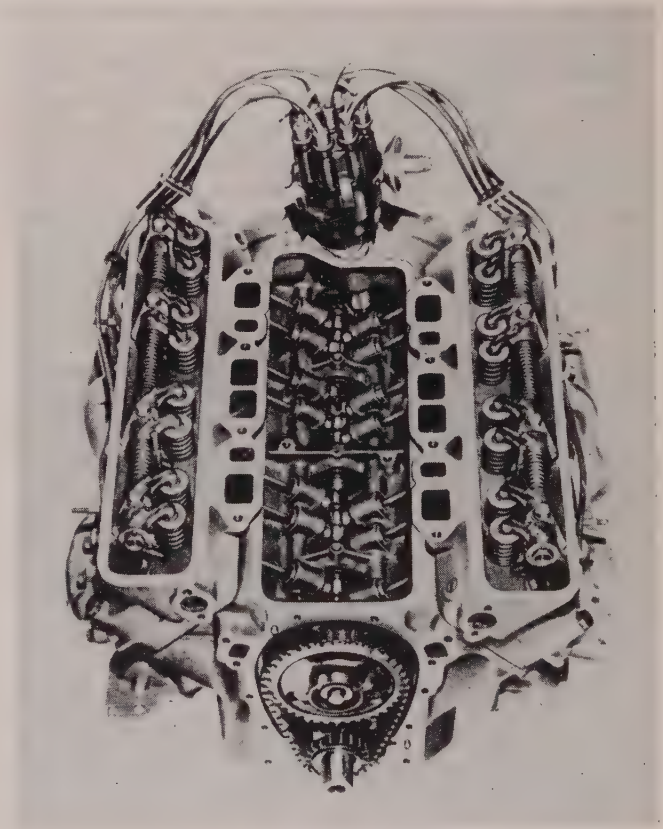


Fig. 32 – Top of Buick V-8 with manifold and covers off

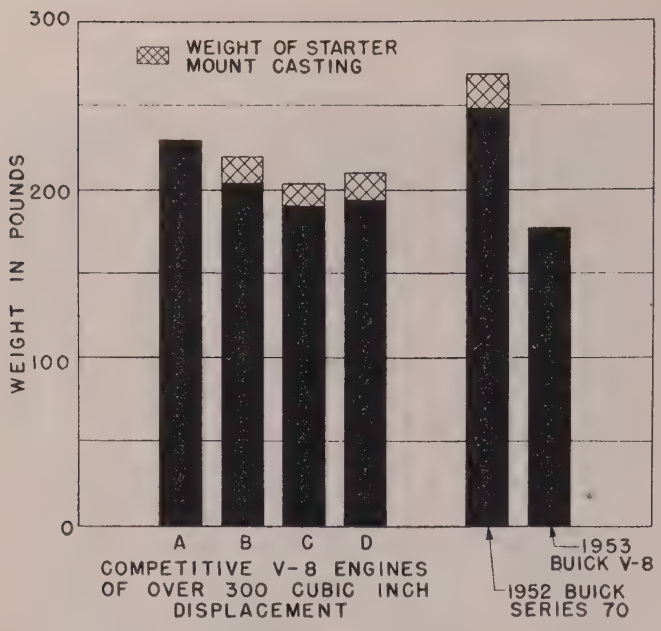


Fig. 33 – Crankcase weight comparison

Valve Train

The arrangement of the valves, and valve-actuating mechanism (Fig. 26), is unique in the Buick 90-deg V-8.

The valves are arranged side by side at an angle of 45 deg to the cylinder axis, and therefore in a vertical plane with respect to the ground. The reversed position of the rocker arms with the centerline of the rocker shaft outside the line of valves, and with the push-rods crossing the valve stems, results in a very compact cylinder head. The design also makes possible a common horizontal gasket surface for the rocker-arm cover and for the intake manifold, which simplifies machining, and even more important, greatly reduces the sealing problem at these critical points. The valves are $\frac{7}{8}$ in. shorter than the straight-8 valves (Fig. 27), and the dual valve springs are 7/16 in. shorter than past production, thereby reducing valve operating inertia as well as reducing material cost.

The pearlitic malleable iron rocker arms are very small and are of the nonadjustable type (Fig. 28). The rocker-arm bearings are broached with eight longitudinal grooves, which not only serve to control the amount of oil metered to the valve stems and push-rods, but have proved to be so effective in preventing rocker-shaft scoring that the antiscuff coating formerly used on the rocker arms is no longer necessary.

The push-rods are of $\frac{1}{4}$-in. diameter solid rod, upset to provide $\frac{3}{8}$-in. spherical diameter ends to eliminate excessive wear at these critical points. The exceptionally short $8\frac{3}{8}$-in. push-rod is made possible by the compact engine design.

Hydraulic valve lifters of the diesel equipment type are used. The bodies are cast iron with chilled wearing faces ground flat and ferrox-coated. The steel camshaft has cams 9/16-in. wide and ground without taper. No attempt is made to spin the lifters, although some of the lifters turn as a result of manufacturing alignment variations (Fig. 29). We have found consistently longer life with the lower unit load resulting from a wide parallel contact of the lifter face on the cam.

Camshaft and Drive

The camshaft (Fig. 30) is carried in five bearings, and is unusually small and light. The timing-chain drive (Fig. 31) is lubricated with the overflow oil from the front camshaft bearing, directed by means of a small stamping to the lower crankshaft sprocket and the inside of the chain.

The fuel pump eccentric is a hardened and chrome-plated cup-stamping driven from the front of the camshaft sprocket.

The distributor and oil pump are driven by a gear at the rear of the camshaft in the orthodox manner (Fig. 32), except that the distributor and oil pump are located on the right side of the camshaft. This arrangement puts the distributor gear thrust upward and eliminates the need for an additional bearing in the crankcase to take the thrust load, and results in some weight and cost saving.

Crankcase

Since the crankcase is the heaviest single part in the engine, it should offer the greatest opportunity for weight saving (Fig. 33). In the Buick design the starter mounting is carried on the flywheel housing, which is an integral part of the crankcase and eliminates the separate starter mounting casting used in other V-engine designs. The crankcase flange is far enough below the crank centerline to provide a flat, continuous oil-pan gasket surface with consequent sealing advantages. Contrary to accepted opinion this construction reduced weight. The dropped flanges (Fig. 34) not only stiffen the crankcase proper, but because of the greater vertical height of the attaching flanges, greatly increase the rigidity of the attachment of the engine to the transmission bell-housing.

The cylinder bores extend 1 5/16 in. below the water jackets. Attachment of the extended portions of the bores to the bulkheads is avoided by casting $1\frac{3}{4}$ in. by 3 in. windows in the intermediate bulkheads which effect a further weight saving. Substantial reinforcing ribs tie the water jackets into the main bearing bolt bosses on each side of the crankcase "windows." The center of each intermediate main bearing is reinforced vertically by a Z-shaped wall, which carries the main bearing oil passages and ties into the center oil gallery and camshaft bearing structure.

Five head bolts per cylinder (Fig. 35) are so placed as to carry the major part of the gas pressure loading on the cylinder head into the water jacket walls, rather than into the cylinder barrels.

The bolt spacing conforms to the structural characteristics of the head and block, with closer spacing across regions of greater deflection. The design has proved to be very effective in reducing gasket and cylinder distortion troubles.

A one-piece embossed steel gasket is used with a double bead around the combustion chambers and ample gasket area at all critical locations.

The main oil gallery, which supplies the main bearings and camshaft bearings, is a steel tube cast in the block. This construction was used primarily to open up the center section of the crankcase casting with consequent easing of strains during cooling in the foundry, and secondarily as a means of saving weight.

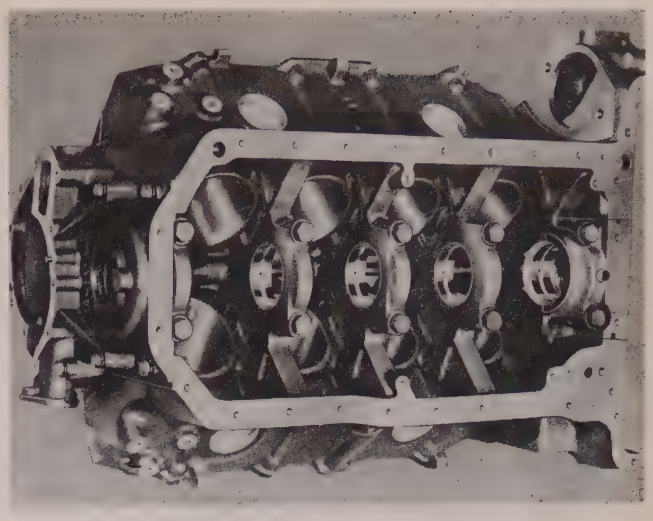


Fig. 34 – Bottom of crankcase

Lubricating System

As automobile engines become more highly developed the demands on the lubricating system have become more exacting. Closer clearances, thinner babbit overlays, higher power outputs, and higher unit loads have all increased this problem. Hydraulic lifters, which have a pronounced sensitivity to dirt, air bubbles, varnish, and sludge, have not reduced the engine designer's difficulties, although the oil companies have done much to relieve the situation with improvements in oils and additives.

In designing the new Buick engine every effort was made to improve the lubricating system (Fig. 36). The oil pan is deep, with the oil screen near the bottom to provide the maximum depth of oil to minimize splashing and foaming, and to insure a constant supply of oil even during fast acceleration or turns. The fixed screen has an area of 19 sq in. to reduce the inlet flow velocity and the tendency to carry foreign material through the screen.

A horizontal baffle extends over the entire sump area and reduces aeration of the oil by preventing crankshaft oil fling-off from churning the sump oil, and oil in the sump from being thrown against the crankshaft. The oil pump is of standard Buick design and mounts on the crankcase flange rail, a position made possible by the dropped rail design. Oil is carried by drilled passages to a full-flow oil filter in which the filter element is mounted vertically, a feature which we feel certain will be greatly appreciated by anyone who has tried to change a filter element in one of the horizontal or angle mounted units.

From the filter the oil is delivered to the rear of the centrally located main oil gallery. This gallery supplies all the camshaft and main bearings, and from the grooved main bearings it feeds the connecting-rod bearings in the conventional manner.

Two other longitudinal oil galleries are used, drilled to intersect the lifter guide holes. These galleries are supplied with oil at reduced pressure through drilled passages registering with a metering groove in the front camshaft bearing. Drilled passages from the front end of these oil galleries lead to the rocker-arm assemblies.

The separation of functions of these oil galleries has several advantages. Pressure in the main gallery is not affected by lifter clearances or leakage, since the oil feed to the lifters is controlled by the camshaft groove. Oil velocities are reduced in the lifter galleries, since they feed only the lifters. The lower oil velocity and lower oil pressure aid in the elimination of air bubbles at the front end of the gallery before the oil reaches the lifters, the air being vented through the rocker-arm and shaft assemblies.

Electrical System

The present trend toward higher volumetric efficiency and higher compression ratios has cre-

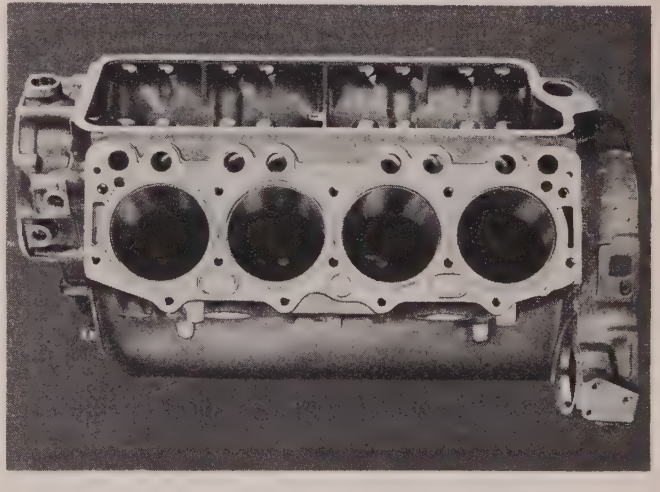


Fig. 35 – Cylinder-head gasket and crankcase

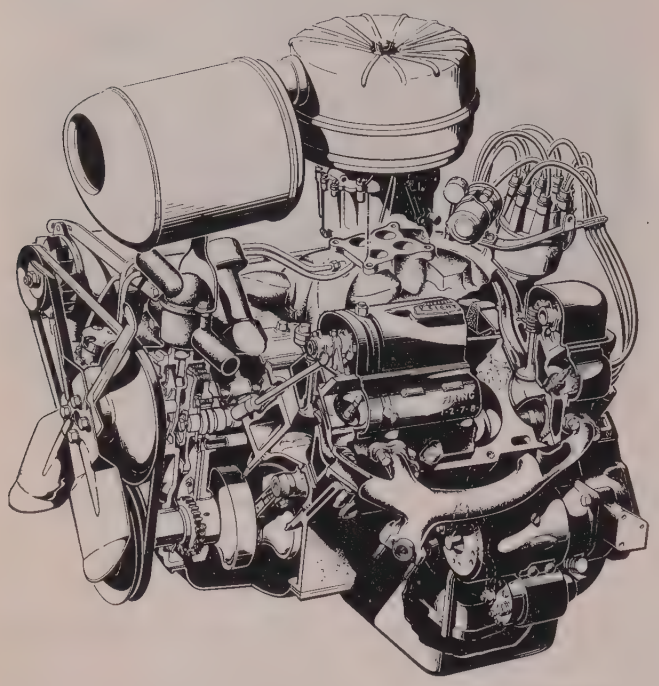


Fig. 36 – Engine lubrication

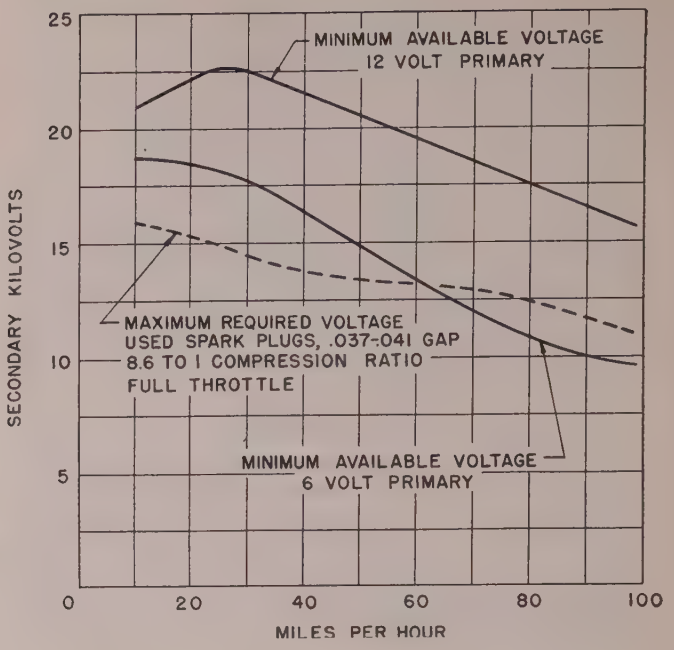


Fig. 37 – Secondary voltage comparison with full electrical load

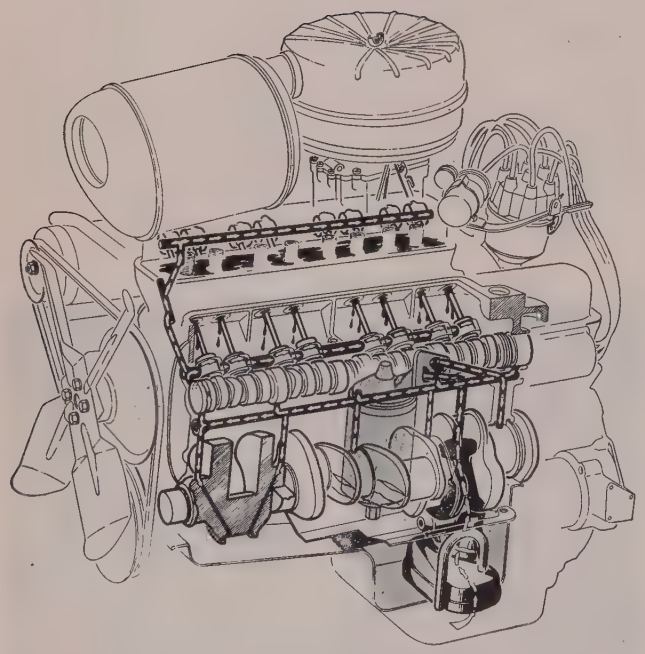


Fig. 38 – Sectional view of engine

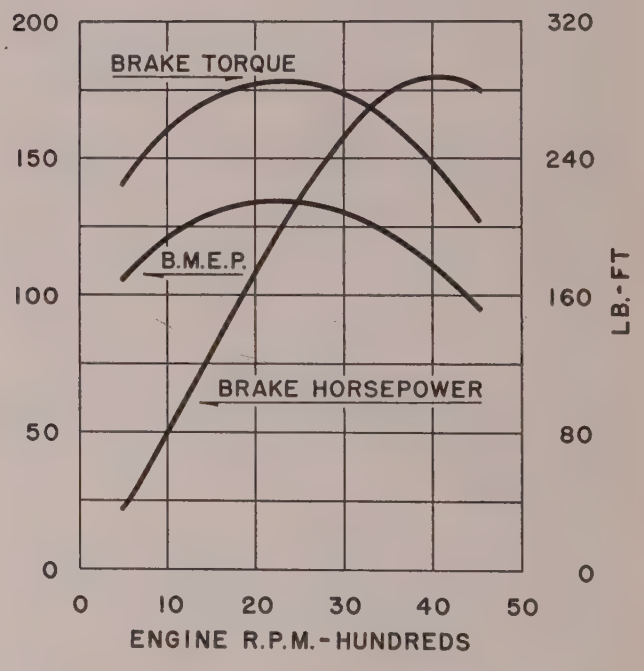


Fig. 39 – Full-throttle maximum power corrected to 100 F

ated a problem for the electrical engineers. The required ignition voltage (Fig. 37) has now become of such a magnitude that the secondary voltage obtainable with a 6-v primary system is closely approaching the minimum firing voltage of the spark plugs, even under optimum conditions.

Several attempts which have been made in an effort to increase the secondary voltage while retaining the 6-v primary system have met with at least a temporary success. However, in view of the ever-increasing secondary voltage requirements, Buick decided that the most satisfactory long range solution was the adoption of a system with a higher primary voltage, and has adopted the 12-v electrical system on the V-8 engine for 1953.

Endurance

The Buick 90-deg V-engines (Fig. 38) have undergone 10,000 hr of dynamometer testing, and over 1,000,000 miles of road testing in cars running at the General Motors Proving Ground. The usual quota of corrections have been made as faults developed, but it is of interest that the second experimental engine, assembled in the fall of 1950, ran on full-throttle high-speed endurance at 4200-4500 rpm several hours longer than any of the notably rugged straight-8 Roadmaster engines have ever run in the same standard test. Road tests have been equally satisfactory and it is safe to say that in its final form the more powerful Buick 90-deg V-8, although 170 lb lighter, is equal to or better than the Series 70 straight-8 Buick engine in durability.

Performance

Because of the current contest in advertised horsepower, we approach the subject of engine output with some misgivings. The problem of whether to try to out-exaggerate the field or to quote actual figures, which may not seem sufficiently high to some, is a difficult choice. We have tested four of the new competitive V-8 engines in our engine test department according to the General Motors test code. We have access to the results obtained on these and other makes of engines, which have been tested in other divisions of General Motors, at General Motors Research, and at the General Motors Technical Center. None of the tests on 1952 engines have shown higher bmep figures, hp ratings, or better specific fuel curves than we have obtained with our new engines under the same test conditions.

We believe the purpose of this paper is best served by quoting actual figures obtained under test No. 7 of the General Motors test code, which is run without muffler, and is corrected to 100 F carburetor air temperature (Fig. 39).

The 60 F correction factor commonly used for advertising purposes will increase the hp to 188, the maximum torque to 298 lb-ft and the bmep to 139 psi.

A gain in performance, which is often overlooked, is due to the lower engine weight, and to the lower inertia of the moving parts.

A gain in fuel economy results from the reduced engine friction, and to the higher compression ratio of the 1953 Buick V-8. Also, it should be remembered that the resulting increased efficiency is evidenced in improved performance as well as better fuel economy.

Conclusions

After we have been in production for a year we will be in a better position to state just how well we have accomplished all our aims and objectives in the design of the new Buick V-8. At the present time we believe we can safely draw the following conclusions about the new engine:

1. The Buick V-8 is as durable as our 1952 line of straight-8's, with sufficient strength and rigidity to permit increasing compression ratios and power output as advances in fuels and combustion-chamber design permit.
2. From our tests the fuel economy improvement will be up to 8%.
3. Our service department states that the Buick V-8 is the easiest of the new engines to service.
4. Our production department estimates that the production cost of the new engine will be 7% less than for the Roadmaster in-line engine. One-half of this saving is due to weight reduction.
5. To the best of our knowledge the Buick V-8 is the lightest and most compact automobile engine for its power output now in quantity production in America.

References

1. SAE Transactions, Vol. 6, October, 1952, pp. 617-636: "The Volumetric Efficiency of Four-Stroke Engines," by J. C. Livengood, A. R. Rogowski, and C. F. Taylor.
2. *SAE Journal* (Transactions), Vol. 50, June, 1942, pp. 213-220; 252: "Air Flow Through Intake Valves," by G. B. Wood, Jr., D. U. Hunter, E. S. Taylor, and C. F. Taylor.
3. SAE Transactions, Vol. 6, April, 1952, pp. 290-303: "Economic Relationship of Engine-Fuel Research," by C. L. McCuen.
4. SAE *Quarterly* Transactions, Vol. 1, October, 1947, pp. 669-679: "More Efficient Utilization of Fuels," by C. F. Kettering.

APPENDIX I

Buick Weight Comparison

General Motors Uniform Parts Classification		Series 70 1952	Series 70 1953	Difference
6A1	Cylinder block and crankcase	248.67	176.52	—72.15
6A2	Cylinder head	93.34	105.99	+12.65

6B	Flywheel housing	20.40	1.03	—19.37
6C1	Crankshaft	114.21	55.91	—58.30
6C2	Crankshaft balancer	13.68	0.00	—13.68
6C3	Flywheel	12.05	9.44	— 2.61
6D	Connecting rods, pistons, and rings	30.74	28.59	— 2.15
6E	Oil pan	13.60	9.36	— 4.24
6F1	Oil pump and drive	5.82	5.15	— 0.67
6G1	Oil distribution system	0.12	0.13	+ 0.01
6G2	Oil filler	0.26	0.50	+ 0.24
6G4	Oil gage rod	0.11	0.12	+ 0.01
6G5	Oil filter	5.09	5.20	+ 0.11
6H	Engine ventilating system	3.13	1.75	— 1.38
6J	Engine front covers	6.89	12.02	+ 5.13
6K1	Fan and drive	4.67	9.52	+ 4.85
6K2	Water pump and drive	13.26	7.66	— 5.60
6K3	Thermostat and engine cooling parts	4.15	6.16	+ 2.01
6L	Intake and exhaust manifolds, and heat controls	46.40	50.61	+ 4.21
6M1	Carburetor	9.81	10.43	+ 0.62
6M3	Air cleaner and silencer	9.89	13.11	+ 3.22
6M4	Fuel and vacuum pump	5.94	5.88	— 0.06
6Q	Powerplant mountings	4.65	4.55	— 0.10
6X1	Camshaft and drive, valve springs, and lifters	48.54	31.47	—17.07
6X3	Valve rocker arms, shafts, and covers	16.95	13.75	— 3.20
6Y1	Generator	24.85	22.24	— 2.61
6Y2	Starting motor and control	25.83	23.92	— 1.91
6Y3	Distributor	6.13	6.43	+ 0.30
6Y4	Spark plugs, ignition coil, and wires	4.93	6.49	+ 1.56
	Total engine dry	794.11	623.93	—170.18

APPENDIX II

1953 Buick V-8 General Specifications

Bore, in.	4.0
Stroke, in.	3.2
Displacement, cu in.	322
Numbering System, Front to Rear	
Left Bank	2-4-6-8
Right Bank	1-3-5-7
Firing Order	1-2-7-8-4-5-6-3
Taxable Horsepower	51.2
Compression Ratio	8.5/1
Maximum Brake Horsepower, Corrected to 100 F	180 at 4000 rpm
Maximum Brake Torque, Corrected to 100 F, ft-lb	285 at 2200 rpm
Piston, Rings, and Pin	
Piston Material	Aluminum alloy
Features of Piston	Cam ground, full skirt, transverse slot, anodized
Number of Compression Rings	2
Width of Compression Ring	0.078
Number of Oil Rings	1
Width of Oil Ring	0.187
Piston-Pin Diameter and Length	0.94 x 3.40
Piston-Pin Locking Method	Clamped in rod
Crankshaft	
Material	Forged steel
Weight, lb	55.91
Bearing Taking Thrust Load	No. 5
Number of Main Bearings	5
Main Bearing Journal Diameter and Effective Length, in.	
No. 1	2.4985 x 1.220
No. 2, 3, and 4	2.4985 x 1.250
No. 5	2.4985 x 1.765
Crankshaft Main Bearing Material	Durex 100A
Crankpin Journal Diameter, in.	2.2495
Connecting Rod	
Length, Center-to-Center, in.	6.00
Connecting-Rod Bearing Material	Durex 100A
Effective Connecting-Rod Bearing Length	0.881
Camshaft	
Material	Forged steel
Type of Drive	Chain
Number of Bearings	5
Valves and Operating Mechanism	
Type of Lifters	Hydraulic
Rocker-Arm Ratio	1.5/1
Valve-Seat Angle, deg	45
Spring Pressure, lb	
Valve Closed — Outer	37.5–42.5
Valve Closed — Inner	19.5–24.5
Valve Open — Outer	85–91
Valve Open — Inner	53–59
Valve lift, in. — Inlet	0.378
Valve lift, in. — Exhaust	0.350
Valve Head Diameter — Inlet	1.750
Valve Head Diameter — Exhaust	1.250
Valve Stem Diameter — Inlet	0.3720
Valve Stem Diameter — Exhaust	0.3714
Valve Timing	
Inlet Opens, deg btc	25
Inlet Closes, deg abc	77
Exhaust Opens, deg bbc	70
Exhaust Closes, deg atc	42
Timing Point	Valve 0.004 in. off seat
Lubrication	
Type of Lubrication	Full pressure
Oil Pump Type	Gear
Oil pressure, Maximum, psi	40
Type of Oil Intake	Stationary
Oil Filter Type	Full flow
Capacity of Crankcase, Less Filter, qt	6

DISCUSSION

Development of New V-8 Combustion Chamber Reviewed

– D. F. Caris and F. A. Wyczalek
Research Laboratories Division, GMC

THE authors have presented an excellent introduction to the features of the new Buick engine. We are glad to discuss our part in the development of the combustion chamber for this engine.

Fig. 15 shows design details of the head and piston. As pointed out by the authors, this design has an intake and exhaust valve at one side of the combustion chamber in a plane 45 deg to the normal. This provides an easy entrance into the combustion chamber and results in high volumetric efficiency. In addition, the flat 45 deg portion of the piston comes within 0.080 in. of the surface of the combustion chamber, resulting in turbulence during the compression stroke. Another important feature of this combustion-chamber configuration is the location of the spark plug. As the authors have pointed out, it is within less than ⅛ in. of the centerline of the cylinder bore. This obviously is a location which results in minimum possible flame travel length.

In order to better appreciate the various features which are incorporated in the design of the new combustion chamber, it might be of interest to review some of the laboratory's earlier work. For many years, this laboratory has directed much of its effort toward improving the efficiency of internal-combustion engines. This work and, of course, the work of a great many others in both the automotive and petroleum industries, has resulted in a continuous improvement in power and economy over the past half century. In 1947 C. F. Kettering pointed the way to further progress in engine efficiency by demonstrating that compression ratios as high as 12.5/1 are practical in automotive engines if sufficient consideration is given to design of the engine structure. Mr. Kettering also indicated that there are at least two ways of achieving higher compression ratios: (1) By means of chemical octane numbers built into the fuel at the refinery by the petroleum technologists. (2) By means of mechanical octane numbers built into the engine by the engine designers.

The petroleum industry has an enviable record of progress in improving the antiknock quality of gasoline over the years, and we have no reason to believe that they will not continue this trend as long as they find it economical to do so.

There are many factors which contribute to an engine's mechanical octane rating or its ability to operate more efficiently on fuel of a given octane level. They include spark advance, carburetion, valve timing, volumetric efficiency, and combustion-chamber design. These factors are all interrelated and have a bearing on the selection of the compression ratio at which the engine will operate. The relation between these factors in the final engine represents the best compromise the designer can make, in view of the wide range of operating requirements which the automotive engine must meet. Of these many factors which influence the compression ratio at which the engine operates, we found that the combustion chamber offered a very promising field for development.

Since the octane requirement of an engine is affected by the weather, in addition to all these other factors, the first step in the evaluation of various combustion-chamber designs, in terms of octane requirements, was to develop a procedure which would allow accurate reproduction of engine operating conditions over a long period of time. Fig. A shows some of the apparatus which was used to control the variables which influence octane requirements. As an example, the humidity, pressure, and temperature, were held to definite values so that variations in the air entering the carburetor would not influence the tests. Such mechanical variables as valve timing, cam design, valve sizes, and compression ratio were also held at definite values.

It was decided to maintain a constant compression ratio of 9/1 for this combustion-chamber development for two reasons:

1. To simplify evaluation of the various designs.
2. Our past experience in this type of work has indicated that we should be able to achieve 9/1 compression ratio with present commercial gasolines.

In addition, the air/fuel ratio was adjusted for maximum knock for each fuel used during the tests. Standard reference blends of isooctane and normal heptane were used to determine the octane requirement of the various combustion chambers. However, since engines are not operated in the field on reference fuels, ratings were also obtained using commercial regular and premium grade gasolines. The present rating of the regular grade gasoline used for the Buick combustion-chamber development is 86 octane research and 80 octane motor, while premium grade now has a rating of 92 research and 82 motor.

Finally, to eliminate the deposit variable, all the designs were rated clean, even though it is a recognized fact that commercial engines operated in the field always have a certain amount of combustion-chamber deposits present. Of course, this factor must be taken into account in selecting a compression ratio for the production engine and, naturally, this has been done.

It should be quite apparent that all this attention to detail demonstrates the importance of taking every possible precaution to make sure that, insofar as is possible, combustion-chamber design was the only variable.

An example of a typical fuel rating test of a combustion chamber is shown by Fig. B. The shape of the chamber is shown by the scale cross-sections. At any given speed, the procedure consists of first obtaining a spark fishhook. This is a plot of power as expressed in terms of indicated mean effective pressure versus spark advance. The spark fishhook is obtained using a fuel which will permit the engine to develop its maximum imep without knock which, of course, indicates the maximum possible power output of the design under detonation-free conditions. Next, the borderline-knock spark advance values of primary reference blends and commercial gasolines are obtained. As an illustration, 80-octane reference fuel has a border-line spark advance of 5 deg and the premium grade gasoline used in these tests has a borderline spark advance of 13 deg. It can be observed that these values are superimposed upon the spark fishhook. When combined in the manner shown by Fig. B, these data can be interpreted by the observer as he desires. As an example, it can be seen that it requires about 92.5-octane fuel for this chamber to operate detonation free at 99% of maximum power. It can also be seen that premium gasoline will permit operation at almost 99% maximum power. Another conclusion that can be drawn from this chart is that regular gasoline, as rated by this combustion chamber, is equivalent to approximately 86-octane fuel.

Since octane requirement depends upon power output when other factors are held constant, it is obviously very important to obtain power measurements in addition to

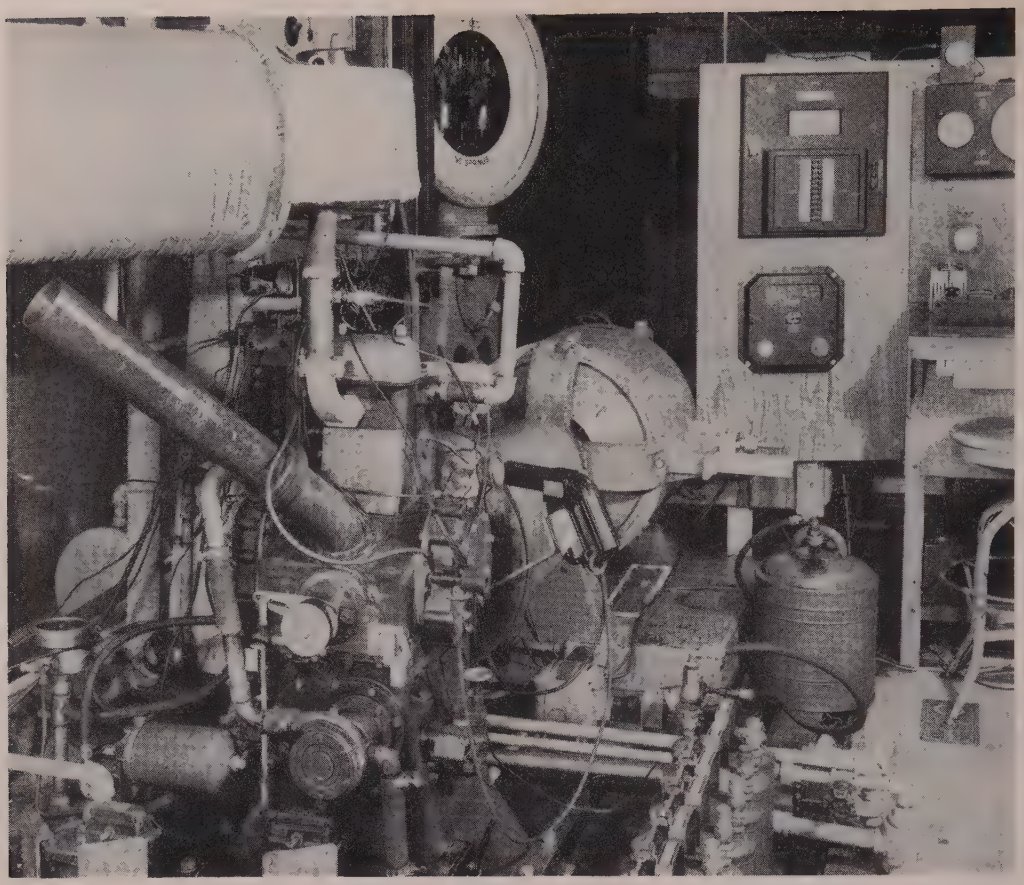


Fig. A – Apparatus used to control variables which influence octane requirements

the fuel ratings. We think this type of presentation is extremely valuable since it presents important factors such as gasoline rating, octane rating, and power measurement on one sheet, where they can be conveniently examined.

Similar tests for each chamber design evaluated were performed and the results are summarized as shown by Fig. C. We refer to Fig. C as an octane tree. In this case, it was made up by taking the octane rating for all designs at 99% of maximum power, 1000 rpm, 9/1 compression ratio, and arranging them along a vertical octane number scale. In addition to the octane number arranged in the vertical direction, the research ratings of regular and premium gasolines are also labeled. As can be observed, these ratings are 86 octane for regular and 92 for premium fuel.

The various combustion chambers are identified by the scale cross-sections of each design. As an example, design A shows cross-sections in two planes through the combustion chamber. As has already been mentioned, the valves are located in the 45 deg plane. The spark plug is located in the center of the engine cylinder. It can be seen that **almost** all changes in combustion-chamber shape for this particular investigation were made by simply changing the configuration of the pistons. It is apparent that this Buick head design with central spark plug location lends itself very well to a study of this nature. It also possesses the added advantage that future increases in compression ration can be achieved by a relatively simple piston change.

More detailed examination of this figure shows that chamber A, the original design for the Buick engine, requires 96½-octane fuel to operate at 99% output; while chamber J, the design selected for the production engine, requires only 88½-octane fuel. This reduction in octane requirement of eight numbers is a fine example of a mechanical octane improvement. Other designs can be selected and examined in a similar manner. As an example, design F, a dome-shaped combustion chamber that was introduced by Buick back in 1934 shows an octane requirement of 95½. This, as was pointed out by Mr. Turlay in his presentation, was the best that Buick had available at that time, but was abandoned in 1938 in favor of the turbulator piston which is shown as design G with a requirement of 93 octane. This, of course, allowed the compression ratio to be increased while still using the same fuel.

These two early designs were revived and modified for testing at 9/1 compression ratio to see how they would fit into the overall picture.

One of the questions which may come to mind at this point is: what are the factors which the designer must build into a combustion chamber in order to obtain mechanical octanes?

Our past experience had indicated that if we could shorten the combustion process the charge would complete burning before detonation had time to occur. Therefore, at the start of this development we **deliberately** set out to speed up the rate of burn, by designing chambers with high turbulence and the shortest possible flame travel.

Referring back to Fig. C and comparing designs A and J; it can be observed that chamber J has a much shorter flame travel and, because of its more effective piston coverage, has much greater compressive turbulence than design A.

The fact that shorter combustion time is accompanied by a decrease in octane requirement can be demonstrated by comparing the spark-advance values shown by Fig. D. This chart compares the test results obtained for the original design (A) and the final production design (J). It can be seen that the spark advance value at 99% maximum power for chamber A is 20 deg and for design J it is 13 deg. This represents a decrease in burn time of at least 7 deg.

As an additional illustration of the effectiveness of short flame travel in reducing octane requirements, Fig. E shows,

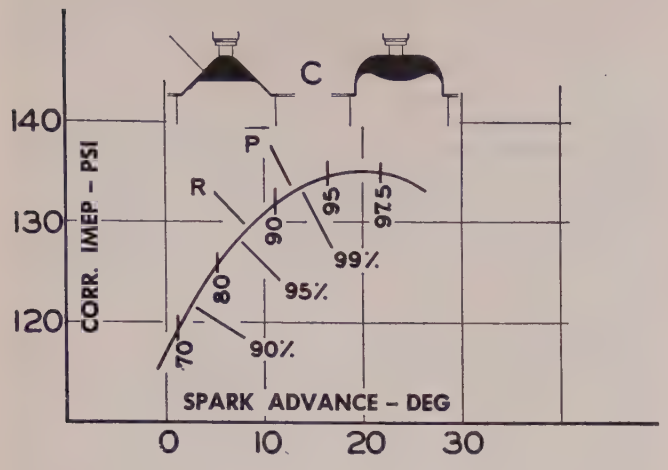


Fig. B – Fuel rating test of combustion chamber – 1000 rpm and 9/1 compression ratio

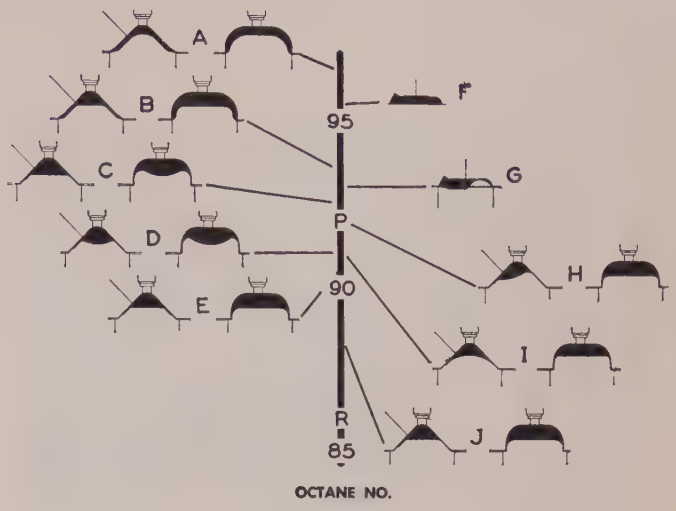


Fig. C – Single-cylinder octane requirements at 99% maximum power, 1000 rpm, and 9/1 compression ratio

the effects of two different spark-plug positions in the same chamber. By comparing the octane requirements at 99% maximum power, which is approximately 92 octane for the central plug position and about 98.5 octane for the point of ignition on the exhaust valve side of the head, it is apparent that Buick's central plug position alone accounts for a 6.5 octane improvement. Further examination shows that at 95% maximum power, which is more in line with the usual production spark settings selected by most manufacturers, Buick's central plug location reduces the octane requirement from 94 to 79 octane, an improvement of 15 numbers in this range of the octane scale. In addition, it can be observed by comparing the 99% maximum power spark advance values that there is a decrease in combustion time of at least 14 deg for the central plug position. That is, from 24 deg to 10 deg.

Another interesting question which may be asked in connection with combustion chamber design is, what effect does it have on power output and thermal efficiency?

As shown by Fig. F, we have compared the imep and indicated thermal efficiency values obtained for the combustion chambers included in the Buick development with the same valve and bore dimensions. We consider the differences shown to be nothing more than random variations within the limits of error of measurement. Therefore, we can conclude that, within the scope of the Buick program, combustion-chamber shape has shown no significant effect on power or thermal efficiency.

In addition to the Buick development program, we have carried on an extensive investigation of present commercial combustion chambers as well as a great many experimental chambers for overhead-valve engines. Our tests have shown no significant difference in thermal efficiency between any of them. However, many have exhibited a wide spread in mechanical octane characteristics, a feature which can be exploited to improve efficiency through the medium of increased compression ratio.

In summarizing this discussion, we have shown that the

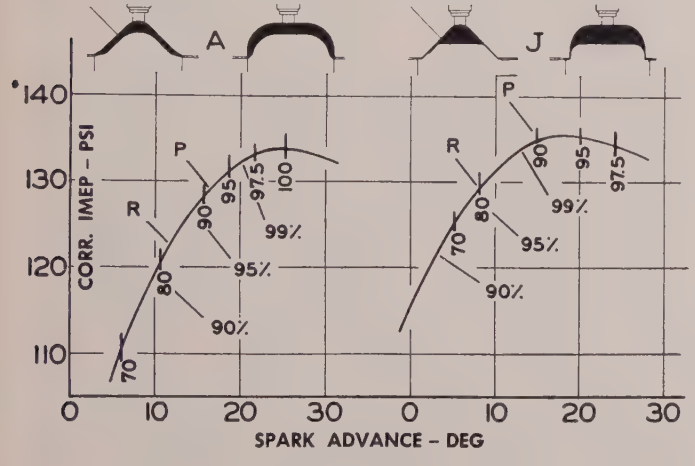


Fig. D – Comparison of test results for original design (A) and final production design (J)

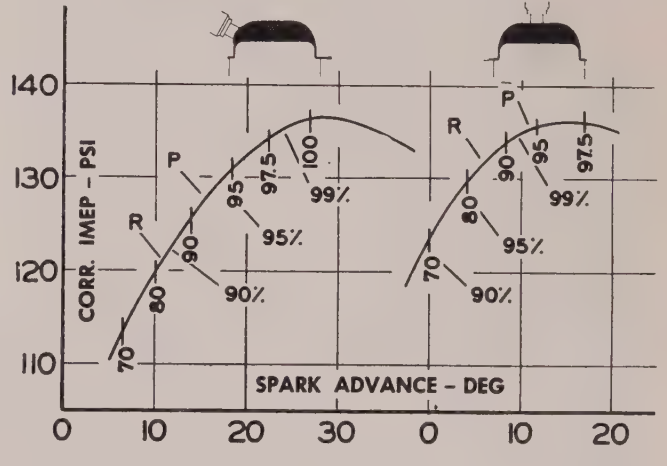


Fig. E – Effects of two different spark-plug positions in same chamber

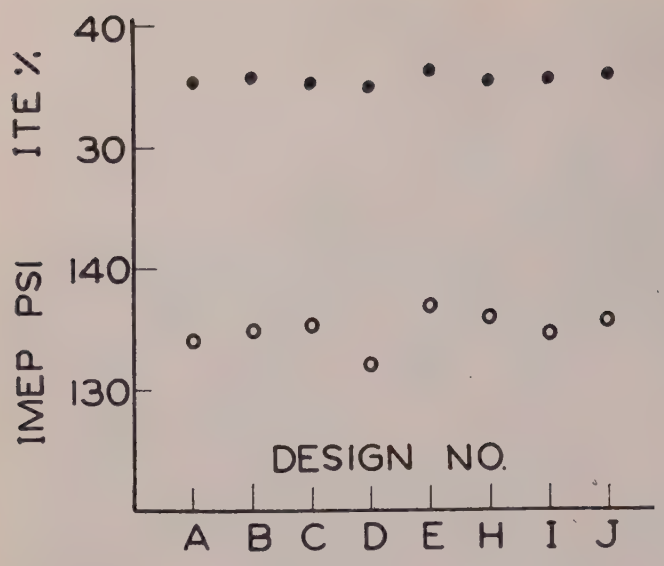


Fig. F – Effect of combustion-chamber shape upon combustion efficiency

combustion chamber for the new engine is the result of a systematic mechanical octane development program. A very realistic gage of the extent to which this program has paid off is demonstrated by the ability of the production engines to operate at 8.5 compression ratio on premium gasoline.

New Engine Elicits Pertinent Questions

– J. R. MacGregor
California Research Corp.

THE authors have performed a herculean task in describing the numerous and intriguing design changes incorporated in the new Buick engine. Not being an engine designer and not having had any experience with this particular engine, it would be presumptuous for me to discuss any of the mechanical details from the design standpoint. However, based on experience with other engines there are several questions and comments that may be pertinent.

The presentation of the background for the valve-timing diagram makes no mention of the modest change in the timing of the events. Is this change simply a byproduct of the modification in the lift diagram, or is it an indication of a desire to fit more nearly the requirements imposed by the hydraulic type of transmission rather than the syncromesh?

The combustion-chamber design is interesting, to say the least. It would seem that the necessity of keeping deposits from the small clearance areas would be very great if interference is to be avoided and quenching retained. Has experience shown that the turbulence is adequate to scour these areas or is it anticipated that other means will be required for periodically removing carbon deposits?

Extensive experience in the removal of deposits from the combustion chambers has shown that not only can the octane requirement of an engine be materially reduced but also an incidental and almost equally important return of lost power can be obtained. The magnitude of this power recovery is dependent upon the amount and type of deposits and the design of the combustion chamber. The power recovery has varied from practically nothing to approximately 16 per cent. A cursory examination of the new Buick combustion chamber suggests that power loss through the laying down of deposits may be rather great. Has any information been obtained on this subject that is available for release?

In view of the influence that breather gases have on the deposits found on the outside of the carburetor and within the induction system it would seem that trouble may be experienced from this cause during those extensive periods of idling sometimes encountered when thermal forces cause the gases to move in the opposite direction from those shown by the arrows in the diagrams. This item will bear close watching.

With reference to the drive for the distributor it would appear that the arrangement which causes friction to oppose gravity might introduce a serious fluttering of the ignition with its attendant undesirable effect on octane requirement. Has some precaution been taken to avoid any uncontrolled vertical play of the distributor shaft?

The lubrication requirements of modern engines have been generally becoming more exacting and it would appear that those of the new Buick engine may even be critical. The large diameter piston with a crown design that suggests high heat transfer both by radiation and conduction; the short stroke which tends to permit more oil throw-off to reach the piston and its crown; and the short water jackets all seem to indicate high crankcase oil temperatures. In addition, the crossover of the exhaust pipe would appear to contribute to the elevation of oil temperature. It would be appreciated if advance information can be given as to the lubricating oil temperatures that are to be expected, particularly during extended high-speed driving under desert conditions.

The manner used to present the power data is very interesting and we shall look forward with great enthusiasm to the acquisition of one of these cars for chassis dynamometer and road test purposes. This will permit the establishment of our own views on power and some of the other operational features that so importantly influence the motorist in his evaluation of both the engine and the petroleum products he uses.

Fuels and Lubricants Sought for New Designs

– D. P. Barnard
Standard Oil Co. (Ind.)

DISCUSSION of this paper is difficult for an oil man, particularly as **this** oil man has had no experience in the design or the development of mechanical devices – including any kind of an engine. These remarks endeavor to record those first impressions which the paper makes on such a representative of the petroleum industry.

First, adherence to overhead valves for 50 years. This is most reassuring to the layman as it dramatizes the extreme reluctance with which changes have always been contemplated in a perfect product. It not only helps to know that the familiar overhead valves are still with us, but it adds to our feeling of security to realize that, throughout the years since Louis Chevrolet laid down the first Model 10, the valves have remained virtuously upright. The cylinders may lean to the right or left, but the valves – no. (I hope someone can point out the significance of this observation over and above its relation to the installation problem.)

The new design aims at the best possible utilization of the improvements which have been made in fuels and

lubricants. By taking advantage of all of the know-how currently available, this utilization will be far ahead of anything that could be accomplished by modification of the earlier design. We recognize other advantages (which are so clearly pointed out in the paper), but from our side of the table we are particularly interested in what the new job will do with and to our products.

The compression ratio of 8.5 fills us at once with admiration and concern: admiration for the strides in engine development which justify placing such a compression ratio in the hands of the general public; concern over whether the petroleum products which our industry makes – and will be able to make during the foreseeable future – will be entirely satisfactory. To us, the $64 question is "how is it done?"

The fuel manufacturer is especially interested in the effects of deposit allowances in holding the allowable compression ratio down, and the rather high maximum torque speed in permitting such a high compression ratio on commercial premium fuels. Our misgivings do not reflect defeat but merely recognize troubles as an inevitable part of the price of progress. Every increase in compression ratio brings up pictures of more combustion-chamber deposit problems. Of course, we realize that the modern engine at its compression ratio is less susceptible than its ancestors of a generation or more ago. Nevertheless, even the most modern gasoline engine design is, as was once pointed out by Bill James, wanting in ash-handling capacity. This will always worry us, since there is no practical way of achieving present octane number levels – and increasing octane numbers still further – without at the same time accepting necessity for ash handling. And not all combustion-chamber deposit difficulties are chargeable to the fuel or even to the lubricating oil. Some of this material is just plain adventitious dirt.

Recognizing that we will never be completely free from combustion-chamber deposits, what to do about them is likely to remain a major problem. The new designs with which we are already familiar have contributed much toward minimizing their effects, and we hope that the new Buick design will extend this know-how. In the meantime, we think that we have been able to help some and, just as the engine builder keeps seeking to improve his designs, we in the oil industry well keep trying to improve our products.

The quest for higher octane numbers has raised levels to the place where the going is getting difficult. Current premium grade octane numbers imply a marked reduction in the number of hydrocarbons available to the fuel manufacturer. Each octane number increase is reflected in the necessity for more complicated procedures, higher costs, and the inevitable processing losses than was the case when levels were appreciably lower. Of course, a 1-unit increase in the range above 90 is worth more than a similar increase in, say, the 70's, but the costs per octane number go up much more rapidly, too, and the inevitable implications must not be disregarded. Lest some of us become too impatient to achieve the magic level of 100 let it be noted that the effective knock-rating improvement in going from present average levels to 100 is at least as great as the sum of all octane improvements made up to the present.

We in the petroleum industry are most appreciative of the frank, accurate, and complete presentations in this and other papers describing the new engine designs. We may not fully understand the many considerations underlying these new jobs, but at least we can welcome the clearly competent efforts to base them on the best information available in order to deliver the most performance and service, with the greatest of reliability, and within the practical limits of economics. In the meantime, we in the oil industry should gladly invoke all of the facilities available to us in our efforts to provide these new designs, together with their ancestors, with fuels and lubricants.

Cylinder Performance in New Engine Discussed

– S. D. Heron
Consulting Engineer

THIS paper is of decided interest from the standpoint of cylinder performance. The Ethyl Corp. has for several years studied cylinder behavior in respect to knock and preignition-limited performance and also in regard to fuel economy and air utilization. The knock-limited behavior of the new Buick engine is obviously very good since it uses a compression ratio of 8.5/1 with premium fuel, and does so with excellent volumetric efficiency. In appraising knock-limited performance, volumetric efficiency must be considered as well as compression ratio. In this connection Socony-Vacuum laboratories showed, several years ago, that a passenger car of about 7/1 compression ratio would, at road load, operate without knock on zero octane number fuel up to about 60 mph.

The combustion-chamber design of the new engine is of distinct interest to the student of cylinder performance. This chamber enables a very short stroke to be used while maintaining a compact shape of the charge at the point of ignition. With a high compression ratio and a short stroke it is difficult, with some combustion-chamber layouts, to avoid layers of charge which are difficult to ignite satisfactorily at light load.

The new chamber permits the use of a very considerable quench area without interference with the flow path of the gas leaving the intake valve. With overhead valves, a large quench area often involves considerable interference with the gas leaving the intake valve. The authors appear to believe that the combustion benefits of quench area, as they have used it, are a result of turbulence rather than due to quench.

The term, quench area, is very generally used in this country in contrast to the term, squish, which is generally used in Europe. It is really unimportant which term is used since either term is descriptive and generally understood. The term quench does, however, imply that the combustion effect is known. Some two years ago J. B. Macauley initiated a study to determine if quench area produced beneficial effects as a result of quench or as a result of turbulence. One phase of the study came up with a quite clear answer to the effect that the benefits were due to turbulence. Another phase, conducted under conditions where quench area produced marked improvement in knock-limited performance, gave no clues as to whether quench, turbulence, or any other cause, was responsible for the improvement.

The Ethyl Corp. studies of cylinder performance are, it is hoped, without prejudice since we do not design combustion chambers except for experimental studies. The types designed have mostly been rather bad although this, in part, has been due to trying to cover too wide a range. Attempting to cover wide ranges of compression ratio and turbulence in one cylinder head often produces a result analagous to trying to produce a musical instrument which will give the combined effects of a bassoon and a violin.

These studies of knock-limited performance of necessity are based upon somewhat limited data and further data might revise some of the present conclusions which are rather destructive of some currently accepted ideas. For instance, it appears that the knock-limited performance of very small cylinders is no better than that of those of 6-in. bore. Also, it appears that current large bore aircooled aircraft engines are just as good at 3000 rpm as are very small water-cooled cylinders at the same speed. The studies have covered a fuel range from zero to above 100 octane number and while academic in many respects do lead to the conclusion that one is liable to be the victim of one's preconceived notions.

Blowby Characteristics Of Engine Requested

– T. A. Scherger
Studebaker Corp.

THE authors have covered their subject so concisely and thoroughly that it is quite difficult to make any constructive additions to the material presented. In general the engine appears to be of rugged, stiff construction, with ample displacement to perform the assigned task. With the short stroke and high overlap journals the crankshaft should be very stiff and should contribute toward smooth engine operation. In mentioning the balance it was stated that a small amount of counterweighting is carried in the crankshaft pulley and in the flywheel. This brings up the question as to the method of balancing. Is the crankshaft balanced in production with the flywheel and crankshaft pulley assembled? If this is the case, what provision is made for servicing the crankshaft pulley in case it is damaged in the field? Doesn't this type of balancing give somewhat higher bearing loads on the front and rear bearings or do the authors feel that with the high overlap of the bearing journals this is not a factor to be considered.

I note that the compression ratio of this engine is 8.5/1. No mention was made of the spark advance. I am wondering whether the spark advance as used is for maximum torque at relatively low speeds or whether some sacrifice in full-throttle power was made for octane numbers, thereby obtaining the benefit of high compression ratio for part-throttle fuel economy.

The ratio of sealing volume per cubic inch of piston displacement is unusually high for this engine. No mention of blowby was made. If the data is available, what are the blowby characteristics of this engine?

New Engine Designs Reflected In Future Gasoline Picture

– A. J. Blackwood
Standard Oil Development Co.

I AM pleased to have this occasion to discuss the paper on the Buick engine, for although I have been in the midst of a petroleum technology for many years, I spent my first 18 months out of college doing engine research and development work with the Mack truck people. Consequently, I found the paper particularly interesting. For an engineer who has been away from the details of engine development work for over 25 years, many of the problems discussed still have a familiar and pleasant ring. From my present position in the petroleum product research and development field, it is certainly gratifying to see some of the familiar problems being so effectively solved.

I would no longer presume to be an automotive engineer capable of offering sound comments on this paper insofar as design details are concerned. I shall, therefore, leave such comments to others who are far better qualified. I would, however, like to offer a few thoughts on three or four points directly related to the paper, and then to offer some general comments on the impact which engines such as this may have on the future gasoline picture.

I was interested in the effort made to have all combustion-chamber volumes exactly the same, and to maintain a uniform induction system velocity from carburetor to exit side of the inlet valve. This should result in uniform filling – an important consideration involved in making all cylinders have the same antiknock requirement.

Mounting the fuel pump on the timing-chain cover and in the direct fan blast seems a step in the right direction to minimize vapor lock. It would be even better to have it in a direct stream of outside air which would be some 10 to 15 deg cooler than radiator discharge air.

I was interested in the approximation of about 100 miles of road test per hour of laboratory test to bring the development to a successful conclusion. In a current research project in our own laboratories, we have been averaging about 25 miles on the road for each laboratory engine test hour. Our work, however, is not seeking to establish any mechanical endurance factors.

The anticipated gain in fuel economy of 8 per cent is most praiseworthy. Part is attributed to a reduction in engine friction. We feel this friction effect on fuel economy has been overlooked in the past. In England, operators have been reluctant to use anything but heavy crankcase oils. Three or four years ago, one of Esso's technical men interested a couple of fleet operators in trying out a light oil of the 5W type. One of the benefits was a marked improvement in gasoline mileage. Now the light oil idea is spreading very rapidly in England.

We were pleased to note the special attention devoted to getting a good lubricating system on this new engine. Other features which we hope some day to find in a lubrication system are an instrument panel oil level gage; an oil pump system which compensates for wear and hot oil so that it maintains pressure; and better seals and gaskets so the garage floor stays reasonably clean.

And now I should like to discuss the fuel situation. We have made many studies of the economics of compression ratio and gasoline quality. Also, we have always been concerned with economic balance between processing losses of crude oil and added mileage to be gained by higher octane fuels and higher compression ratio engines and have recently made a further study of this point. In 1925, a 60-octane gasoline was typical, while by 1952 the octane increase over the years had amounted to about 30 points. During this time both the volume demand and percentage required from crude increased. This increase was obtained by thermal and catalytic cracking of higher boiling fractions. The processing losses from cracking are chargeable to the required increase in volume. Beyond this, however, there have been processing losses chargeable to octane increase alone. These result from applying thermal reforming and catalytic reforming to a small fraction of the lowest gasoline octane constituent, namely heavy naphtha.

On the basis of the requirement in 1952 of 39 per cent gasoline, 34 per cent distillate, 14 per cent residual fuel, and 17 per cent lubricating oil, asphalt, wax, and fuel for refinery processing, and, in the process of making a leaded pool gasoline of 90 research octane number, the losses due to reforming amount to only 2.4 per cent on the total gasoline produced.

This however is not the important point. Based on the increased efficiency resulting from the higher compression ratios which could be used with the 90 research octane number gasoline available in 1952 as compared to the limited compression ratio to operate on the 60-octane gasoline of 1925, it was calculated that about 67 per cent more work could be obtained from a barrel of gasoline of the higher octane number. Adjusting for the 2.4 per cent gasoline loss, and if we take full credit for the entire octane number increase from 60 to 90, the net gain is about 65 per cent. In other words, 2 gal today gives roughly the mileage of 3 gal in 1925 as a result of a better quality gasoline and engine performance. It seems clear to us that over the past 30 years the efforts to increase octane number and the higher compression ratios which such fuels permit, have been constructive from the standpoint of conservation of petroleum resources.

Now what about the future? Another comparison was made between the 1952 assumed 90 research octane number leaded pool and a future pool of 96 research octane number. In this second study, production of products other than gasoline was held at 1952 levels. Also, in this case catalytic reforming replaced thermal reforming and more of the low octane virgin naphtha was catalytically

reformed in the 96-octane number case. The final comparison is on a slightly different basis than in the first study. It was determined that with high compression ratio engines designed for 96 research octane number, it would be necessary to process 98 barrels of crude to get the same amount of work as is obtained from 100 barrels of crude processed for 90 research octane number gasoline. In other words, less crude would be necessary, less gasoline would be made, but the work obtained from the higher compression ratio engines would be the same. There is still room in the future for improved gasoline quality and engine adjustment that will take advantage thereof.

In my closing remarks I would like to observe that both the engine manufacturers and the petroleum industry have done an excellent job of giving the public more economical and better performing engines over the years and that trend will no doubt continue. However, it should be recognized that any improvements in economy which can be gained by better so-called mechanical octanes, through better piston design, better valves, better induction systems, better cooling systems, and so forth, provide improved economy even at the current octane number levels. On the other hand, since there is a 40-million pool of cars already on the road, and because the petroleum industry cannot make octane increases to satisfy specific cars which have higher octane requirements, the economic advantage is going to be slower if we rely entirely on chemical octane improvement.

Seeks Additional Data On Fuel Requirements

– C. J. Livingstone
Gulf Oil Corp.

AMONG the many features of the new Buick V-8 engine, I am impressed by the authors' report of their evaluation of the engine. I am sure the severity of their testing has been adequate to predict the endurance of the engine when it is delivered to the customer. However, as one who will be concerned with its fuel requirement later on, I am disappointed that the authors did not see fit to present more data on this phase. I am sure there are a good many oil people who are interested in the fuel requirements of the engine as installed with the new transmission in the automobile.

Our experience with other high-compression ratio engines indicates that the most critical fuel requirements occur when the car is operated with extremely light-load factor. In my own case, I have been driving one of the newer V-8s with 8/1 compression ratio. It has wonderful performance and I experienced practically no detonation or autoignition with present premium fuel. A few months ago it became necessary for my wife to take over the driving of this car for a period of 90 days. Before the three months was up the engine was detonating on premium fuel and was in trouble from autoignition. Consequently, I am more interested in fuel requirement data when the car is driven as my wife would drive it, than when it is put through the vigorous paces by the test engineers.

When an engine gets into trouble from detonation with premium fuels we can correct the trouble by pulling back the spark even though we may have to sacrifice performance, but when we encounter autoignition there seems to be no solution at present. Consequently, I believe the oil people would be interested in any data in fuel requirements after deposit accumulation at light load, that the authors might be able to present. We would also be interested in learning how the new engine transmission combination rates fuel in respect to the research octane number. If the new engine at 8.5/1 compression ratio can be operated without objectionable knock with present-day premium fuels, and is free from autoignition after light-load deposit accumulation, Buick has done a remarkable job in the development of this engine.

Fuel and Combustion Reactions Considered

– E. F. Miller
Socony-Vacuum Oil Co., Inc.

THE authors have centered their paper largely around mechanical design and construction, and while we are interested in many of these features, we will comment only on those items which relate to fuel requirements and combustion.

From a fuel requirement standpoint the outstanding feature of the new design is the combustion chamber. We are glad to see the amount of effort being devoted to combustion-chamber shape, on the part of the various engine manufacturers.

We recognize that turbulence and spark-plug location and quench areas influence the combustion process. The design in this case must produce considerable turbulence.

Do the authors attribute the ability to achieve 8.5/1 compression ratio on current premium gasoline largely to the combustion-chamber design or is some of this increase a result of spark advance compromises? Any quantitative indication would be of considerable interest.

Some of our experience shows that turbulence exerts some control on the formation of combustion-chamber deposits and we wish to ask whether, in the course of their development work, the authors developed information of this type.

Because of the various design influences, any observations on our part as to how well this design is satisfied by premium grade gasoline must await our experience with the cars in the hands of customers.

By now I believe most of us are familiar with the fact that cars of a given make and model usually differ in octane number requirement by 15 to 20 units as a result of driving habits, and mechanical, maintenance, and deposit variations.

The importance of this 15- to 20-octane-number variation in requirement can be appreciated when it is compared with the difference in quality between housebrand and premium gasolines which ranges from 5 to 10 octane numbers at the present time.

The machining of combustion chambers we assume is an attempt to minimize one of these mechanical variations. Any reduction in the variation of requirements would constitute a distinct step forward.

While the paper is not entirely clear to us in its reference to the extensions of the hot spots to the carburetor riser, we assume this was done for the purpose of controlling carburetor icing.

While we believe the application of heat is the most effective means of controlling icing, we recognize that the amount of heat to be applied is a difficult choice for the designer. Too high a temperature in the region of the carburetor may produce objectionable deposits.

Another factor of the design related to combustion that interested us was the stroke/bore ratio of 0.8. From our studies of this design factor, we have learned that the improvement in engine efficiency of some of the new engine designs results from the compactness of the combustion chamber in reducing heat losses.

While we have no reason to believe that going under one in the stroke/bore ratio by a modest amount hurts engine efficiency, we became worried, however, when the authors said, ". . . space requirements, as dictated by the body stylist, are one of the most important, if not the most im-

portant factor, in determining the future trend of engine design."

In a second place the authors say, "the short stroke – big bore adopted in all the new V-8 engines has many real advantages, but the stroke/bore ratio of the new Buick engine was chosen chiefly because from our investigation it produced the minimum external engine size."

We hesitate to hold the authors rigidly to these statements but we had not realized that style controls such fundamental engine design items as stroke/bore ratio.

We long since have learned in this country that engineers are required to make design compromises in the interest of making a product more attractive and therefore more salable to the public, but we hope there is a limit beyond which we as engineers will not go, for fundamentally, in this instance, we are selling transportation.

As a further question pertaining to the control styling exerts, we wish to ask whether underhood space limitations are becoming such as to restrict ventilation, thereby producing high temperatures with their adverse effect on the vapor-locking tendency of the fuel system.

Requests Additional Data On Fuel and Performance

– A. C. Sampietro
Willys-Overland Motors, Inc.

I WONDER whether the authors would be willing to supply performance data in addition to that given in Fig. 39.

Specific fuel consumption and heat-to-coolant data for both a clean engine and for a dirty engine, with the spark retarded to avoid knock, would be of pronounced interest.

Describes Fuels and Effects Of Carbon Accumulation

– Henry W. Boylan
Buick Motor Division, GMC

I WILL try to sum up all the questions that have been asked regarding the Buick combustion chamber, the fuels required for this chamber and the effect of carbon accumulation, in a general statement covering the results of our tests and a comparison with our past in-line engine.

We feel from our tests that the Buick V-8 engine will operate as satisfactorily on premium fuel as our past model in-line engine. On our carbon accumulation schedule for wide-open throttle operation, which is the equivalent to 24 hr on power runs from 600 rpm to 4500 rpm, we take a torque loss of 2½% from the clean condition at maximum torque. At peak horsepower we take a loss in torque of 3½% from the clean condition.

On our carbon-accumulation schedule which consists of 1 min at 600 rpm closed throttle, and 3 min at 1500 rpm open throttle for a 24-hr period, we have a loss in torque of 4½% at 1000 rpm and 5% at 2000 rpm. This is comparable to the loss on the in-line engine which amounts to 4½% at 2000 rpm.

Our road tests are made at the General Motors Proving Ground on their octane rating schedule. This schedule consists of entirely low-speed, part-throttle operations. Using this schedule and primary reference fuel for determining requirements, this engine requires 88.5 octane fuel for the engine clean; and a 91.5 octane fuel for the engine not clean.

During the operation of the car on the slow-speed carbon-accumulation schedule, the engine is regularly checked for autoignition. Our engines showed no sign of autoignition during this period. When rating the engine not clean for the various fuels, a slight amount of autoignition was experienced at 1800 rpm and 2000 rpm while rating Red Crown 86-80 and primary reference fuels with an octane rating of 82.5 and 85.0.

At the end of the carbon-accumulation schedule we had trace knock on the Uniontown method at 1800 and 2000 rpm – the spark timing 3 deg btc. Compression ratio will have an average buildup for the engine of ½ ratio.

The question has been asked as to how much power loss we take at the low end due to retarded spark. We take approximately a 5% loss at 600 rpm, with this loss diminishing to 1% at 2200 rpm. From this speed, and on out, the spark is set mbt (minimum for best torque). This power loss taken at the bottom end of the power curve is no different than the loss taken on most of the new engines on the market today and is particularly inconsequential in the Buick car where a Dynaflow transmission is used. The Dynaflow does not allow the engine to operate at wide-open throttle much below 1700 rpm.

In the case of the synchromesh-transmission equipped cars, it is necessary to reduce the compression ratio to 8/1 and use premium fuel to obtain satisfactory operation from a detonation standpoint. With Buick's new-type pistons and combustion-chamber design, it is necessary to maintain the design dimensions of the quench or squish area even when reducing compression ratios. For this reason, in changing from a 8½/1 ratio to a 8/1 ratio it is necessary to install new pistons rather than just increase the gasket thickness.

Mr. Scherger asked a question on blowby about which I did not quite understand. I can say this – the rate of blowby in cubic feet per minute on the new V-8 engine is not any more, and averages slightly less, than our past in-line engine with a comparable displacement.

Authors' Closure To Discussion

In reply to Mr. MacGregor – The distributor used in the new engine is of the same general construction as used in past production models, in which the thrust is also upward. The lower distributor bushing is designed to take the thrust load. We believe this construction results in less possibility for variations in end clearance than with the same of the other designs in which the thrust is downward and is carried on a bearing in the crankcase.

In reply to Mr. MacGregor and Dr. Barnard – Buick has used a high-speed timing for some time, as was stated in the paper. Such a timing, compared to more conventional timing, gains at high speed and loses at low speed. The low-speed octane requirements are reduced which permits a small increase in compression ratio, but not enough to gain back all the low end loss. Of course, the compression increase gives an additional gain at high speed.

In working out the timing for the V-8 advantage was taken of the fact that with the Dynaflow transmission low-speed torque is not as important as with other transmission types. Some changes in the straight-8 timing events were also necessary because of the different requirements caused by engine design differences such as valve position and manifolding arrangement.

In reply to Mr. Scherger – The crankshaft, pulley, and flywheel are not balanced as an assembly. The crankshaft and the pulley are balanced separately, as in the past production engines. The engine is balanced at final assembly as a unit, the flywheel being drilled to correct any out-of-balance condition. These operations are the same as used for past production engines and should cause no new service problems.

In reply to Mr. Miller – The extensions of the hot spots are intended to reduce icing troubles. Other V-engine manifolds have a hot spot under the vertical riser section, but none at the horizontal branches. We chose to provide hot spots at the T-section branches, and provided some additional heat to the riser section with the hot-spot extensions.

EUROPEAN POSTWAR CARS

Maurice Olley, Chevrólet Motor Division, GMC

This paper was presented at the SAE Annual Meeting, Detroit, Jan. 14, 1953.

BEFORE discussing postwar European cars it would be well to discuss postwar Europe.

Because American citizens can circulate freely all over the world, except beyond the iron curtain, they do not seem to know of the dollar curtain which isolates them from the rest of the world. They do not realize that even when they are in Europe and mixing with Europeans, they are surrounded by an invisible barrier. The Europeans are caged from America, and from each other, by bars of nonconvertible currency.

In the final analysis, all international business is barter. And the currency of one country is of value to another only in proportion to the ability of the one country to supply wanted goods to the other. As the countries everywhere strive to become self-sufficient, exchanges of their currencies come more and more to resemble exchanges of wampum for cowrie shells.

The dollar curtain results from the unwillingness of the United States to permit the import of the goods with which the outside world could try to

CONDITIONS in postwar Europe, and their relation to European postwar cars, are reviewed in this paper. The author discusses prevailing economic factors on both sides of the Atlantic, the importing and exporting of cars throughout the western world, and the outlook for automobile manufacturing in the various countries.

With the aid of more than 60 illustrations and cartoons he graphically presents a picture-story of British, French, German, and Italian passenger cars which are popular in Europe today. He concludes that European cars are generally less costly to buy and to operate than American cars; they are smaller, lighter, narrower, use less fuel; they are more nimble on the road, have better road-holding ability, better braking, and allow a more intimate view of the road. But, he also points out, they are generally low in performance, are much noiser than American cars, less completely appointed, and much less comfortable.

The Author

MAURICE OLLEY (M '21) is director of the Research and Development Section, Chevrolet Motor Division, GMC. Early in his automotive career, he worked for Sir Henry Royce in England and later for Rolls-Royce in America. He has also worked for Cadillac and Vauxhall, and during World War II he was with the British Ministry of Supply.

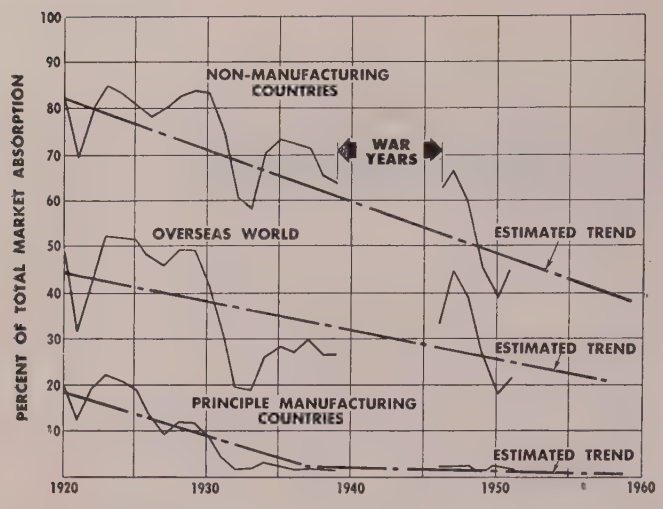


Fig 1 – North America's share of overseas market for passenger and commercial vehicles

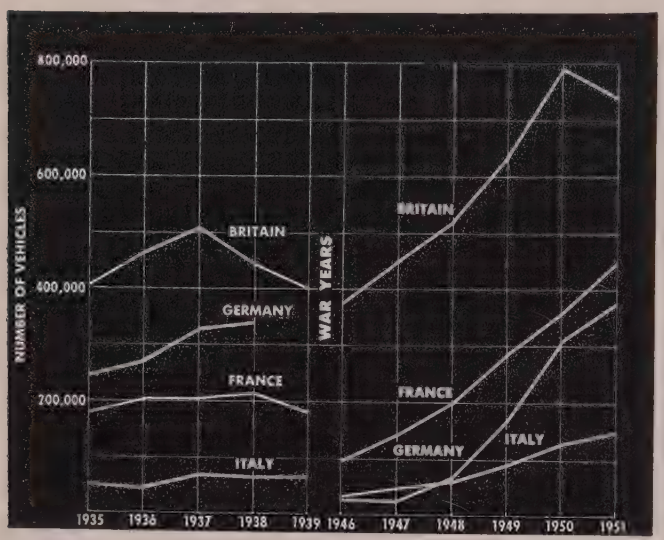


Fig. 2 – Total vehicle production of Britain, France, Germany, and Italy

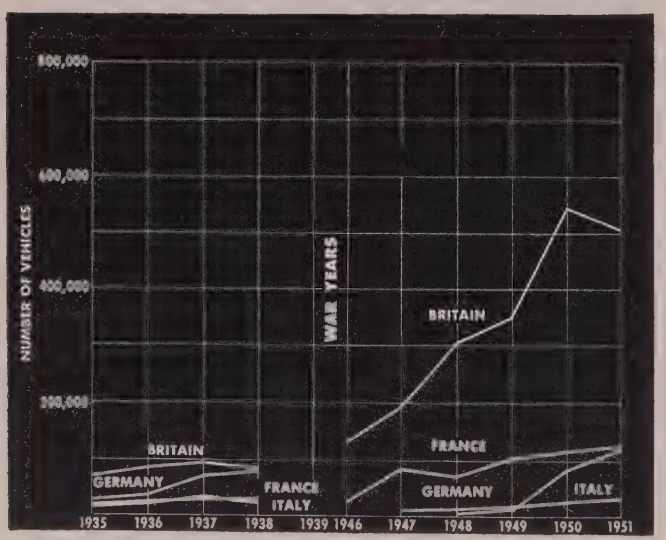


Fig. 3 – Total vehicle exports of Britain, France, Germany, and Italy

pay its debts. As a result, the outside world has almost no dollars.

Britain has a sterling barrier too, and unfortunately for Britain, it appears to be increasing in thickness. It comes from the same cause, a general shortage of sterling currency abroad, because Britain is not buying anything more than she can help. In the postwar struggle for solvency, British governments, whatever their complexion, have had to forbid the importation of goods for which, in the long run, Britain cannot pay. American automobiles are among such goods. They have been excluded from Britain since the war.

Britain is overcrowded, unable to supply from her own soil even a minimum of food for her own people. As a result of two victorious but disastrous wars in a generation, she has ceased to be a creditor nation, so that food no longer comes to her in the form of dividends on foreign investments. She has almost no raw materials of her own except coal, and barely enough of that to supply her own needs, as witness the comfort standards of British houses under coal rationing. So postwar Britain must make exportable goods by the work of her hands and exchange them abroad for food and raw materials. That is the only way in which her people can eat. If she cannot import goods, she must export people.

The inflation which has gripped the North American countries since the war has made it much harder for Britain to buy from us, and therefore much harder for her to sell manufactured goods to the world, but at the same time it has made it easier to sell these goods back to us. This accounts for the appearance among other things, of British sport cars in this country.

Although many of the West European countries are more self-sufficient than Britain, the plight of Britain colors the affairs of Europe in many ways. As a small example, many European countries have found that the profitable British tourist trade has decreased disastrously. In the early postwar years, the general desire of Britons to escape for a while from the restrictions and austerities of their own island to the greater ease of the Continent, made it necessary for the treasury to clamp down on the issue of foreign currency. And these restrictions are still in force. In particular, for British subjects, the issue of dollars for visits to the United States and Canada is rigidly restricted.

The British motor industry, although it is small compared with that of this country, and is a much smaller proportion of the total manufacturing strength of the country, is one of Britain's most important exporters and food-earners.

Any British government, whether left or right, has ridden these exports with a spur and a tight rein, using all its powers of permit and allocation to enforce that 75% of all passenger cars, and nearly two-thirds of all commercial vehicles shall be exported. Under these conditions, Britain has

become the leading vehicle exporter of the world. The results are shown in Figs. 1, 2, and 3.

Fig. 1 shows the way in which the United States and Canada are obtaining a smaller and smaller share of the overseas markets, a share which in the case of the principal manufacturing countries, is now almost zero. This chart just goes to show that a country which will not import cannot do an export trade.

Fig. 2 shows the rise of vehicle production in Britain, West Germany, France, and Italy. Fig. 3 shows the export figures for these countries, and in particular the great postwar rise in British vehicle exports.

This does not mean that there is any surplus of cars in Britain. On the contrary, there is a desperate shortage. If, as a British resident, I had ordered one of the standard production cars in 1948, I should still be waiting for it, model and price unknown, with a prospect of delivery some time this year.

Naturally, under such conditions, the yearly face-lifting which occurs here is unknown, and the public are spared the synthetic ecstasies about the latest yearly models which afflict this country.

This is true of all European cars. For example, the Citroen cars have remained basically unchanged since 1934, the Volkswagen since 1938, the Rolls-Royce and Bentley and the small Renault since 1945, and the smaller British Fords since well before the war.

Since the American passenger car has been widely considered as the standard of the world, it may be wondered why the British, for example, who cater to the world, do not simply reproduce American models of the more moderate size. The world at large would have been delighted to buy one of the smaller American cars for "soft currency."

But all foreign trade is a chancy business (as government trading agencies, with their ponderous commercial pacts, have learned to their sorrow in the postwar years). National planning cannot extend beyond national boundaries, and at any moment economic or political reasons may decide a foreign government to wreck your national plans by excluding your products entirely, or admitting them only if 50% of their construction is performed locally with local materials and labor. Britain itself, and its dominions, have been glaring examples of this policy. So any British manufacturer is wise to keep an eye on local requirements, and to manufacture only such vehicles as can be sold at home, should all other doors be closed.

Since European manufacturers have the same point of view, each country still tends to produce vehicles of a type which can be sold at home.

Sooner or later, each country has to face the question of how much of the nation's resources one individual is justified in using for his own personal transportation, and how much road space he is entitled to occupy. In many countries, such

Fig. 4 – Thousands of miles of washboard roads confront the European motorist

Fig. 5 – Old cobblestone highways are still in use in many European countries

Fig. 6 – Dirt roads with washboard surfaces, create a demand for small, economical cars

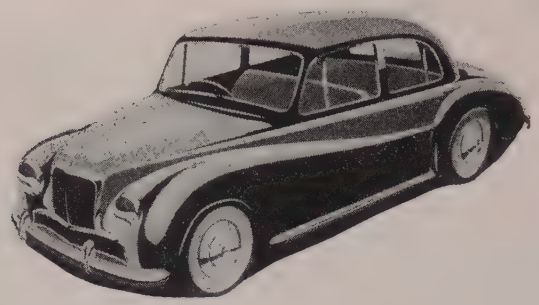


Fig. 7 – Postwar British cars show a conflict between local design ideas and a strong American influence

as Britain and Belgium for example, which are struggling to feed themselves from a restricted acreage, the actual surface area put out of commission by roads is a matter of vital importance. Modern highways, as depicted here by the designers of the American future, destroy irreplaceable farm land, and the smaller countries of Europe cannot afford them.

In contrast, therefore, to American automobile designers, the European manufacturers have to keep their imaginations under restraint as regards mere size, and strive to make a vehicle which their own employees and the general public all over the world, can afford to use. This general public is slowly graduating upwards from the bicycle and motorcycle. (The European manufacturers supply the bicycles too – some 4,000,000 a year from Britain alone.)

One of the problems facing European manufacturers is how to turn out satisfactory automobiles when nearly all the workers come to work on bicycles. I suggest that a great deal of American know-how comes from the fact that the workers are personally interested in the product, which is an essential background of their lives. Possibly this is the reason why Britain still builds the best bicycles.

Another important restriction on output is the general shortage of materials, and the lack of aggressive competition in the supply of materials and finished parts.

However, there are signs of a renaissance in this respect, particularly in the extensive and increasing use of aluminum alloys in European vehicles.

I do not think we give proper credit to European nations for the things they have contributed to the modern automobile. I know I am treading on quicksand, but I recall Italian cars of 1912 with front-wheel brakes, and independent front-suspension, early French cars with independent suspension, front-drive and steering-column gearshifts, German cars of the early thirties with swing axles and wishbone front-suspension, British cars of the thirties with fluid drive and planetary transmissions. I remember the fundamental British work on engine balance, and on the nondetonating cylinder head, and such refinements as expanding carburetors, direction indicators, and adjustable steering wheels.

Recently, also, one can notice two new essentially American features in European engineering – first, the increase of cost-consciousness and design for production; second, the increasing use of destructive testing techniques, whether by means of fatigue testing rigs, or by use of proving grounds.

In Britain, one sees crates of cars and trucks and c.k.d. cases containing automobile parts for assembly abroad, consigned to ports having outlandish names, some of which one had scarcely heard of. There are evidently parts of the world which are enjoying real prosperity. People are riding bicycles in Africa and the Near East, who never dreamed of owning such a luxury. And some of them are beginning to consider assisted bicycles or even motorcycles, and a few even see an automobile in their future. But in spite of ambitious road-building programs all over the world, the actual conditions for the use of automobiles are similar to American conditions of 40 years ago, before our highways were built. There are thousands of miles of washboard gravel between main cities, and lots of dirt roads, and a lavish lack of bridges (Figs. 4, 5, and 6).

There is also a demand everywhere for small vehicles which will fit on narrow roads, which are economical to run and which can be towed out of the bad spots by oxen.

The export market is of infinite variety, but appears to have one common characteristic, that it cannot use a heavy car, with a long wheelbase and low ground clearance, except as a parade vehicle for official receptions.

The European industry is realistic. It cannot rebuild the cities all over the world to avoid the sort of congestion which afflicts downtown Detroit today. So it builds small vehicles which have some

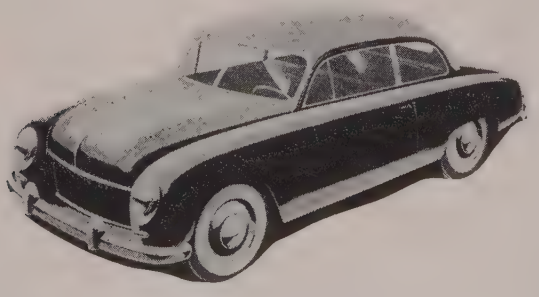


Fig. 8 – German postwar cars show essentially national design features

chance of weaving their way through the cities as they exist.

Generally speaking therefore, European cars, while each possessing its own national characteristics, differ from the American in certain fundamentals. They are generally less costly to buy and to operate. They use less of the precious motor fuel, which is everywhere heavily taxed, frequently rationed, and often in short supply. They are smaller, lighter, and narrower. They are more nimble on the road, have better road-holding ability, better braking, and allow a more intimate view of the road all around the vehicle. On the other hand, they are generally low in performance in the hands of the average driver; they are much noisier, less completely appointed, and less comfortable.

In appearance, the European postwar cars are nearly all more or less related to American styling. The Italians have refined it, the French and British have each resisted it along different lines, but are generally surrendering to it. The Germans have been slow to conform to it and have shown signs of developing a distinctive scarab-like design of their own, associated with their characteristic mechanical originality and lack of traditionalism.

Figs. 7, 8, 9, and 10 show typical postwar cars – British, German, French, and Italian. These illustrations have been kindly contributed by Count Mario Revelli of the GM Styling Section as representing the apparent trends in European production cars. He has also contributed some interesting comments.

In general, all four styles show the American influence, which the British and Germans have accepted perhaps unwillingly, the French with reservations, the Italians enthusiastically. It was remarkable how suddenly after the war the Italians seized on the American styling and refined it, so that now it is difficult to say whether the other European nations are following the American or the Italian styling.

In all four styles the wheels are set almost flush with the sides of the vehicle. Overhangs, both front and rear, are less than in American cars, so that the wheelbase is a greater proportion of the total length of the vehicle.

Interior width is based on two passengers, not three. Interior height is generally acceptable, though in some cars the doors are too low for comfort.

Baggage space is more or less restricted. Bumpers are less massive and ornate.

British Car – There is evidently a conflict in British design ideas, between the all-British designs, such as the Morris, Austin, Armstrong-Siddely, Triumph, or Jaguar, and the cars showing strong American influence, such as Ford or Vauxhall.

There is still a certain hankering for the sharp-cornered design, which goes back to the hansom cab and genteel Victorian horse-carriages – a design, incidentally, which gives a very thin roof, and good entrance height.

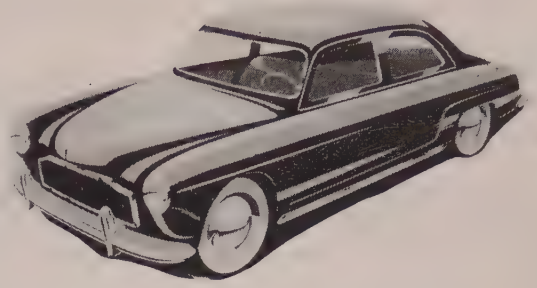


Fig. 9 – French small car is considered best in Europe

In most British designs the front overhang is quite short, the shortest in Europe. Fender lines and the suggestion of a radiator are retained, sometimes, especially in the shorter cars, with peculiarly depressing results.

Headlights are still mounted in some designs on the catwalk, with small sidelamps on the fenders as required by British law to indicate the full width of the vehicle.

The British, like the French and Italians, prefer 4-door cars. The trunks are frequently swept away smoothly, in such a way that baggage space is restricted. The British also use various devices to house the spare wheel separately from the baggage, often with the effect of reducing the useful height of the baggage space.

German Car – The Germans evidently intend to keep some essentially national features, and to develop the 2-door 4-seater car along inexpensive lines. The German autobahn system, allowing sustained speeds, makes the 2-cycle engine possible and popular, and this is bound to affect ideas of acceptable car styling.

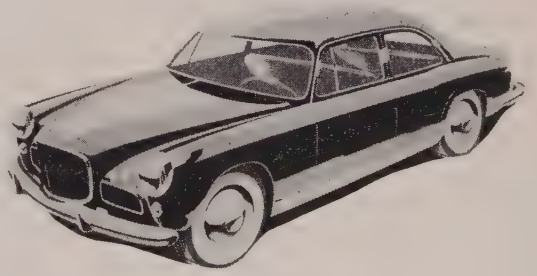


Fig. 10 – Italian cars are like American Models, with lines slimmed down and refined

In general, the cars are somewhat boxy, parallel in plan view, with reasonable trunk capacity. Roofs are somewhat thick and vertical entrance room poor. The driver is frequently too far forward for comfortable leg room.

Fender lines are disappearing completely, rear windows are small, and road vision not good. The headlights are frequently flush-mounted in the fenders.

French Car – The French small car is considered the best in Europe. Some entirely new models will appear shortly, in the Citroens, Peugeots, and others.

Roof thickness is intermediate between the British and German.

Fenders are disappearing completely, but in spite of this a boxy appearance is being avoided.

The French preference for four doors is shown in the small Renault, the smallest 4-door sedan in the world. This shows good sense. The small bodies do not permit crawling across without considerable loss of dignity. So each of the four passengers sits back in his appropriate hole and slams the door on himself. This is the way one handles these small cars. You do not so much get into them, as put them on, like an article of clothing.

French front grilles have shown various compromises, but lately there appears a clear horizontal tendency, similar to the American.

Italian Car – The Italian lines are like the American, slimmed down and refined, as though by judicious use of a low-calory diet. Here are the shortest overhangs in international production. Roofs are thinnest and entrance room is good. Fender lines have almost disappeared. Rear windows are increasing in width, and road vision is uniformly good. Grille designs show still definite striving for individual characteristics.

One must not judge these European cars by American standards. These cars are not just funny little things used by people who don't know any better. The American car, if scaled down to the dimensions necessary for European use, would not contain normal human beings. It would contain the same number of parts and would therefore cost almost as much to build. There is no room in European designs for mere advertising space, for large areas of tin enclosing nothing but wind, to make a car look big. Since the roads and streets are narrow, the car must be narrow, so that, to get the necessary seat width, it is desirable to have the seating compartment entirely between the front and rear wheels.

In general, the Europeans do not like the appearance of very short hoods, so their hoods are frequently full of legs and feet, with the small engine at the far end, and the windshield well back in the driver's lap. Wheelbases are frequently longer than the American, relative to the overall length of the car, especially in those designs which make the strongest bid for maximum road-holding ability.

From what has been said, one can see that the European design problem is to combine a large interior with a small exterior. Only those parts of the vehicle which are useless for containing passengers can be used for the necessary services, for powerplant, fuel, battery, or spare wheel. There is usually a shortage of baggage space, especially on rear-engine cars, and it is still usual to carry baggage in racks on the roof (in itself a strong argument for stiff springing). An advantage of placing the passenger space squarely between front and rear wheels is that it becomes possible to make four doors from 2-door dies.

This movement forward of the passenger space is, of course, nothing new. It has been going on here for years, and has caused the same problem – the overloading of the front wheels. European powerplants are smaller and lighter than ours, and a good deal of light alloy is used in many of them. But the vehicles themselves are lighter, so the problem is the same – the awkward tendency of 60% of the car weight to fall on the front tires.

One European solution is to drive the front wheels. If the center of gravity is low, the wheelbase relatively long, and the acceleration of the vehicle moderate, so that the set-back of weight to the rear in acceleration is not excessive, this appears to be a good solution. Some of these front-drive vehicles are reputed to be the best cars in Europe, even in the Alps. The body portion becomes a simple box without any floor tunnel and with a low floor line. Some of these cars use rear wheels mounted independently for parallel wheel motion, but the better practice calls for the use of a plain axle at the rear, or even for a swing axle.

One undoubted advantage of front-drive cars is their uncanny directional stability, and the precision of their handling. They are good in cross-country work too, because the front wheels can lift themselves over an obstacle.

A feature of the German 2-cycle-engine cars is that, with one exception, they are all front-drive vehicles.

The other European solution is to put the engine at the rear and drive the rear wheels. This puts over 60% of the car weight on the rear tires, and this weight balance remains practically the same at all passenger loads.

These cars are in general exceptionally good for traction on wet granite paving or in snow. They also roll very little on turns, partly because of the rear swing-axle and also because the mass of the powerplant is located on the roll axis.

However, it has always appeared to me that they are a poor bargain, at least in the form in which they are at present built. Having ruined the trunk space at the rear by putting an engine in it, they are all short of baggage space. The space between the wheels in front has to take the spare wheel, and fuel tank, and is the wrong shape for carrying baggage. The forward fuel tank is a collision risk, as is the mass of the engine in the rear. These cars

are frequently noisy, perhaps due to drumming of the rear-engine bulkhead, also because the powerplant, since it carries the swing-axle, cannot be mounted softly. Finally, it appears impossible to make a car with heavily loaded rear tires handle safely in a wind even at moderate speeds, without fitting twin tires at the rear, which is impossible with a swing-axle. The European cars of this type, even though they use up to 10 lb more inflation in the rear tires and up to 10-deg caster angle, and even fit centering springs on the steering linkage, still wander badly in a high wind.

Let us take a look at the general size of European production cars. To avoid presenting you with lists of figures, I have made averages of the figures for six small European cars, eight small-medium cars, and seven medium cars. They are all cars of standard production "family" types, omitting all luxury cars or sports cars. (Table 1.)

I have listed the wheelbase and track, the overall length and width, number of cylinders, engine size, number of forward speeds, axle ratio, curb weight, and price. I might also make a rough guess at the average fuel consumption, which would be, for the small cars about 40 mpg, for the small-medium about 30 mpg, and for the medium about 25 mpg. Typical maximum level speeds would be for the small cars 45-55 mph, for the small-medium 55-65, and for the medium about 65-80 mph.

It is difficult to present a good cross-section of the European industry because of its variety, and the large number of cars of negligible production volume. Also, it would be misleading to give space to certain luxury cars, of small production volume, supplied nowadays chiefly to government departments and foreign bureaucrats. It is difficult to find one's way around among the sports cars, some of which are just a passing thought, while others enter into serious production.

I have taken a German survey of standard production cars of British, German, French, and Italian origin, and have listed them for body-frame construction, engine position and drive, front suspension, and rear suspension.

Body-Frame Construction – (Table 2.) The thing to notice here is that, of 69 cars listed, 38 are of unconventional construction; 27 are of full frameless construction; 7 are built on a platform underbody; and 4, all German and all with 2-cycle engines, are built on a tubular backbone. The British and French are both going strongly for frameless construction and are overcoming the difficulties of assembling these in assembly plants abroad. More and more convertibles are appearing in frameless construction.

Engine Position and Drive – (Table 3.) Front-drive and rear-engine cars are still in a minority, and only time will show whether they will increase. The British cars are all conventional in this respect.

Front Suspension – (Table 4.) The coil-spring wishbone suspension is almost as firmly established abroad as it is here, although the torsion-rod wishbone has some important adherents, particularly in Britain.

Rear Suspension – Table 5 shows the increasing devotion of the German engineers to the swing-axle. It is remarkable that even some of the German front-drive cars fit swing-axles at the rear

Table 1 – Average Specifications for Typical European Passenger Cars

	Six Small Cars	Eight Small-Medium Cars	Seven Medium Cars
Wheelbase, in.	84.1	96.3	103.3
Tread, in.	47.4	50.7	53.15
Overall Length, in.	142.4	162.0	171.8
Overall Width, in.	56.0	63.4	66.35
Number of Cylinders	2 or 4	4	4 or 6
Displacement, cu in.	41.0	82.0	132.5
Forward Speeds	4	3 or 4	3 or 4
Axle Ratio	4.670	4.845	4.206
Curb Weight, lb	1418	2264	2604
List Price, Sedan	$900-$1200	$1255	$1543

Table 2 – Body-Frame Construction of Typical European Passenger Cars

	32 British Cars	17 German Cars	13 French Cars	7 Italian Cars
Frame	19	5	5	2
Full Frameless Construction	12	4	7	4
Platform Construction	1	4	1	1
Tubular Backbone	—	4	—	—

Table 3 – Engine Position and Drive for Typical European Passenger Cars

	32 British Cars	17 German Cars	14 French Cars	7 Italian Cars
Conventional Front Engine, Rear Drive	32	9	7	7
Front Engine, Front Drive	—	5	6	—
Rear Engine, Rear Drive	—	3	1	—

Table 4 – Front Suspensions of Typical European Passenger Cars

	32 British Cars	17 German Cars	13 French Cars	7 Italian Cars
Coil Spring Wishbone	19	7	7	3
Transverse Leaf Spring	3	6	3	2
Torsion Rod Wishbone	8	—	2	—
Torsion Rod Parallel Link	—	2	—	—
Telescope Spring	2	—	—	2
Rubber	—	2	—	—
Radius Arm Coil Spring	—	—	1	—

Table 5 – Rear Suspensions of Typical European Passenger Cars

	32 British Cars	17 German Cars	14 French Cars	7 Italian Cars
Rigid Axle and Leaf Spring	28	4	5	4
Rigid Axle and Torsion Rod	2	1	4	—
Rigid Axle and Coil Spring	—	—	2	2
Torque Tube	2	1	—	—
Swing-Axle	—	10	1	—
Semiswing Independent	—	—	1	1
Parallel Radius Arm	—	1	1	—

Table 6 – Types and Average Displacement of Typical European Passenger-Car Engines

4-Stroke Engines	Number of British Cars (Total 32)	Displacement of British Engines, cu in.	Number of German Cars (Total 19)	Displacement of German Engines, cu in.	Number of French Cars (Total 12)	Displacement of French Engines, cu in.	Number of Italian Cars (Total 7)	Displacement of Italian Engines, cu in.
4-Cyl Vertical Water-Cooled	20	99.6	5	94	6	95.2	5	78.7
6-Cyl Vertical Water-Cooled	11	153.9	4	147	2	158	—	—
4-Cyl Horizontal Water-Cooled	1	90.7	—	—	—	—	—	—
4-Cyl Horizontal Aircooled	—	—	3	71.3	—	—	—	—
2-Cyl Horizontal Aircooled	—	—	—	—	3	35.1	—	—
4-Cyl Diesel Water-Cooled	—	—	1	108	—	—	—	—
6-Cyl-V Water-Cooled	—	—	—	—	—	—	1	107
4-Cyl-V Water-Cooled	—	—	—	—	—	—	1	55
8-Cyl-V Water-Cooled	—	—	—	—	1	131.7	—	—
2-Stroke Engines								
2-Cyl Vertical Water-Cooled	—	—	4	36.1	—	—	—	—
2-Cyl Vertical Aircooled	—	—	1	17.9	—	—	—	—
3-Cyl Vertical Water-Cooled	—	—	1	42.5	—	—	—	—

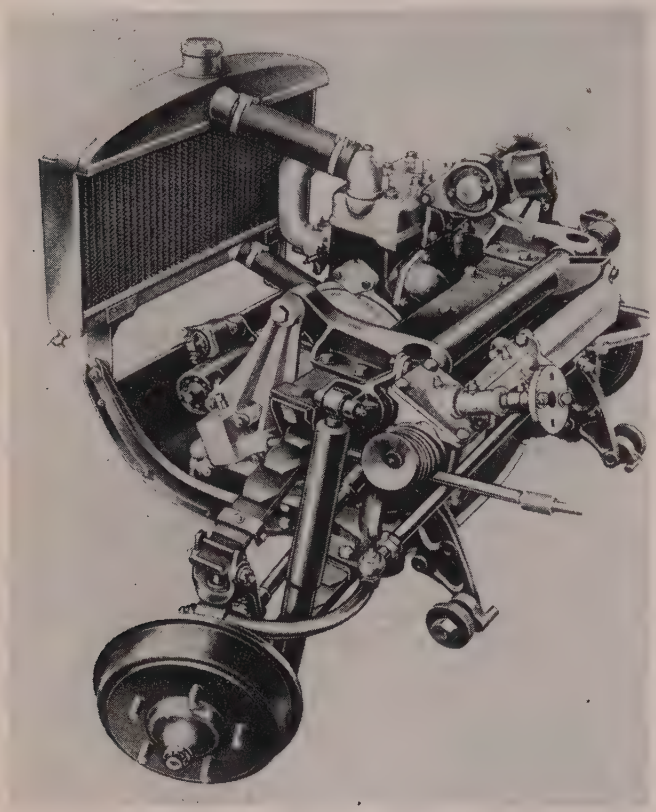


Fig. 11 – Goliath, a 2-cycle German engine

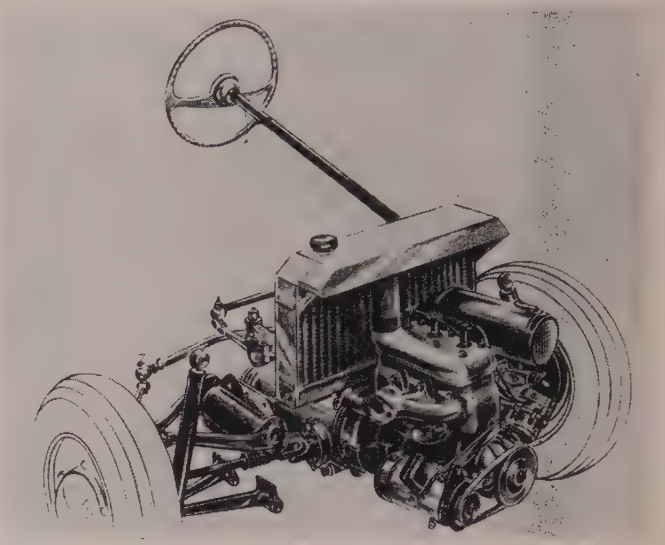


Fig. 12 – Hanamag, a 3-cyl German engine

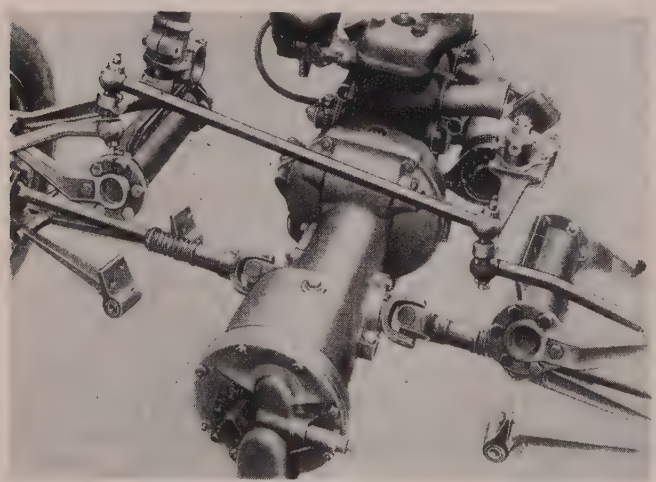


Fig. 13 – Transmission and front suspension of Hanamag, shown in Fig. 12

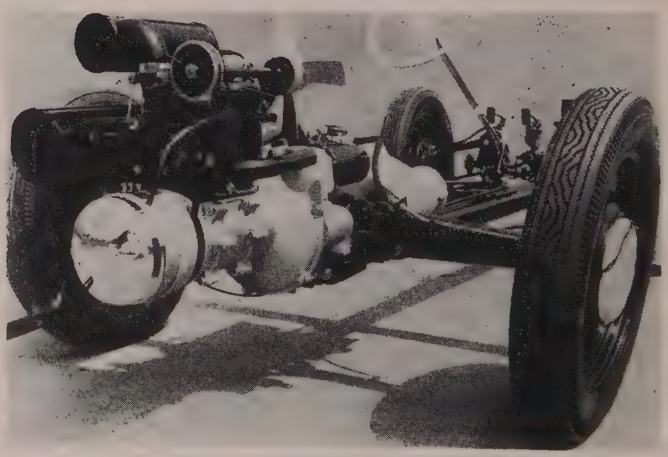


Fig. 14 – Champion, a German engine, mounted at rear

although these are "dead" swing-axles, and it would seem easy to avoid using them. However, they fit in conveniently with a tubular backbone.

Engines – Table 6 shows the preponderance of the vertical water-cooled 4-cyl engine in Britain, and its popularity elsewhere. I have averaged the size of these engines, and it is remarkable that they come out at 99.6 cu in. in Britain, 94 in Germany, and 95.2 in France, while the 6-cyl engines average 153.9, 147, and 158 cu in. respectively in the same three countries. The small 2- and 3-cyl 2-cycle engines in Germany are also shown. One can say that the European car has 25 cu in. per cylinder.

Since the sectional area of cars and the resulting air resistance do not decrease in proportion to their weight, the load factor on European engines is high. Touring speeds on modern European roads are nearly as high as in the United States. Therefore there is need for exceptional workmanship, or alternatively, for more frequent overhaul. In this connection the increase of 2-cycle engines in Germany is interesting. Figs. 11, 12, 13, and 14 show three of these, the Goliath (36 cu in.), the Hanamag 3-cyl (40 cu in.), the same engine with its transmission and front suspension, and the 24-cu in. Champion engine mounted at the rear.

These engines are not too good for fuel consumption or idling, unless fitted with the Bosch injector (shown in Fig. 15) but they appear to be immune from cylinder wear and contain a minimum of parts requiring service. The only service work needed is to remove the cylinders every 10,000 miles or so, and scrape carbon out of the ports – a relatively unskilled job often done by the owner. By American standards these small engines are not acceptable, but this does not mean that they will always remain so.

Another interesting engine is the small Mercedes 108-cu in. 4-cyl 4-cycle diesel, of which Mercedes are making about 900 a month for their standard small sedan car. (This is about 25% of their entire output.)

Fig. 16 shows a section of the cylinder head. Power output is 40 hp at 3200 rpm. The idling noise of this engine is outrageous by American standards, but the performance on the road is smooth and pleasant, with that instant response to the accelerator which is typical of a good diesel. Consumption of diesel fuel is 38 mpg on a 4-5-seat sedan car.

Figs. 17 to 59 show how some of these European cars are put together.

Starting with the smallest, Fig. 17 shows the 2-CV Citroen, a front-drive car of 94½-in. wheelbase, with a curb weight of 1200 lb, a 2-cyl air-cooled engine of 23 cu in., a top speed of only about 40 mph.

It has parallel independent suspension front and rear. The canisters on the wheel arms are interesting since they contain sealed-in harmonic balancers

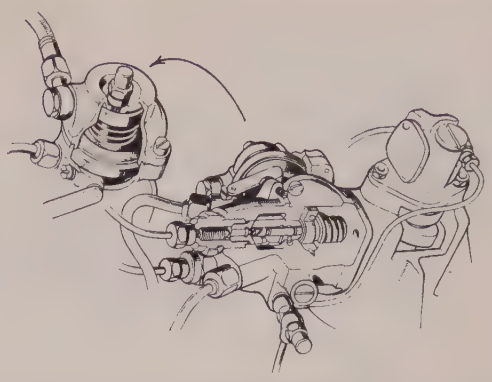


Fig. 15 – German engines are frequently fitted with Bosch injector

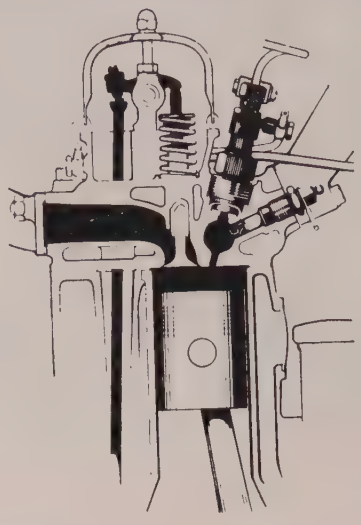


Fig. 16 – Section of cylinder head on German engine

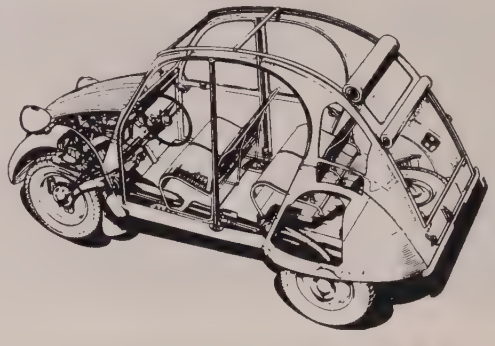


Fig. 17 – View of French Citroen, showing how it is put together

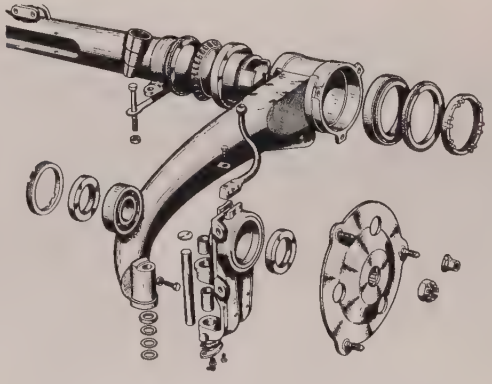


Fig. 18 – One of front wheel arms on Citroen

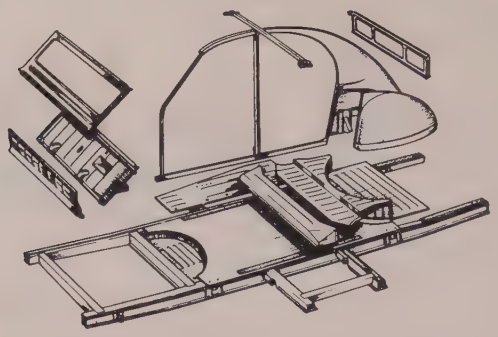


Fig. 19 – Platform underbody and body shell of Citroen

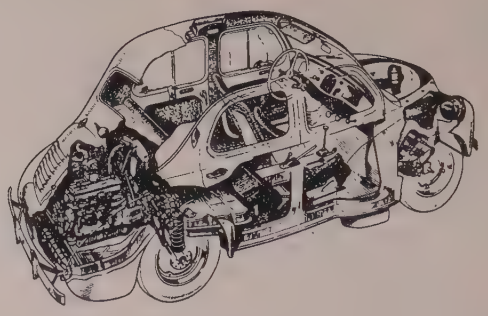


Fig. 23 – French Renault, popular in both Canada and the United States

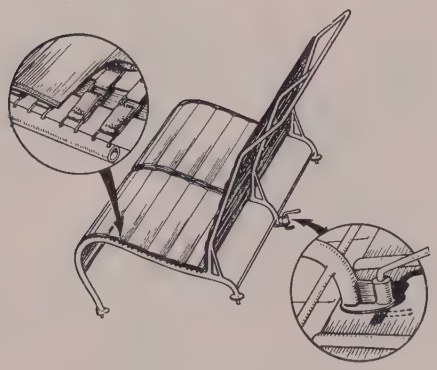


Fig. 20 – Remarkably light seat-frame construction is feature of Citroen

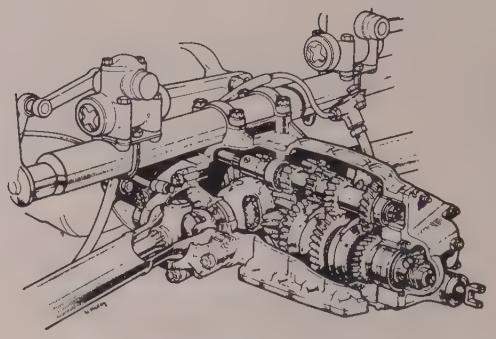


Fig. 24 – Detail of 3-speed synchromesh transmission in the Renault

Fig. 21 – Dyna Panhard, a popular small car in France

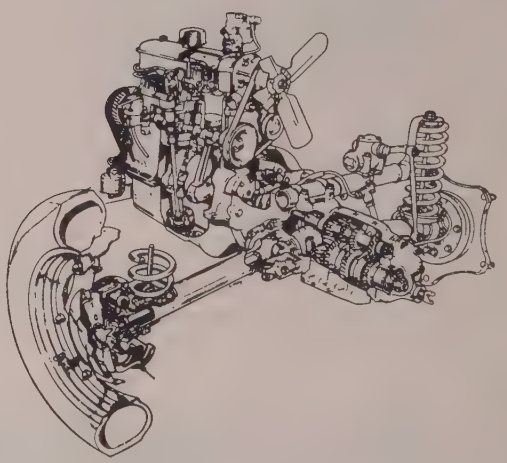


Fig. 25 – Detail of rear swing-axle of Renault

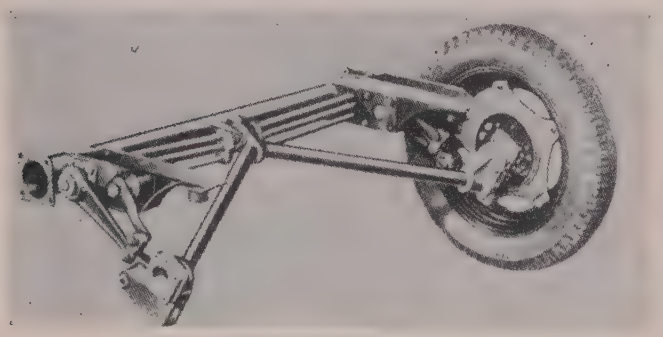


Fig. 22 – V-shaped rigid axle of the Dyna Panhard

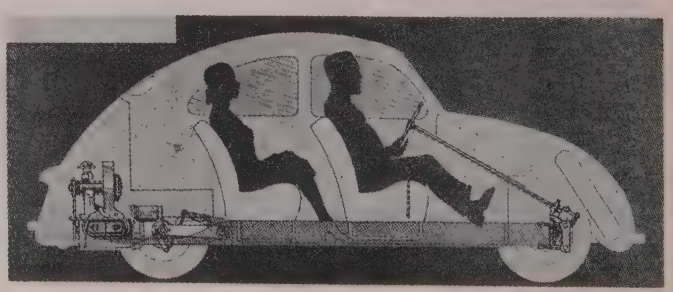


Fig. 26 – German Volkswagen with aircooled engine at rear

to help keep the wheels on the ground.

Fig. 18 shows one of the front wheel arms of the Citroen. Fig. 19 illustrates the platform underbody and tent-like body shell. Fig. 20 shows the remarkably light seat-frame construction. (This is motoring reduced almost to its lowest terms.)

Fig. 21 shows the Dyna Panhard, a front-drive car of only 83-in. wheelbase, curb weight 1370 lb, with a 2-cyl aircooled engine of 37 or 45 cu in. in the two models. It has a more or less conventional chassis, unusual in these small cars. Front suspension is by transverse springs. Rear suspension is by torsion rods operating on a peculiar V-shaped rigid axle, shown in Fig. 22. This little car is well-considered in France, and has extraordinary road ability.

Figs. 23, 24, and 25 show the 4-CV Renault of which there are now considerable numbers in Canada and the United States. Also shown is a detail of the 3-speed synchromesh transmission, the complete powerplant, and rear swing-axle. The engine is a 4-cyl overhead-valve wet sleeve of only 46 cu in., and the curb weight is only 1340 lb, with a 4-seat, 4-door, all-steel sedan body which is roomy for a car of this size.

Fig. 26 shows the Volkswagen. It has a 94½-in. wheelbase with a 4-cyl horizontal opposed air-cooled engine at the rear. The Volkswagen engine is 69 cu in. The similar souped-up engines for the Porsche car are of 66 and 78.5 cu in. In a car of this size and weight, this begins to give some real performance, so that whereas the Volkswagen only claims 62 mph, both versions of the Porsche show close to 90 mph.

Suspension is by the well-known Porsche torsion rod and parallel link suspension at the front, using laminated torsion rods of an overall square section, and by the Porsche type of swing-axle with torsion rods at the rear. Curb weight of the Volkswagen is 1630 lb, and of the Porsche about 1700 lb.

Fig. 27 illustrates the Porsche platform-type underbody. The Volkswagen underbody is similar in principle. Fig. 28 shows an open model of the engine crankcase, clutch housing, drive gears, and transmission. A synchromesh transmission is now being fitted on this vehicle.

The Volkswagen is a regular mine of interesting features; for example, the exceedingly light and simple body construction, with the hollow sills used as heater ducts to lead warm air forward from the engine.

Another feature is the extremely simple form of the universal joints at the inner ends of the swing-axle shafts which are formed with flattened ends in the DeDion type of joint.

Fig. 29 – The chassis of the German Gutbrod car with a 2-cyl, 2-cycle engine of 36 cu in., a wheelbase of 79 in., a curb weight of only 1520 lb. This is a 2-seater car. The engine cooling is thermo-syphon.

Fig. 30 – A detail of the dead coil-spring swing-

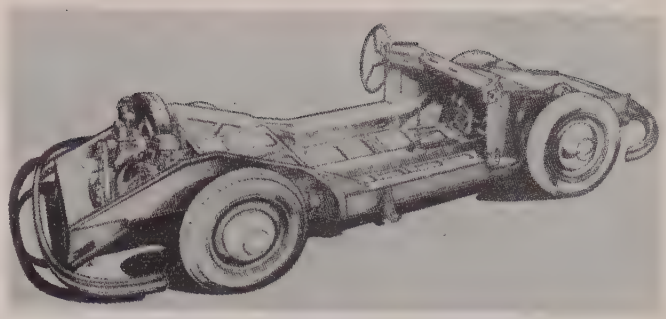


Fig. 27 – Porsche platform-type underbody

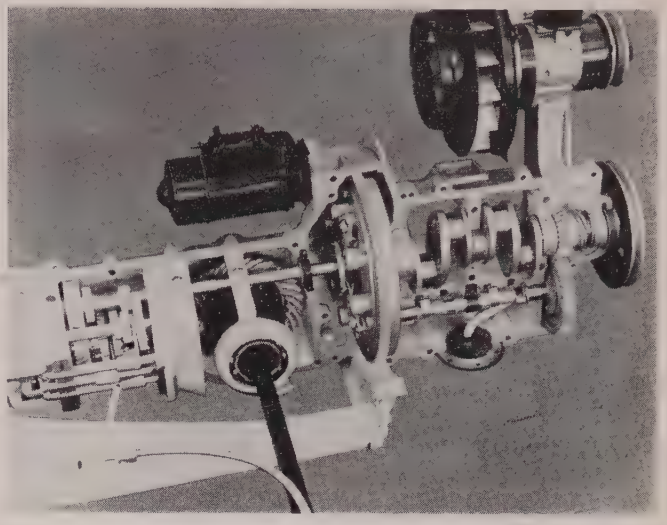


Fig. 28 – Open model of Volkswagen crankcase, clutch housing drive gears, and transmission

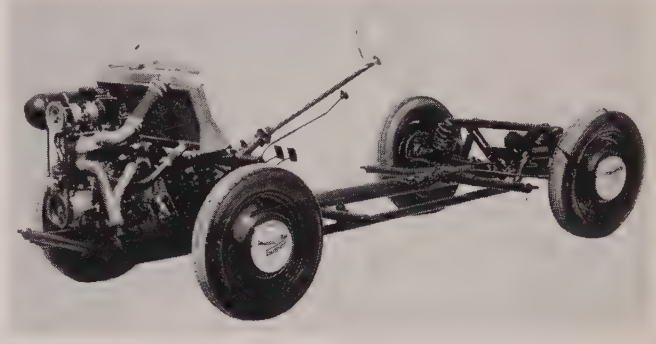


Fig. 29 – Chassis of German Gutbrod car

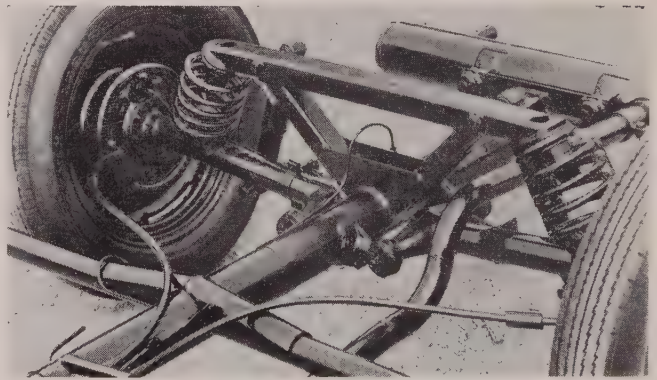


Fig. 30 – Detail of dead coil-spring swing-axle used on Gutbrod, and of tubular backbone chassis

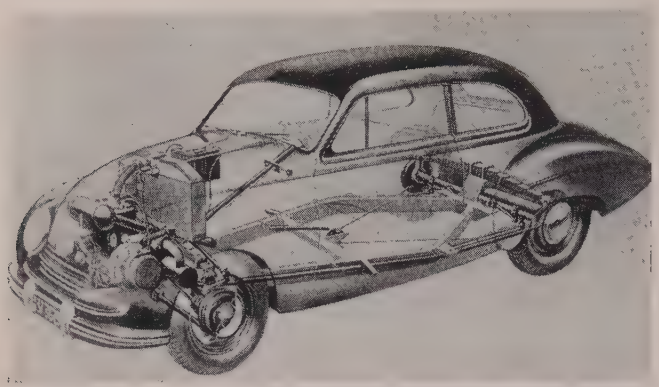


Fig. 31 – Layout of modern DKW, showing 2-cyl 2-stroke engine mounted transversely ahead of shafts driving front wheels. Note axle-beam connecting two rear wheels

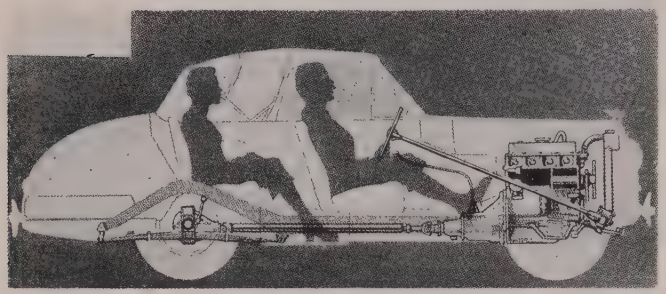


Fig. 32 – Austin A40, a typical small-medium car

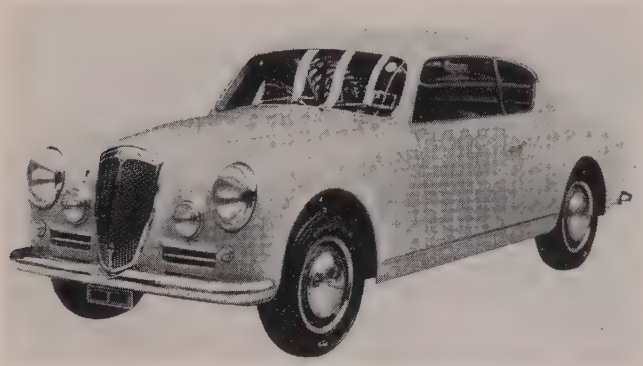


Fig. 33 – Latest variation on Lancia Aurelia

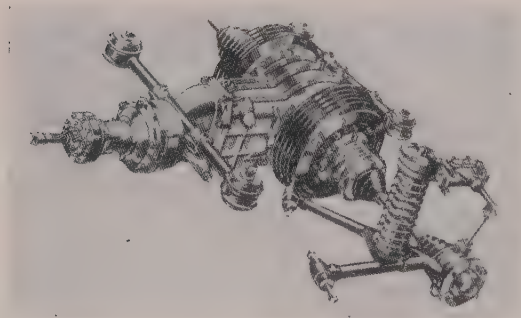


Fig. 34 – Independent rear suspension with 4-speed gearbox and clutch are all at rear of Lancia Aurelia

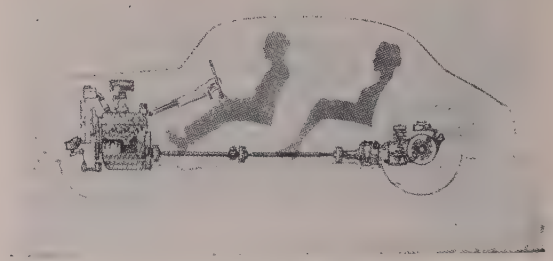


Fig. 35 – Rubber joints on driveline and square dash corner are permitted by removing clutch to rear of Lancia Aurelia

Fig. 36 – The Hotchkiss Gregoire has great front overhang

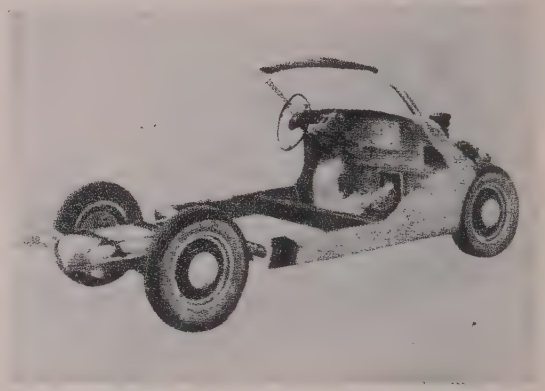


Fig. 37 – Cast body-frame of Hotchkiss Gregoire

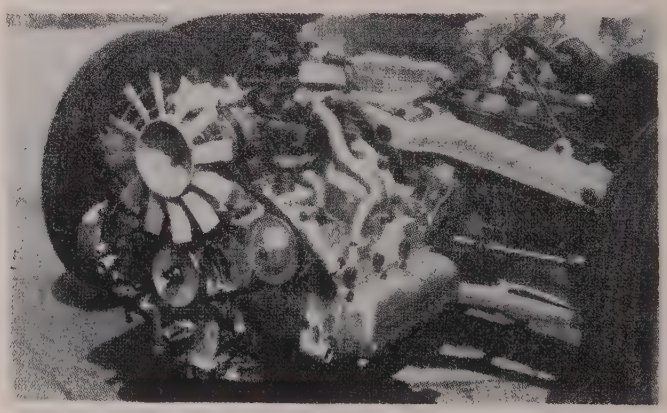


Fig. 38 – Overhung engine of Hotchkiss Gregoire

axle used on this car, and of the tubular backbone chassis.

Fig. 31 – The DKW front-drive, 2-cycle car made by the Auto Union concern, with a 2-cyl engine of 42 cu in. set transversely. As may be seen this has a separate chassis frame, transverse leaf springs, and a rigid axle in the rear. Weight of the 4-seat car is 1900 lb and wheelbase 92½ in. This is not a small car by European standards, but rates as a family sedan.

Fig. 32 – The Austin A40, a typical British small-medium car. It has a 92½-in. wheelbase and 2230-lb weight. A conventional 4-cyl engine of 73 cu in., with coil-spring wishbone front suspension and hotchkiss rear axle. This car has a separate frame. The later and smaller Austin 7 is Austin's first venture into frameless construction.

Fig. 33 – This is the latest variation on the Lancia Aurelia. It has a 113-in. wheelbase, with a curb weight of only 2500 lb, and front suspension by telescope coil springs. The rear suspension is shown in Fig. 34.

Fig. 34 shows the semiswing coil-spring independent rear suspension with 4-speed gearbox and clutch, all at the rear.

Fig. 35 shows in outline the rubber joints on the driveline and the square dash corner permitted by removing the clutch to the rear. The combination of long wheelbase and independent rear end gives the car remarkable road-holding and handling qualities. All speeds are indirect, this being done to keep the driveline down and give a low floor. The engine and transmission are both in aluminum.

Fig. 36 – The Hotchkiss Gregoire, remarkable for its great front overhang, and its cast aluminum wheel rims, bolted direct to cast aluminum brake drums. It has a 98½-in. wheelbase, front drive with a 4-cyl horizontal water-cooled engine of 134 cu in., and a curb weight of 2500 lb. The body frame unit is largely constructed of light alloy castings, as are the engine and suspension parts. Fig. 37 gives an idea of the cast body-frame construction. Fig. 38 gives a picture of the overhung engine, which seems to be in a fairly vulnerable position.

Fig. 39 – The Alfa Romeo has a 4-cyl engine of 115 cu in. giving 80 hp at 4800. Curb weight is 2430 lb and wheelbase 103.5 in., frameless construction, coil-spring suspension front and rear. The rear-axle construction is shown in Fig. 40. This has rubber-mounted radius arms, and a rubberized driveshaft, of a type frequently seen on European cars and useful in suppressing road noise.

Fig. 41 shows what the Italian coach builders can do in the way of appearance on cars of this type.

Fig. 42 pictures the Fiat 1400. It has a 4-cyl engine of 85 cu in. with bore and stroke of 3.23 and 2.60, an extremely short-stroke engine developing 44 hp at 4400 rpm. It has a coil-spring suspension front and rear with wishbone independent front axle and a rigid rear axle. Dimensions of

Fig. 39 – 4-cyl Alfa Romeo

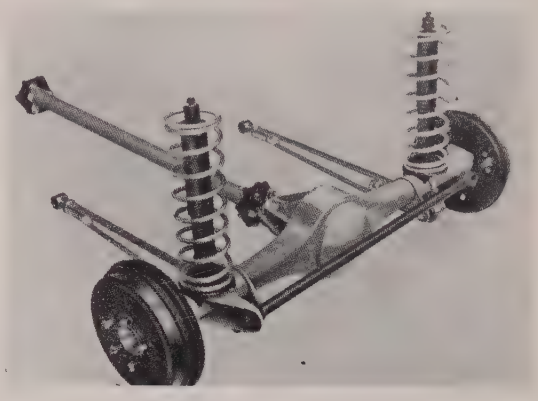


Fig. 40 – Rear-axle construction of Alfa Romeo, showing rubberized driveshaft, useful in suppressing road noise

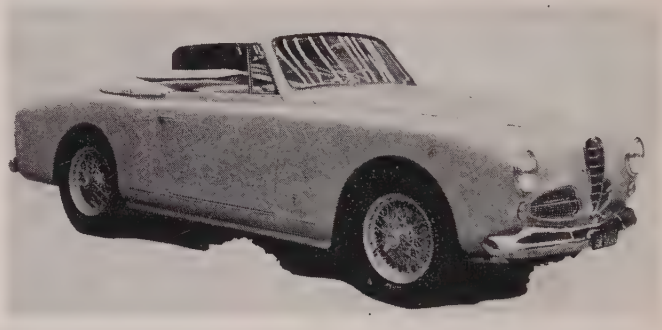


Fig. 41 –Italian coach builders contribute modern styling to postwar cars

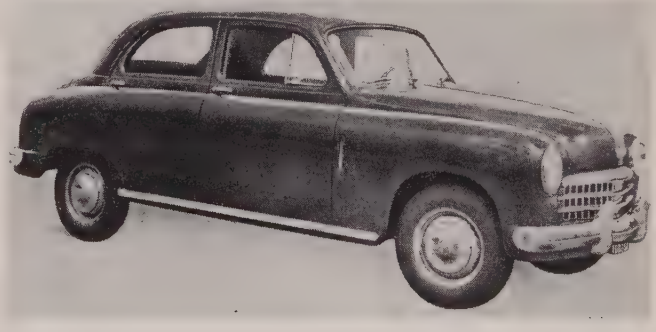


Fig. 42 – Italian Fiat has a 4-cyl engine

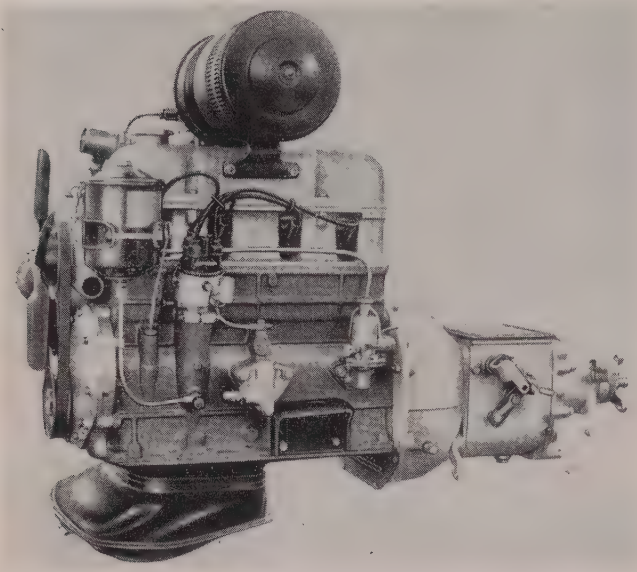


Fig. 43 – Fiat engine, mounted on springs, is remarkably smooth

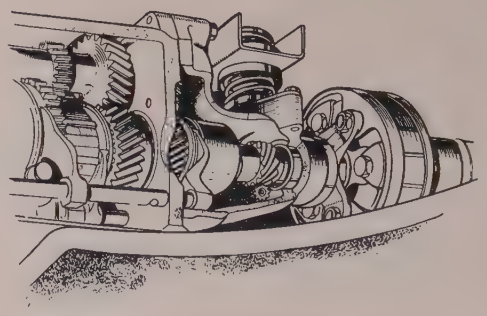


Fig. 46 – Rear of 4-speed transmission of Italian Fiat

Fig. 47 – Fiat uses rubber coupling and other devices to reduce engine roughness and road noise

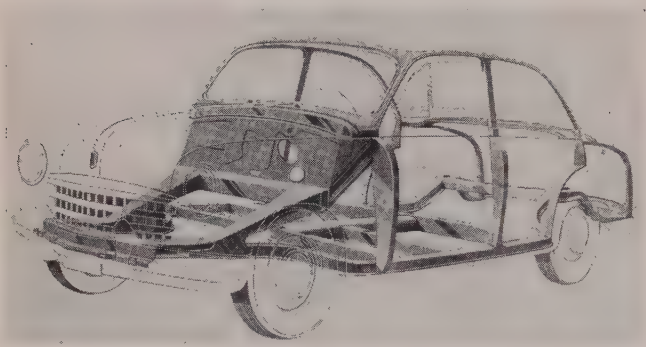


Fig. 44 – Frameless construction is typical of postwar British and European models

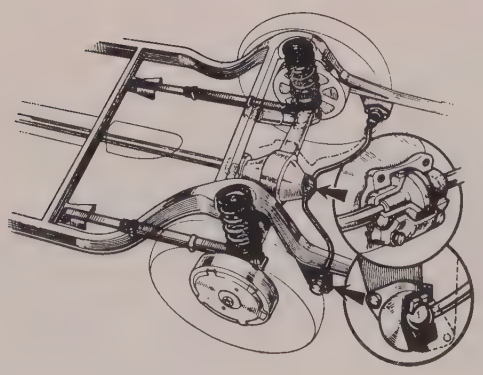


Fig. 45 – Rear-axle construction of Italian Fiat

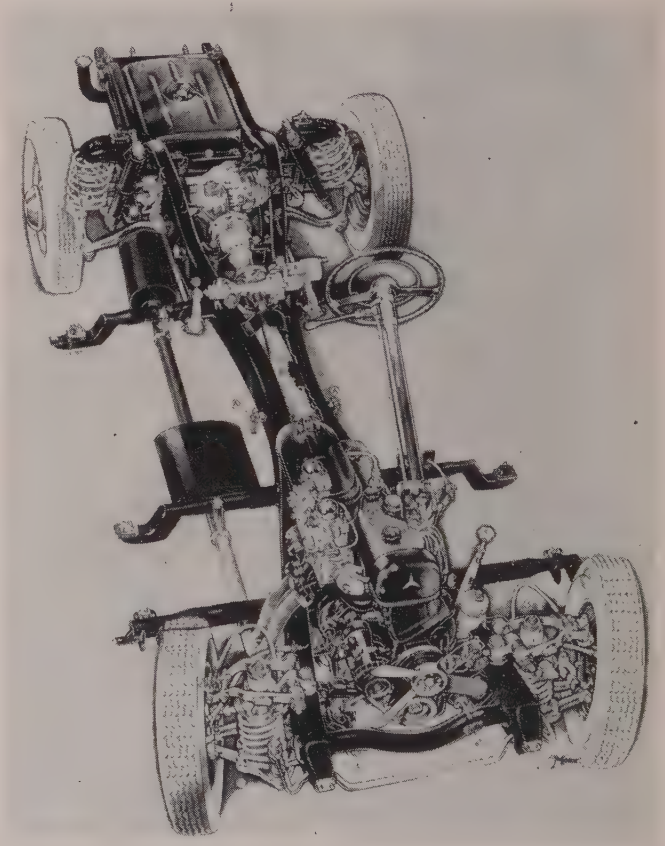


Fig. 48 – Chassis of Mercedes Benz, showing typical construction of postwar Mercedes models

the car are 104.3-in. wheelbase and a curb weight of 2500 lb.

Fig. 43 shows the engine, which is mounted on springs, and is remarkably smooth, while the car itself is also remarkably free from shake. Fig. 44 gives an idea of the frameless construction and is typical of postwar construction on new British and European models.

Fig. 45 shows the rear-axle construction, by coil springs, with cantilever leaf springs acting as radius arms only, and a stabilizer bar in the form of a bent rod, which not only centralizes the axle but also acts as a toggle and reduces the spring rate at the normal standing height of the vehicle, by some 30 lb per in.

Fig. 46 shows the rear of the 4-speed transmission on the same car, with the rear mount of the powerplant on a coil spring, and the overhung gears of the fourth speed overgear.

Fig. 47 shows the lengths to which Fiat has gone to reduce engine roughness and road noise, using rubber coupling and the front half of the propeller shaft in the form of a flexible torsion rod mounted in a light tubular casing to prevent whirling.

Finally, to wind up our survey, Fig. 48 illustrates the chassis of the Mercedes Benz 300, showing the typical construction of the postwar Mercedes models, the frame made of oval tubing, and waisted at the center, with a box at the center to complete the X construction.

This rates in Europe as a very large car. It is, however, a genuine production model of which the output last summer was running at the rate of 350 a month. It has a 120-in. wheelbase, with a 6-cyl engine of 183 cu in. developing approximately 130 hp at 4600 rpm. The curb weight of the standard sedan is rather over 3700 lb.

Fig. 49 shows the rear end of the frame, with the swing-axle and the supplementary torsion rod by which the driver adjusts the rate and standing height, to keep the swing-axle in a favorable position whatever the load in the car.

Fig. 50 shows the typical Mercedes front suspension, with the antishimmy mounting of the wishbones. Fig. 51 shows the typical Mercedes swing-axle.

Now, for just a quick glance at the sport cars.

Fig. 52 shows the Jaguar XK120C, and Fig. 53 its rear end with the torque-controlling means on a rigid axle, by which the transverse torque is neutralized by the layout of the linkage.

Fig. 54 – The Jaguar engine, a 6-cyl of 210 cu in. with overhead camshafts, spherical combustion chambers, water-heated intake, and expanding carburetors, giving close to 180 hp. Note the brackets for stiffening the powerplant in bending.

Fig. 55 – The Aston Martin DB3 with DeDion rear axle, parallel link front suspension, springing at both ends by torsion rods, on a simple tubular frame.

Fig. 56 – The Jowett Javelin front end with 4-cyl

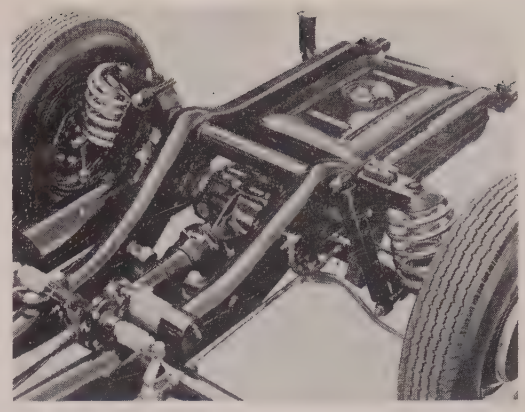


Fig. 49 – Rear end of frame of Mercedes Benz, showing swing-axle and supplementary torsion rod

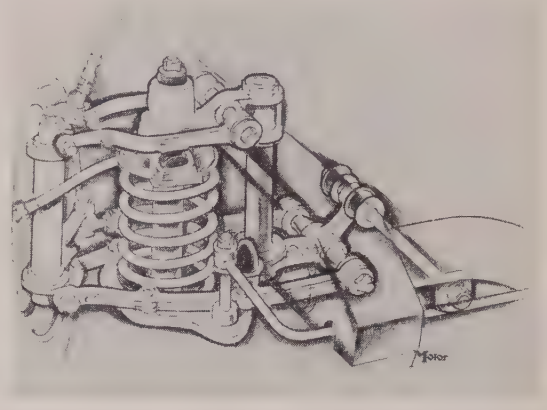


Fig. 50 – Mercedes front suspension, showing antishimmy mounting of wishbones

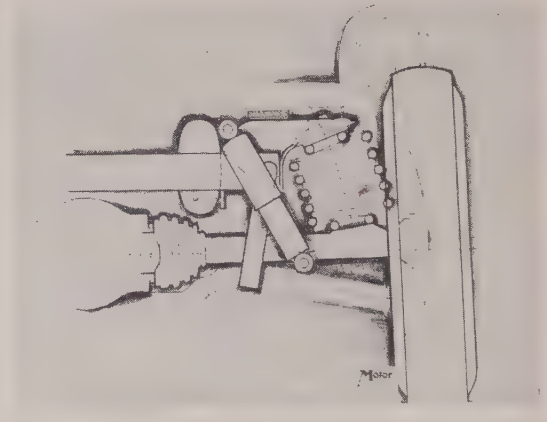


Fig. 51 – Typical Mercedes swing-axle

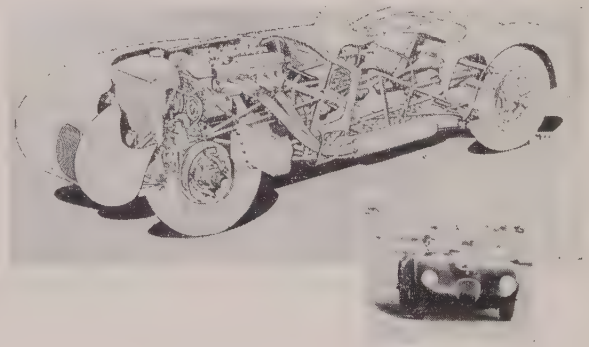


Fig. 52 – Jaguar, a favorite sport car

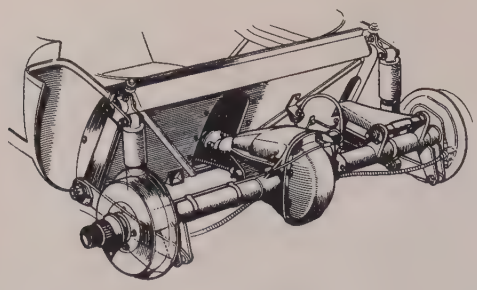


Fig. 53 – Rear end of Jaguar sport car, showing detail of torque control

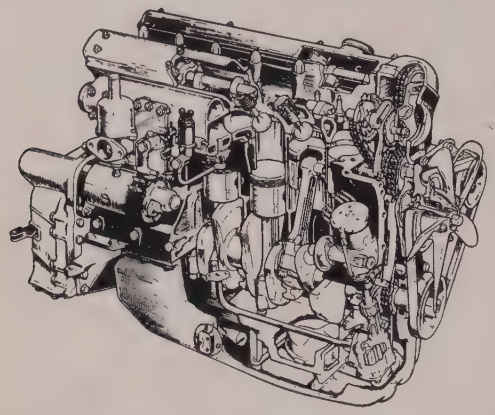


Fig. 54 – Jaguar 6-cyl engine with overhead camshafts and spherical combustion chamber

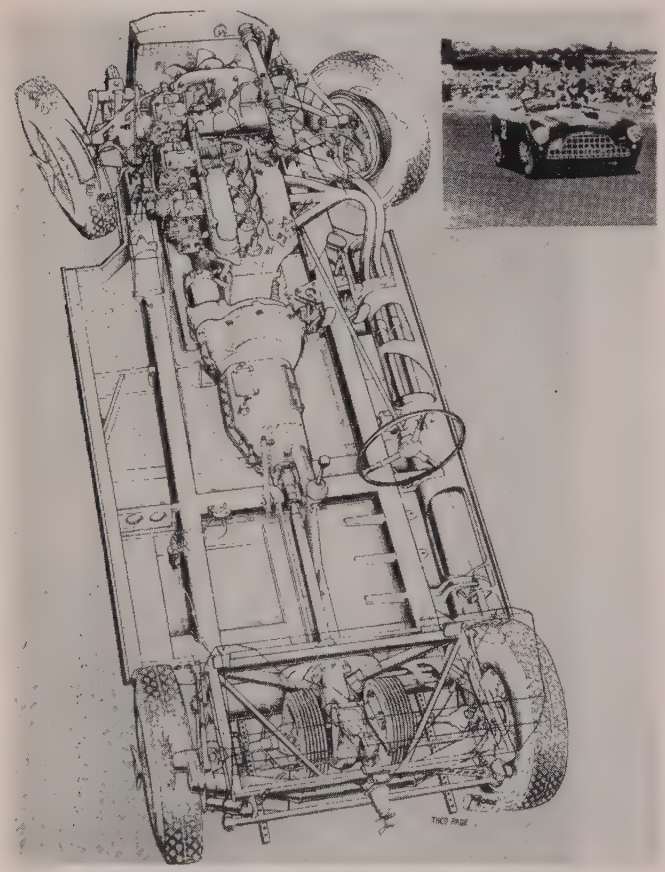


Fig 55 – Aston Martin DB3 with DeDion rear axle

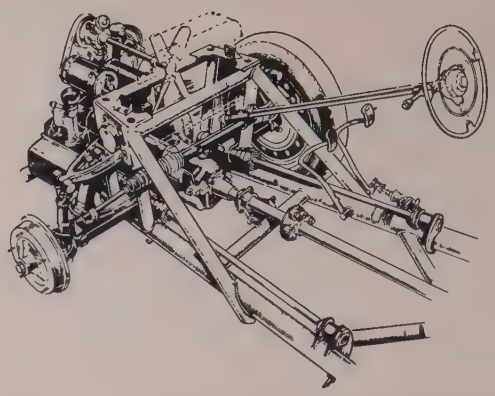


Fig. 56 – Front end of Jowett Javelin with 4-cyl horizontal engine

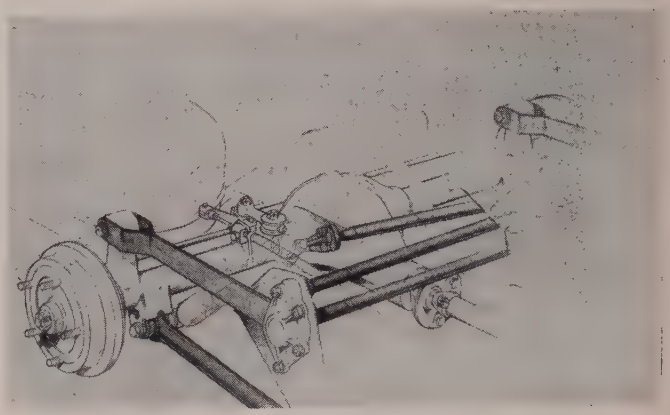


Fig. 57 – Detail of rear axle and suspension on Jowett Javelin

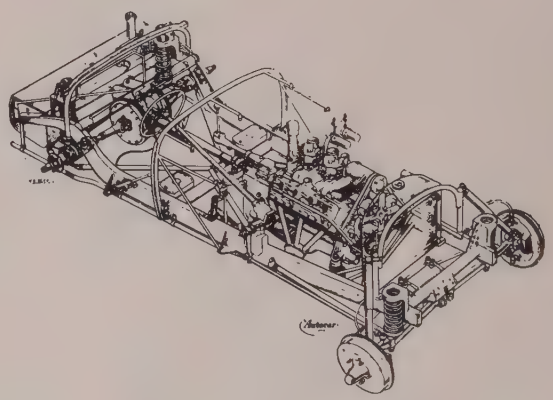


Fig. 58 – Allard, with conventional boxed frame construction, and DeDion rear axle

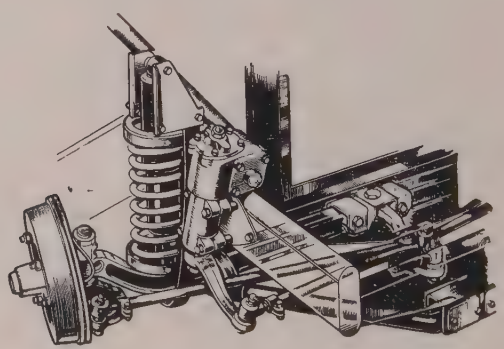


Fig. 59 – Detail of front swing-axle on Allard

horizontal engine and torsion-rod wishbone suspension. Fig. 57 – A detail of the rear axle and suspension on the same car.

Fig. 58 – The Allard car with a more conventional boxed frame construction, DeDion rear axle, and split front axle giving a swing-axle effect. Fig. 59 – A detail of the front swing-axle.

I cannot pass up this opportunity to present a European comment on sports cars in general, coming from a noted British technical editor. He writes as follows:

"It seems to me that the logical reason for the sports car can be found in three different aspects.

"First, that the cost of running a motor car absorbs so large a fraction of the average man's income that he is forced by the mere fact of ownership to give up almost every other form of pleasure." (My correspondent gives figures to prove that, at least in Britain, this income fraction is of the order of 30%.) He continues:

"If, therefore, he is not deriving pleasure from the car itself, he has made a very bad bargain, and the sports car definitely gives more pleasure to the driver than the normal family sedan.

"An alternative aspect is that every phase of utility transport, for example, walking, riding, rowing, sailing, and so forth, is inevitably displaced by more efficient means of getting around, and lives on solely as a sport or pleasure. Logically, therefore, in the long run the only cars on the road will be sports cars. I hope I shall live long enough to see this come to pass.

"A third point is that it could reasonably be argued that the only logical human activities are those which prepare people for war, and best ensure their survival. If this is true, it will be very logical to have the highest possible number of sports cars driven at the highest possible speed, so as to sharpen the wits of the drivers with a view to making them better future soldiers, and accelerate the reactions of the pedestrians, so that they will quickly adapt themselves to taking cover when the big bangs start again." (Survival of the quickest.)

No claim is made that these few notes and illustrations are anything like a complete cross-section of the European passenger-car field. This is far too rich in variety to present in one short survey. All that has been attempted is to give a picture of a mentally vigorous industry which is exploring along some lines which the American industry has scarcely tried. Such things as the Mercedes swing-axles, the front-drive cars, and possibly the 2-cycle engines certainly seem to demand attention.

Above all, there is the question of road-holding. This has been in the forefront of European attention for many years, and it is only fair to say that the European opinion of American cars for road-holding is as low as the American opinion of European cars for ride. American engineers in general fail to understand the mentality which lays such stress on road-holding and apparently ignores ride.

It is all a question of driving conditions and habits. Low-powered cars cannot maintain high speeds if they continually have to slow down for corners. Small European cars on tour have the throttle open a good part of the time, corners included. Many European roads are rough, few corners are banked. Hence the chief demand on the suspension is that the car should stay reasonably upright and maintain its place on the road. European cars were first with independent front suspension and are still ahead of the game in devising improvements on the conventional rear axle, and in producing cars which run as though they were on car tracks.

Britain was the last country in the world to adopt independent front suspension on its passenger cars, simply because its surfaces are good while its road layout still makes no concessions to use by power-driven vehicles, so that it is impossible to maintain speeds.

We cannot interchange our mentalities in this respect.

The British motorist doodles along his lovely country lanes, admiring the scenery and dodging almost subconsciously the bicycles, children, dogs, wheelbarrows, flocks of sheep and all the thousand assorted hazards of British motoring, with as much pleasure as the Continental driver in his noisy minicar tears around the curves of his bumpy *route nationale,* or as an American sets out on his superb toll roads in a vast boudoir on wheels, strolling along at an effortless 70 mph. Each of them is well satisfied, but interchange their vehicles and see what happens.

The American in the minicar feels humiliated and disgusted; in the British car he feels more comfortable but does not seem to be getting anywhere. The Continental driver in the American car comes adrift on the turns, cannot see around the corners of Alpine passes, and burns out his brakes. The Britisher in the American car feels that the car is a foot wider than the road he is driving on, and in the minicar has to shift gear at every sudden hill and every blind turn.

I have only one more word. The world outside our borders is being conditioned to the British and European rather than to the American type of postwar car.

Acknowledgment

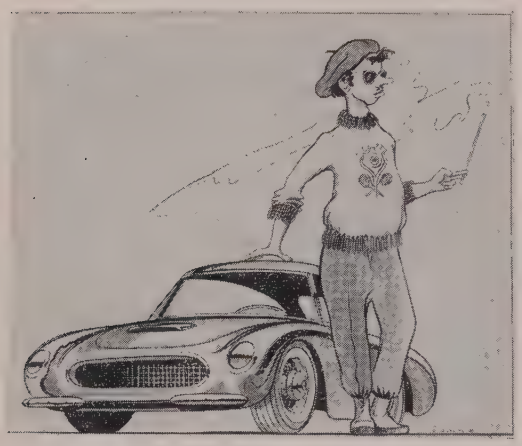
Acknowledgment is made to Laurence Pomeroy of *The Motor* of London for his valuable contributions to this paper. Also to Maurice Platt of Vauxhall Motors Ltd., to Count Mario Revelli and H. Lagassey, Jr., of GM Styling Section for interesting illustrations, and to Chevrolet for permission to present this paper.

For further illustrations and extensive discussion, see following pages ⟶

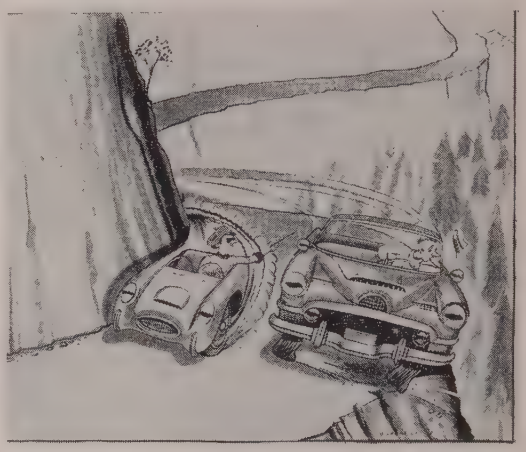
European or U. S., Cars Are Built To Fit Special Cases . . . Not to Swap Places

NARROW British cars are tailored for ease in navigating narrow British roads

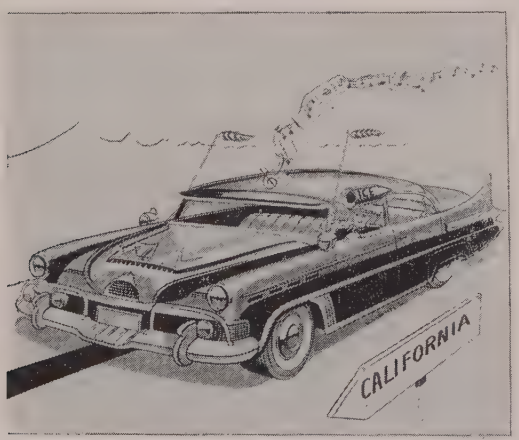
BROAD American cars cause big squeeze when transplanted to British soil



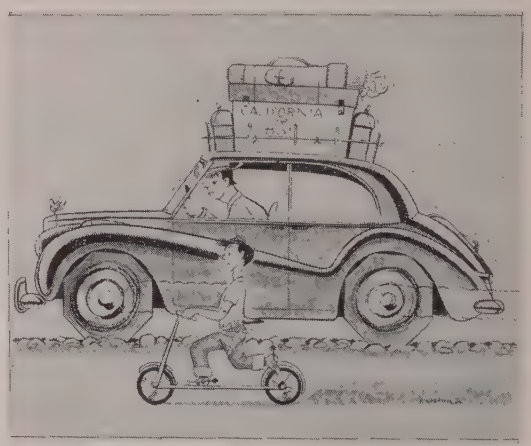
SMALL, cheap-to-operate cars that hold the road well are what Continentals want



LARGE American cars sent abroad really miss those straight, wide U. S. roads



LUXURY, comfort, and high speed top the list of things an American demands in his car



AUSTERITY, rough ride, and low speed sum up an American's impression of a European car

DISCUSSION

Performance Characteristics Of Jaguar-U.S. Cars Compared

– H. A. Toulmin
Ethyl Corp.

LAST summer the Ethyl Corp. purchased a 1952 Jaguar XK-120 fixed-head coupe in order to determine the characteristics of its very high specific output engine on modern highly sensitive fuels. This car was chosen because it has proved its reliability as a sports car in the many races that it has won throughout the world. In this discussion today, I will compare some of the performance characteristics of this automobile with a 1952 U. S. production car which is looked upon in this country as a high-performance vehicle.

Fig. A shows a view of the Jaguar car. It is apparent that the frontal area and general lines are quite low in comparison with our cars. The low frontal area is achieved by using seats of very low cushion height (Fig. B) and enclosing the gearbox and driveshaft in a large tunnel through the center of the body. The car has seating capacity for two people and has a very small luggage compartment which is accessible from the rear.

The car is fitted with a 6-cyl valve-in-head engine of double overhead camshaft construction. The bore and stroke are 3.27 by 4.17 in. giving a total piston displacement of 210 cu in. The combustion chamber is of the hemispherical type, similar to that used by Chrysler. The standard compression ratio is 8/1. With high-lift camshafts and a dual exhaust system such as used in our car, the engine is rated at 180 bhp at 5300 rpm and a maximum torque of 203 lb-ft at 4000 rpm. With standard camshafts and a single exhaust system, the engine is rated at 160 bhp at 5200 rpm. These horsepower outputs are equivalent to 0.86 and 0.76 hp per cu in. of piston displacement, as compared to 0.63 for the highest U. S. engine. Twin horizontal S. U. carburetors were used.

A 4-speed manually operated transmission with synchromesh in second, third, and fourth gears is used in conjunction with a rear-axle ratio of 3.64/1. The curb weight of the car with a full tank of gasoline is 3075 lb.

To put these specifications in a different form, we might say that the Jaguar is about 300 lb lighter than a standard

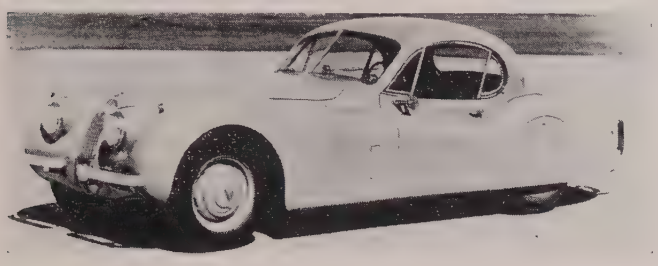


Fig. A – View of the Jaguar car

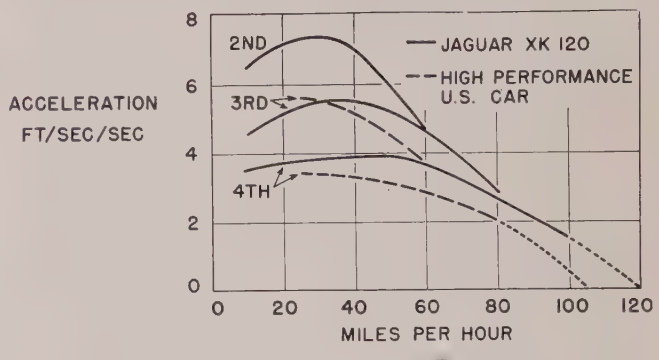


Fig. C – Acceleration characteristics of Jaguar compared to 1952 U. S. car

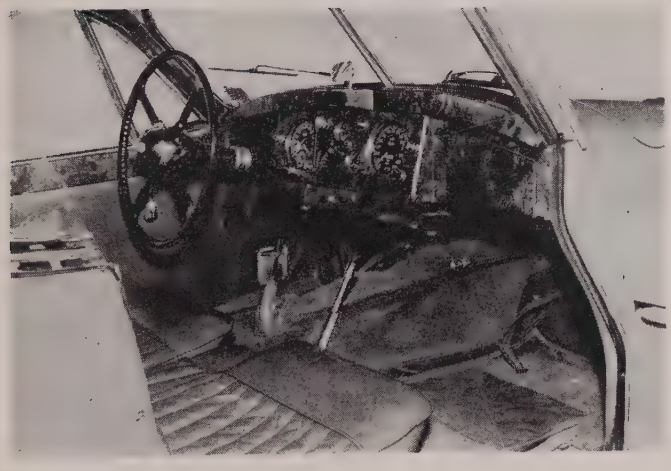


Fig. B – Low frontal area of Jaguar is achieved by using seats of very low cushion height

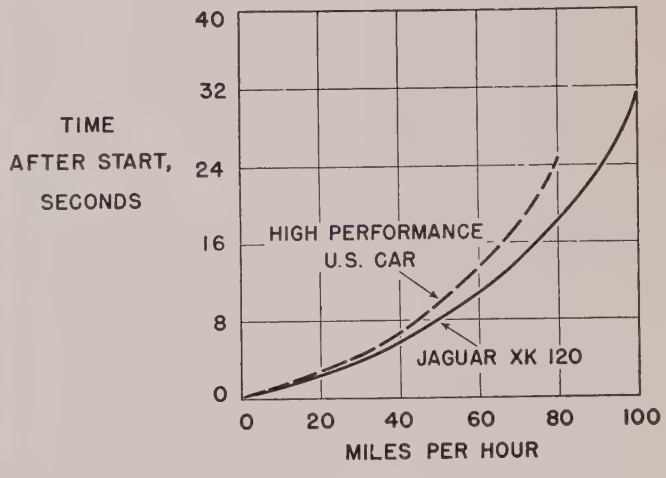


Fig. D - Comparative performance of Jaguar and U. S. car accelerating through gears, starting from rest

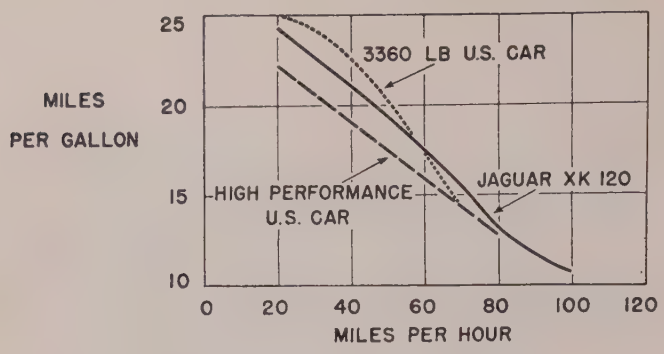


Fig. E – Fuel economy of Jaguar compared to two U. S. cars

Chevrolet and has an engine of 6 cu in. smaller displacement.

Since this car was designed for an entirely different market than our U. S. cars, it is extremely difficult to make direct comparisons. The general proportions, weight distribution and layout of the Jaguar, make it an extremely stable vehicle and one that can be cornered very fast with a feeling of complete confidence. Ordinary highway turns can be negotiated at speeds of 80 to 100 mph with practically no roll or sway. The engine is extremely flexible in spite of its tuning modifications and it is possible to accelerate smoothly in top gear from 10 mph to the maximum speed of the car of 120 mph. The car's brakes are excellent, making high road speeds possible with safety. On the other hand, by U. S. standards the engine and gearbox are quite noisy and the ride is very harsh.

Fig. C shows acceleration characteristics of the Jaguar compared to the high-performance 1952 U. S. car previously mentioned. Even though the Jaguar engine speed for a given car speed is higher than that for the U. S. car, the performance is maintained over a wider speed range and is superior to that of the U. S. car at practically all speeds. In fourth gear for instance, the performance of the Jaguar is approximately 30 per cent better than the U. S. car at speeds from 50 to 80 mph. Above 80 mph the Jaguar performance excels by a greater margin.

Fig. D shows the comparative performance of the two cars accelerating through the gears, starting from rest. This curve shows that the Jaguar has only a slight advantage up to 40 mph, but from this speed upward the difference becomes progressively greater.

The fuel economy of the Jaguar is compared to two U. S. cars in Fig. E. It can be seen that the fuel economy of the Jaguar is better than the high-performance U. S. car; however, this is not a fair comparison as the test weight of the U. S. car is 1500 lb heavier and the frontal area is greater. When the economy is compared to that of a U. S. car of approximately the same displacement and only slightly greater weight, it is seen that the U. S. car is more economical than the Jaguar at low speeds but is poorer at high speeds due to the greater frontal area of the U. S. car. The poorer economy of the Jaguar at low speeds is probably due to the fact that the fuel/air ratio is maintained approximately the same at both road load and full load, instead of using lean mixtures for light load as is the common practice in this country.

Fig. F (see p. 526) shows the octane-number requirement of this car. It is very low at the lower engine speeds and peaks at 83 octane number at approximately 3200 rpm. This requirement curve was obtained after the car had been driven at high speeds during the performance tests and does not represent the maximum requirement of the vehicle. When the car is driven at slow speeds in city traffic, light knock has been obtained on fuels of premium quality; however, after the car has been driven at high speeds the requirement is reduced to that shown on the curve.

Claims Car Safety Is Sacrificed to Styling

– J. A. Kloth
National Advisory Committee for Aeronautics

I WAS interested in Mr. Olley's discussion of the economics which affect the type of cars produced in various countries. I agree with his theories about the European countries, but believe that a different type of economy controls the design of the American car. This economy is made up of a great variety of factors: the feminine viewpoint, styling, comfort, vanity, and even wishful thinking. These are just a few of the more unusual points upon which our automotive economy is based; but surprisingly, they greatly affect the type of car which is produced.

The feminine view certainly seems to be a strange economic factor; but since women control the majority of the finances, their opinions govern the purchase of a good many automobiles. And the women do go for beauty, style, and comfort primarily! Since most women do not know a crankshaft from a rear axle, I would imagine that the old hatpin gets a good workout when it comes time to buy a new car.

Styling, of course, is an important point in the manufacture and sale of a car, but at the present time the American manufacturer has gone completely overboard on this angle. I quote from the paper by V. P. Mathews and J. D. Turlay on the new Buick V-8 engine: "In fact, space requirements as dictated by the body stylist are one of the most important, if not the most important factor, in determining the future trend of engine design." Charles Chayne of Buick, speaking to us at an Akron-Canton (Ohio) section SAE meeting, told how Harley Earl, GM's chief stylist, allowed only 18 in. for the placing of a 300-plus hp engine in his design of LeSabre. In Europe, the body designers take the chassis from the manufacturer and then build their beautiful cars around a unit which has already demonstrated its stability and handling. Over here we try to fit the chassis and engine to the body beautiful. This is indeed getting the cart before the horse. Is it any wonder we get those outlandish weight and balance figures. I find it disgusting to get stuck on a slight slope in the wintertime or get the front wheels in a bit of mud and be unable to get them out without the aid of a tow truck. Lack of good handling qualities is the greatest defect arising from a lack of proper balance. We are sold cars capable of an honest 100 mph though they are unsafe above 50. Quoting Mr. Chayne again, "If you build a car which will go 80 mph, you have to design it so it is controllable at 100 because some darn fool will be going down a steep hill with a tailwind to boot and find himself doing it." Why aren't our cars built that way?

Comfort is right behind styling in importance as a design consideration in America; and like styling, is also a disastrous consideration. The American people, especially women, demand a comfortable and soft ride. The good road-holding characteristics which unnecessarily were abandoned to give this comfort have contributed tremendously to the terrific accident toll. Through good design practices, good handling and comfort can go forward hand in hand.

Vanity is a large item in our automobile economy. Keeping up with the Joneses is a trait for which the Americans are well-known. The type of car we own is a typical measure of our financial standing. The biggest, the fastest, and the smoothest riding car is our symbol of affluence. This feeling of vanity overwhelms even our desire for safety.

Wishful thinking is an unusual factor in determining the type of car we buy, but it is strangely true. The majority of people seem to think that the United States is just crisscrossed by beautiful toll roads where you can drive straight and smoothly all day long. To quote Mr. Olley, ". . . an American sets out on his superb toll roads in a vast boudoir

on wheels, strolling along at an effortless 70 mph." Just one question though – where are all these beautiful roads? Of the 15 to 20 thousand miles which I drive each year, only a few hundred are on freeways and turnpikes; and if the rate of progress of the Ohio Turnpike is any measuring stick, it is going to be a long while before that few hundred increases.

Mr. Olley's paper is well prepared on the highlights of European cars. It shows many of the better ideas. But – a good many of the cars he mentioned would be more at home on our roads than he would like to admit. Most Americans are amazed at the complete control which you have over a foreign car at all speeds. I myself received quite a scare when a friend gave me a demonstration of the capabilities of his Jaguar XK-120; and after driving it myself under all kinds of conditions, I haven't been satisfied with an American car since. The safety and ease of the quick steering, the positiveness of control, and the feeling of complete security convinced me of the superiority of foreign cars. The smaller size of the average European cars is much more practical in the economy of space, too. Parking lot operators in Cleveland are turning down owners of larger cars for lack of space. Also if the number of vehicles on our now overcrowded highways continues to rise, I'm going to be in the market for one of those good English bicycles Mr. Olley mentioned, and take the place of the vanishing pedestrian on the sidewalk.

I have joined the ranks of foreign car owners and will never buy another American car unless they are designed and built properly safety-wise. I don't doubt for one minute that, if the American driver were allowed to drive a foreign car for one week, the majority would make the same choice I did.

Let's take not merely one, but many pages from the European book of automotive design; ease the congestion on our road systems; and above all, reduce the rate at which we are filling our cemeteries with traffic accident victims.

European Contributions to Modern Automobile Reviewed

– Gaston Fleischel
Automobiles Peugeot

THE author has made an accurate survey of a highly controversial matter. Disregarding everybody's national pride, which is often a cause of unwitting misunderstanding and partiality, it is only natural that big differences in habits, tempers, wealth, climate, taxes, and roads of the countries involved, result in very different cars, which have little appeal to those who live under quite different conditions.

I have the feeling that we do not always give proper credit to the European nations for the improvements they have contributed to the modern automobile. Some of Europe's manufacturers have been in automotive business since a very early date. For instance, the forerunners of what has grown now into la Société des Automobiles Peugeot, with a yearly production of 80,000 cars, presented a steam tricycle at the Paris International Fair in 1889, the same year the Eiffel Tower was built. In 1891 their first gasoline quadricycle (light two-seater) was delivered in France to a customer after the same model crossed, solely by its own power, that country from the Swiss border to Brest in Brittany, a distance of about 550 miles, at an average speed of 8.5 mph. Since then, Peugeot has been responsible for the introduction in mass production of several devices now widely used everywhere in modern cars. In 1932, all Peugeot models were equipped with the front-wheel knee action; in 1935, the same firm introduced as optional the semiautomatic 4-speed planetary transmission, an improvement over the preselection type used before. Immediately after World War II, Peugeot adopted the push-rod-operated hemispherical-chamber engine with only a camshaft located as usual. The same construction was afterward adopted and widely advertised by three American companies. Many other examples may be given.

While American manufacturers are leading far ahead with roomy, luxurious, silent, and very comfortable cars, and achieving higher mass production, the European engineers are confronted with customers impoverished by two world wars, sometimes both fought on their soil, with heavy taxes on gasoline and engine displacements, not to mention the tax collector looking for large cars as a sign of wealth for higher taxation. They are induced, therefore, to produce smaller cars, and are more concerned with safety, low cost of performance, and general operation. Their problems are difficult; they must forget about the easy way out. Still, they are responsible for many of the technical realizations used everywhere, from the general idea of the standard car to the automatic transmission.

Sees Trend Toward Smaller Cars in U. S.

– W. D. Appel
Ford International

MAURICE OLLEY always makes complicated problems appear so simple by getting down to fundamentals.

His analysis that the world economic situation is responsible for the curtailment in world trade is a very straightforward and blunt statement of fact. It can only be corrected by having every country import from every other country the equivalent of exports to that country. In that respect, we have been rather selfish in expecting the world to buy our goods without a reciprocal action on our part to buy their products.

The paper summarizes technical details of European cars which, in general, are both smaller and different from our own. They are not smaller merely because the Europeans would not enjoy roomier cars where road conditions permit, but they are smaller because of two basic reasons:

1. Operating and maintenance costs.
2. The necessity for lower first cost.

Lower operating and maintenance costs can only be achieved by means of lighter cars having lower performance standards. This need eventually leads to a smaller car as the ultimate solution.

Lower first cost is achieved by a car having fewer and simpler parts but, here again, a reduction in the size of the parts helps to reach this goal.

European cars can generally be classified in four groups:

1. The shoehorn, 4-passenger type of car in which two passengers barely have room to sit side by side. Examples of this are the 4 CV Renault, the Dyna Panhard and the 2 CV Citroen.

2. The next larger size is a car in which two passengers can sit comfortably on the front and on the rear seats. Examples of this are the Morris Minor, the Austin A-40, the Opel Olympia and the German Ford Taunus cars.

3. The next larger category is what I would call the 4- to 6-passenger car in which there is ample room for two on each seat, but three can be carried in an emergency. This might be considered in the upper middle price brackets of European cars and does not represent the volume market.

4. On top of this, there is the full-sized luxury vehicle which is as large as our American cars but in which production is very low and the price is very high.

The first two classes are unknown in American production, while the third, the 4- to 6-passenger car, has almost disappeared in favor of the car having comfortable room for three people side by side.

The largest, or luxury-price class, is exemplified in this country by Lincoln, Cadillac, Chrysler and Packard.

The reason for the wide variety of technical design in European cars is twofold. In the first place, the European engineer and entrepreneur is essentially an individualist who wants to create a vehicle which is distinctly different from that of his competitors. He has little desire to copy the features of other vehicles even if they are good, because of this innate desire to create in his own individual design.

The second reason is the extreme competition and low volume which exists, due to the large number of manufacturers in the field. In England alone, there are 26 manufacturers producing 31 makes of cars. In each make there are sometimes three and four models; whereas, in the United States there are eight manufacturers producing 20 makes averaging slightly over two models per make.

This type of competition in low volume allows the ingenious European engineer to develop and try out new designs from which the large American manufacturer would shrink until time had proved their value and the public has accepted them. The American engineer has the opportunity of reviewing these very interesting features which are developed in Europe and adapting them to his own production if, as, and when a worth-while design is developed and he feels it either necessary or desirable to use it.

As far as lighter and smaller cars are concerned, I pointed out that the United States appears to be operating wholly in cars of the larger size. If and when economic conditions in this country force us to the smaller and more economical cars, it will probably be through successive downward steps, instead of by making a sudden change from a big car to the small classes.

There is inertia in the expression of human desires and the human being is not flexible enough to accommodate himself to big changes at one time. He will, however, absorb a considerable change over a period of time by taking the change in small doses.

There are already signs that cars are moving backward from the full 6-passenger size to the 4- to 6-passenger cars which we had about 15 years ago.

Additional Viewpoints Presented by Discussers

– A. C. Sampietro and B. G. Bowden
Willys-Overland Motors, Inc.

MR. OLLEY'S papers are always so entertaining, that in presenting all his technicalities in reference to European postwar cars and their relation to European economic conditions, one does not realize how much food for thought is available until after mature consideration.

In arriving at our conclusions, it can be seen that we are not entirely in agreement with all of the author's viewpoints. This is probably unimportant – the important aspect is that Mr. Olley has made us not only think about, but also delve into matters that could become quite controversial unless certain plain facts emerge.

Concerning Britain's predicament generally, the country may be overcrowded as Mr. Olley says, but at least one remedy is in her own hands; to provide overseas purchasing power by exporting one of the greatest base materials – coal. Britain could export at least 50,000,000 tons of coal a year, which would substantially contribute to her adverse trading balance, and at the same time make a worthy contribution to the rehabilitation of Western Europe. Miners to provide the labor for this increased output are available in Germany, Italy, and Spain; in fact the European miners are eager to work in British mines, but the British Union of Miners will not issue permits for foreign workers in the nationalized government mines.

Mr. Olley states that the U. S. is unwilling to permit the import of goods with which the rest of the world can pay its dollar debts. What goods are these? Certainly not automobiles, as there is no quota control and only a nominal tariff of 10 per cent. Britain's sterling barrier may be a case of the biter bit. We feel that Britain has caused more economic trouble to peoples of other nations trading with her than anyone else. With sudden changes in the rate of exchange and arbitrary unilateral freezing of credits, the dictatorial bureaucratic financial control seems to have plunged the country into a morass of economic problems. It has to be experienced to be believed.

In the phase of economics connected with the automobile industry and its export operations, Mr. Olley states that the unwillingness of the U.S.A. purchasing public to buy European cars is one of the contributory causes of Europe's economic problems – wouldn't it be fairer to exempt not only the American public, but also the American automotive industry on this count?

All European cars can be sold in this country without let or hindrance, at least as many as the public are willing to buy. On a purely competitive basis, the European cars must be sold on merit, without forced subsidy (which has been known to take place so that export quotas can be attained), and the only on-costs over the European home market price is the tariff duty of 10 per cent on importing into America, plus, of course, shipping charges.

The situation is the antithesis of the attitude of many European countries to competitive buying against American cars. A large demand for American cars does exist in Europe, but the governments, by devious means, make it practically impossible for American cars to be economically placed on their home market. There is either a complete prohibition on importation, a customs duty many times the 10 per cent imposed on cars into America, or else a landed price limit which demands the establishment of an assembly plant for American cars only in the popular price field. American used cars are offered for sale in Britain (usually ex-diplomatic corps) and are generally purchased at what we would call fantastic prices; but the demand proves that people will pay anything up to $2,500 for a 1948 Chevrolet – even a prewar 1939 Hudson will fetch $1,200! !

One must not lose sight of the fact that statistics have a profound function in making true comparisons. In 1951 the world total of passenger cars in use was over 54,000,000, of which the U. S. claimed 40,000,000 – the remaining 14,000,000 being spread over the rest of the world; in fact 152 separate registrating countries. The ridiculously small percentage of each country in relation to the U. S. figure can be appreciated when the next highest figures are: U. S. S. R., 2,500,000; United Kingdom, 2,300,000; Canada, 1,800,000; France, 1,600,000; Australia, 800,000; Germany, 600,000; and Italy, 400,000. These figures include all makes of cars, so that the percentage of American cars in use throughout the world must be considerably greater than the ratio of 3 to 1 of cars in use in America to the rest of the world; bearing in mind that of Canada's total, 95 per cent are American basic designed automobiles. One wonders really why such emphasis is always laid on Italian and German automobile design trends when both countries enjoy, only approximately, the use of 500,000 cars each.

One must appreciate the paramount facts that cost and high volume production are probably the main contributory causes of design development trend, and much as many American engineers would like to follow certain European trends, the dictates of home market requirements relating to cost is always of vital importance.

In describing European car features, Mr. Olley's paper implies criticism of American cars on three main counts:

1. Visibility.
2. Road worthiness.
3. Bulk.

Perhaps most of these criticisms were well deserved up to 1950, but now at least one production American car has achieved that compromise, for want of a better word, be-

tween alleged American and European trends. We refer to the Aero Willys.

The Willys certainly gives as good an intimate view of the road as any European car – certainly much better rearwards, with road-holding and steering comparable to that of a good competition model; and only the smallest of the European cars have less bulk. I wonder if Mr. Olley has driven a Willys for 700 or 800 miles. It is interesting to note the comparative dimensions (See Table A) with those given in the author's chart, Fig. 11.

We particularly draw attention to the weight and price comparisons.

Mr. Olley states that American cars are impractical in the Alps; a view consistently put forward by the British semitechnical press, and a view that to us is very puzzling. Rightly or wrongly, we feel this is a red herring, that is, quoting as an example a country with only 146,000 cars in use out of the world's total of 54,000,000.

We could pass on some first-hand knowledge of mountain driving, but it would probably be better to inquire whether Switzerland still buys a large proportion of American cars, because although they have mountains there, they also operate buses as large as our Greyhound; as shown in one or Mr. Olley's excellent illustrations. It would not be irrelevant at this point to say that power steering would be a complete blessing on a road like the Stelvio, with its succession of hairpin bends. One of us knows, as he learned to drive on this famous pass.

Concerning references to road conditions; the thousands of miles of washboard roads betwen cities only applies to very low-populated countries – we do not know of one gravel road in Britain in general use. Therefore, the cars per road mile in relation to road condition has a bearing on this fact. Even in South Africa, one of us knows from personal knowledge and use, that the roads between large cities are excellent. Johannesburg is 400 miles from Durban, with only 4 miles of gravel; Johannesburg to Capetown is 1000 miles with 20 miles of gravel the remainder being first-class surfaces as on the best American highways.

When dealing with the individual characteristics of European cars, Mr. Olley failed to bring out certain disadvantages such as the very high cost, and frequent overhauling necessary to keep an engine in tune. We know from personal experience an M.G. needs a valve grind every 10,000 miles, a Jowett Javelin every 7,000 miles; and to check the rear spark plugs of the Javelin it is necessary to lift the complete car and take the front wheels off.

On the subject of road-holding, this must be combined with safety. Present-day traffic conditions and the high percentage of accidents force us to pay more attention to road-holding. The emergency stop that is achieved at the expense of overturning is no criterion of safety, so that in retaining road-holding, a comfortable ride must also be a non-tiring ride. A tired driver is a menace, and how tired he can become with certain European suspensions over American distances!

We would like to give an extract of a statement made by the Earl of Selkirk, speaking for the British Government in the House of Lords on Dec. 16, 1952. This statement was printed in the British magazine *Motor*, and we quote it without comment! "The Nigerian Government had requested permission to import a limited number of American cars for the use of persons engaged on essential work which involved driving prolonged distances on bad roads. The Government were unable to resist the request from Nigeria, but they hoped and understood the new British cars were likely to meet the demands of Nigeria more fully in the future."

The international economic situation is one that does not warrant discussion through the medium of the SAE other than the effect it has on automobile design. We feel that America has kept, and will continue to keep abreast of the times; and in particular meet the threat of any major domination by European design trend that may leave the American automotive industry behind the pack when world trading resumes its normal competitive marketing.

Table A – Comparative Dimensions

	7 Medium European Cars, Average	Aero Willys
Wheelbase, in.	103.3	108
Tread, in.	53.15	57
Overall Length, in.	171.8	180.8
Overall Width, in.	66.35	72
Number of Cylinders	4 or 6	4 or 6
Displacement, cu in.	132.5	161
Forward Speeds	3 or 4	3 or overdrive
Axle Ratio	4.2	4.1
Curb Weight, lb	2604	2511
List Price, Sedan	$1543	$1499

We feel that it behooves us as engineers to consider the different standards of living and consumer demand as major facts we cannot ignore. We might try to improve these conditions as citizens, but as engineers and members of the SAE we can and must continue to improve our vehicles and see whether the seed that Mr. Olley's paper has planted will bring forward safer and more economical vehicles in America.

We must not overlook the trend now growing in America toward the smaller type of car – the Aero Willys, Nash Rambler, and shortly, the Hudson Jet, to name only three makes in production at figures approaching perhaps the total European production of all makes in all countries. Maybe other manufacturers have similar models on the shelf that will keep company with Willys, Nash, and Hudson; who knows.

We cannot agree with Mr. Olley that European cars receive less annual face lift than American cars – he quotes the small British Ford as still on the market in its original 1938 prewar style. We would have added, "and it looks like it." Before the war both British and Continental manufacturers were very active and aggressive in their search for sales with new models, features, and the like. After the war, the policy of controlled scarcity has made a situation come about, whereby anything can be sold – therefore why should the manufacturer become involved in marketing new models and developments which have no effect on sales, as the annual production quota is allocated before the first car rolls off the line.

We have attempted in our comments to perhaps amplify some of Mr. Olley's thoughts – we hope that with this first-hand knowledge of conditions in Europe and Mr. Olley's very excellent presentation, that an all-around picture of existing problems and future developments will prevail upon all forward thinking engineers to stop, look, and listen, before condemning any or all of the European picture.

Needs of Worldwide Markets Considered

– P. H. Pretz
Ford Motor Co.

THE author's comment that the rest of the world is being conditioned to European rather than American cars should provoke some serious thinking.

When insufficient material is available for our domestic market we are apt to ignore too completely the excess of legislation further curbing our interest in export markets. If the needs of worldwide markets are not considered, we may never regain the prewar outlet of approximately 10% of our cars.

Concerning the technical aspects of the paper I should like to ask Mr. Olley three questions:

1. By stating that the European cars have a more intimate view of the road, do you feel that they have better

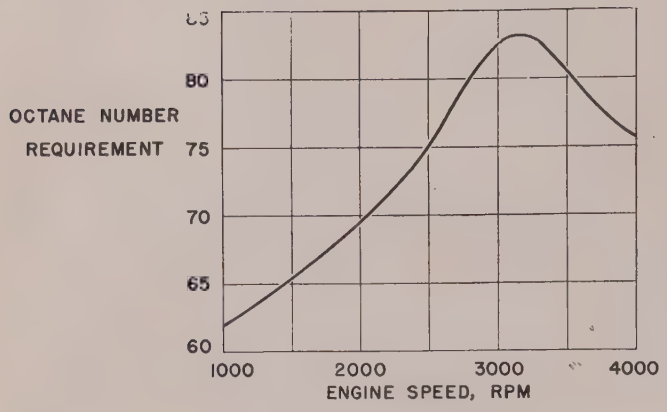


Fig. F – Octane-number requirement of Jaguar for primary reference fuels

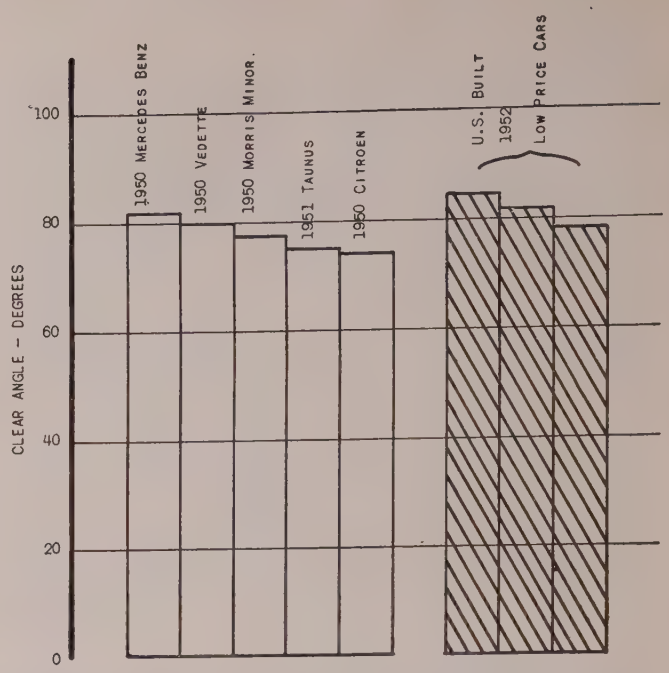


Fig. H – Additional comparative table of horizontal field visibility – total clear angle through windshield based on U. S. and foreign-built cars

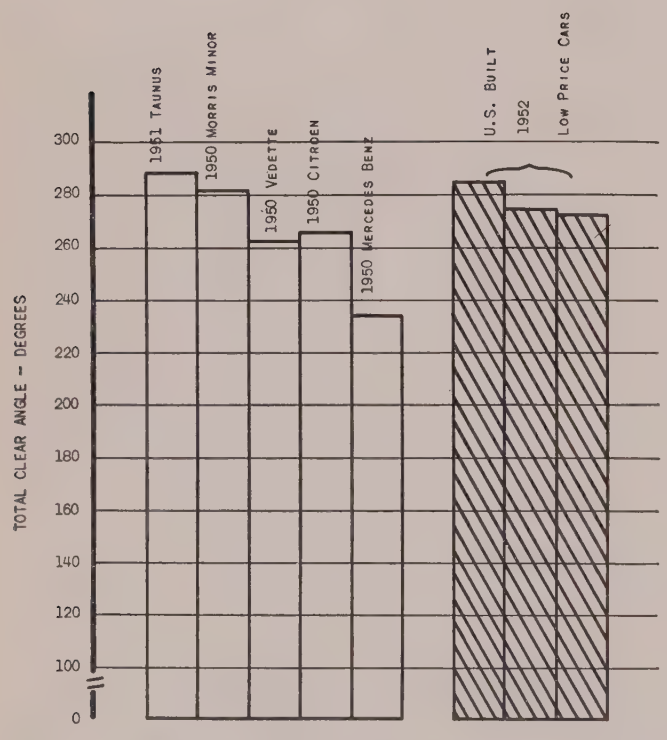


Fig. G – Horizontal field visibility – total clear angle through windshield based on U. S. and foreign-built cars

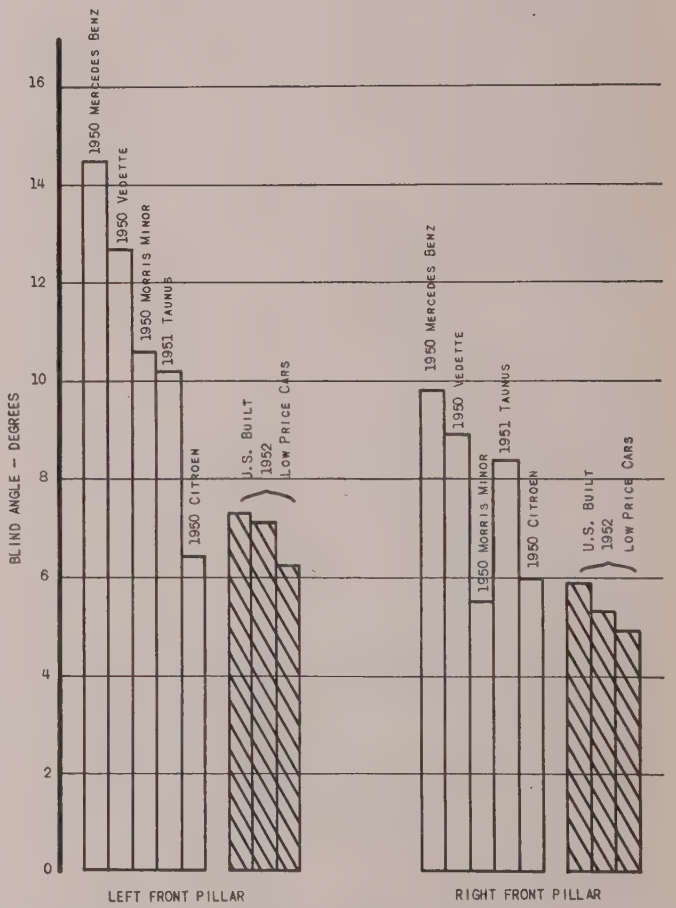


Fig. I – Horizontal field visibility showing blind angle due to front pillars, based on comparison of U. S. and foreign-built cars

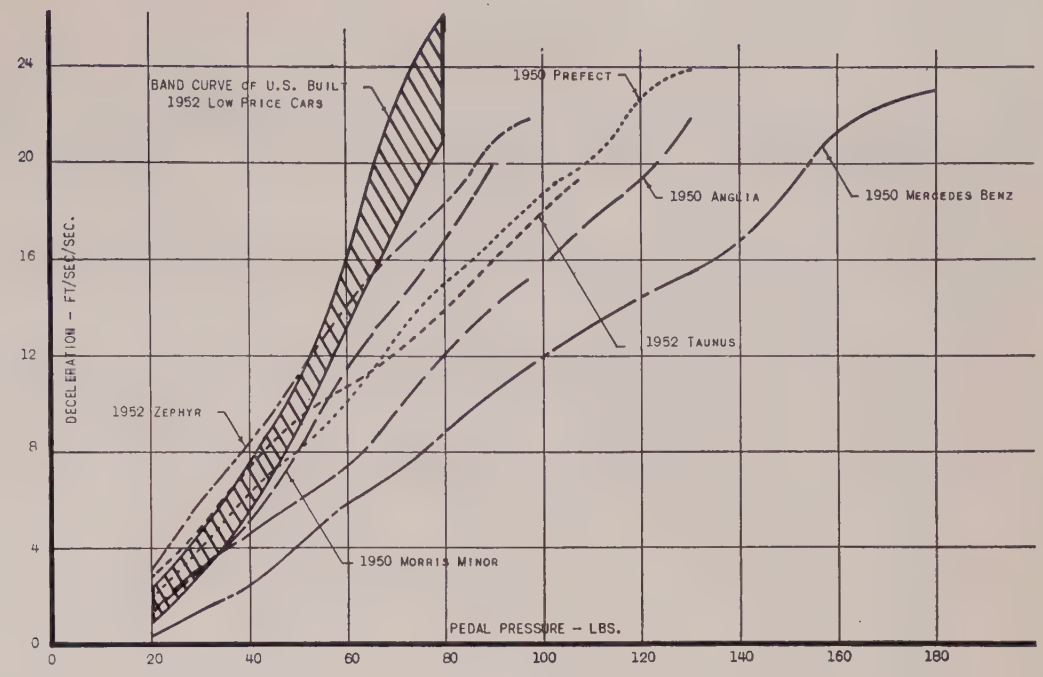


Fig. J – Comparison of brake performance of U. S. and foreign-built cars (50 mph stops)

all-around visibility than our cars, with the trend toward wrap-around windows, greater glass area, and shorter lower hoods?

This question is prompted by three illustrations.

Fig. G shows total clear angle in the horizontal field through all windows, and very little advantage appears for either foreign or domestic cars.

Fig. H shows the clear angle in the horizontal field through the windshield only. Here, although the foreign cars have narrower windshields, the driver is seated closer to it, and again there appears little advantage for either type.

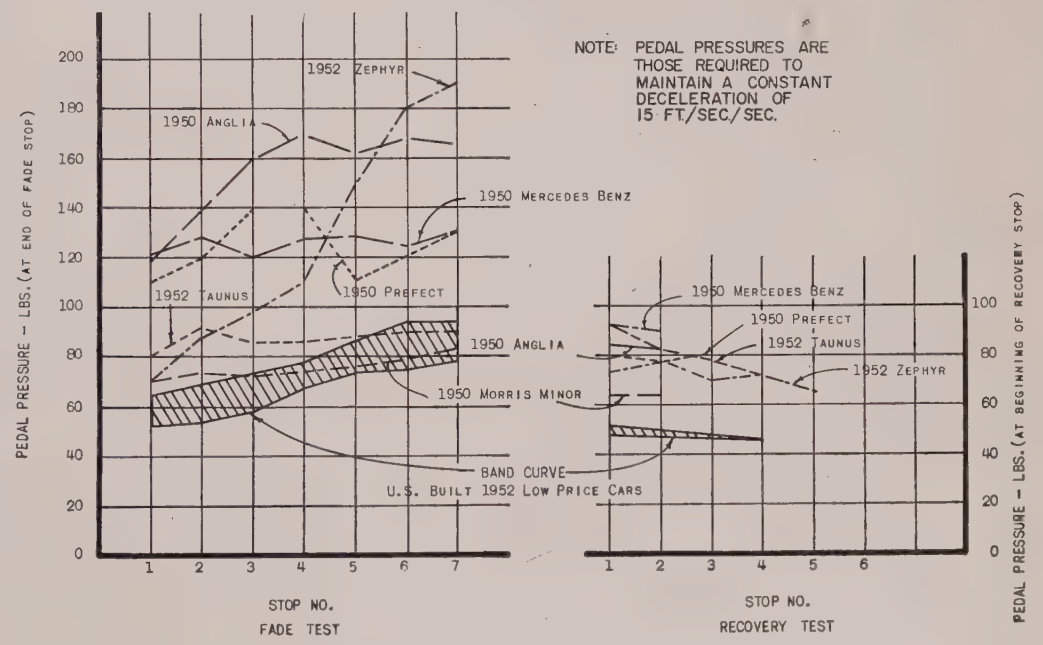


Fig. K – Brake fade recovery characteristics of U. S. and foreign-built cars

Fig. I, however, shows greater blind angles from the front pillars of foreign cars, probably caused by seating the driver closer to them.

2. Was your remark that the European cars have better brakes based upon life, pedal effort, or fade? As far as life goes, braking on their sport cars in the Mexican Road Race was no good either!

Fig. J shows decidedly lower pedal effort for a given deceleration on U. S. cars. The band curve represents results on the big three.

Fig. K shows our measured results of fade and recovery. Here the band curve represents an average of six U. S. cars while six European cars of two nationalities are shown by individual lines. One could hardly conclude from this that U. S. cars are inferior on fade, unless Alpine conditions provide something different than our test methods bring out.

3. I gather that you felt three cars, in particular, had a mine of interesting engineering features: namely, Volkswagen, Lancia Aurelia, and Fiat 1400. Since we are interested in buying one of these would you mind giving us your personal opinion as to which of these mines has the richest ore?

Author's Closure To Discussion

In reply to Mr. Appel – Would the Europeans enjoy roomier cars? I would reply yes, with reservations. Most European amateur drivers would not and do not like a car as large as some which we are building today. In their opinion it is just too big for convenience. I think that when it comes to parking and city traffic, some American drivers would agree with them.

I would say the preferred car in Europe regardless of cost, is the comfortable 4, tight 6-type, or Mr. Appel's class 3, such as the British Ford Consul and Zephyr or the Vauxhall Wyvern and Velox. The body dimensions of these compare with the average of the bodies used on, say Rover or Rolls-Royce Bentley cars. They are based on the idea that no one really enjoys sitting in the middle of a 3-width seat.

In reply to Mr. Toulmin – His remarks on the Jaguar are extremely interesting as setting down a typical specification for a high performance or "sports" car of European type. He remarks on the lack of a lean mixture road-load setting on the carburetors. This is unfortunately true on a number of European carburetors. The American carburetor manufacturers appear to have done a better job in this respect, but at the same time the "feel" of these expanding carburetors, and their smooth performance over a wide speed range, is something which we might well copy over here.

In reply to Mr. Pretz – 1. I think the European cars in general have a more intimate view of the road, forward because the hoods are shorter, sideways because the belt line relative to driver is lower and the car narrower, and generally because the driver's eyes are closer to the road. Then again perhaps my original statement was not quite adequate. It might be more adequate just to say that the driver feels more intimate with the road, because for one thing he has a lower ratio steering gear and therefore less "rubber" between his hands and the front wheels. Perhaps also because stiffer springing and smaller tires let him know more definitely what the road is like.

2. With regard to European brakes, these of course are variable in performance between various makes of cars. Let me put it this way. The European engineer thinks of brakes as something to stop the car, just as we do here. But he also thinks of brakes as something to control the speed of the car on long down grades. These down grades are commonly steeper and narrower than American mountain highways, and they are liberally decorated with hairpin turns which are frequently blind. So speeds must be controllable at will for comparatively long periods between say 35 and 5 mph. Mountain passes in the Alps bring out the differences between cars, and most European engineers are very conscious that these conditions have to be met. Their lighter cars give them a great advantage.

3. With regard to the three cars named I would say that all three contain items of great interest.

In reply to M. Gaston Fleischel – I think M. Fleischel's remarks bring out excellently the point that a little austerity is good for the soul (although admittedly the austerity in Europe has gone too far). Various discussers have commented that European designers are not afraid to try things, that the cars have individuality, and so forth.

I think the question in all our minds is whether the wealthy American industry should not also try these things, and see how the public reacts. Therefore one welcomes examples of European production on the roads of this country, as well as the new American cars of typical European size.

M. Fleischel speaks of American cars as "roomy, luxurious, and very comfortable." I wonder why the smaller types of cars made in Europe, cannot be made, by attention to detail, just as luxurious and comfortable without sacrificing any of their present excellent qualities of road holding, and so forth.

In reply to Messrs. Sampietro and Bowden – These comments are extremely interesting and in many respects all too true. However, perhaps political aspects of the paper, introduced only as an explanation, should be disregarded.

The particulars of the Willys are interesting. There are several similar examples on the American market of a car approaching close to Mr. Appel's No. 3 classification.

As regards the Alpine passes, I have already mentioned their importance with respect to brakes. This has nothing to do with the number of automobiles registered in Switzerland, but rather with the fact that the Alpine passes are the crossroads of Western Europe, and most continental drivers encounter them sooner or later. In any case, they are only quoted as a type of road met with all over Europe, and elsewhere, in mountainous regions.

Referring to the comments on washboard roads, the writers are correct in stating that there are no typical gravel roads in Britain. That is why, in postwar years, washboard roads came to British engineers as an unpleasant surprise. There are quantities of such roads in Africa, Australia, and the Near East, in parts of the world which are not necessarily of low population, in fact some of them have large populations, but they are underdeveloped and full of development potential as regards road transport. That is why an industry which is expanding into the export field must take account of these conditions. This was true of conditions in the United States in the twenties when the cars came before the roads, and the demands of the cars brought the present roads into being.

With regard to engine overhaul I stated in my paper that the small size of European engines meant that the load factor was high, and therefore there was need for exceptional workmanship or more frequent overhaul.

The work required to push a vehicle through the air means that the light weight of European cars is not reflected in a proportionately less demand for power or lower fuel consumption. The power required varies largely with the frontal area and this does not decrease in proportion to the weight. Thus a large bus is much more economical in ton miles per gal than a motorcycle.

In reply to Mr. Kloth – I think Mr. Kloth in his enthusiasm for European cars, introduces an idea we need to look at carefully. Is it not possible to combine the style, the luxury, smoothness, docility, and general good manners of the American car, with the keen accuracy in handling and superb road-holding of the best European models, and all at a competitive price? I am sure it is.

Fast Replacement or Heavy Maintenance

Charles C. Hudson, Tennessee Valley Authority

This paper was presented at the SAE Annual Meeting, Detroit, Jan. 12, 1953.

THE argument of replacement versus maintenance is fundamentally – when to replace. In reviewing literature for this paper I went back first to an SAE paper by Dr. Joel Dean,[1] presented five years ago, which impressed me deeply. Dr. Dean called his plan for replacement the capital earnings method. It could have been called the economic justification method. Dr. Dean opened his paper with a statement that this most controversial subject had brought forth dozens of papers and articles with such widely different solutions that there appeared no basis for agreement. He also stated that current practice is probably even more varied. These two statements were facts and have stood the test of time! After five years the additional literature and current practice are still *varied.* A check with fellow fleet-operators, both in the United States and the United Kingdom, brought forth differing plans and data but, surprisingly, there is a uniformity of thought, if not practice, toward the idea which I shall attempt to present here – namely, fast replacement of mass produced standard vehicles, with maintenance reduced to the lowest point consistent with safety and reliability.

[1] See SAE *Quarterly* Transactions, Vol. 2, October, 1948, pp. 518-531: "How to Determine When a Motor Vehicle Should be Replaced," by Joel Dean.

FAST replacement – with maintenance reduced to the lowest point consistent with safety – is the theme presented by this author. On the basis of TVA fleet policies, he argues the case for fast replacement. He follows up with an analysis of the question: when is fast replacement fast?

He cites high job-shop overhead costs, which makes unit replacement of accessories, and early replacement of vehicles, economically inviting – except for large, special trucks. He proposes mass-production overhaul of components on large-volume, nationwide basis.

He emphasizes the need for balanced component designing, looking forward to a future when all components in one vehicle will wear out together.

The Author

CHARLES C. HUDSON (M '36) is assistant chief of the transportation branch of the Tennessee Valley Authority. He studied mechanical engineering at the University of Tennessee. He worked for the Mississippi River Commission and the Tennessee Highway Department, and in 1933 joined the transportation branch of TVA.

Table 1 – Parts Distribution System

Case for Fast Replacement

With job-shop labor costs running $3 to $4 per hr, and with little opportunity in a job shop for application of the more efficient mass production techniques, we are in an age calling for less maintenance and faster replacement. If you doubt the $3 to $4 per hr cost, I suggest that you recalculate your job-shop costs. Include mechanics' base pay plus vacation, sick leave, premium pay for overtime, after-hour call time, social security and/or pension plan costs, staff time for union meetings and negotiations, personnel work, various levels of supervision chargeable to the shop, and purchasing costs. Whether these are included in your present accounting plan is immaterial. They *are* shop costs. (See Appendix 1.)

Of course, mass-production factory costs are up too, but there the volume permits development of new techniques, and small savings here and there help to offset rising basic costs. Admittedly, much more can be done in job shops to increase efficiency, but the percentage favors the repetitive-operation, high-volume production plant.

It takes 1.2 hr (flat rate) to overhaul a small carburetor, but only 3 hr to build it. It takes 28.5 hr (flat rate) for a piston, ring, valve, bearing job on an engine, and only 110 hr to build the short engine assembly. It takes 1.3 hr to overhaul a distributor, but only 2 hr to build it. Such examples are endless. The prices that we pay for some accessories at times indicate more hours for production, but don't forget the handling costs and profits between factory production and the fleet operator. Some of these middle costs are necessary, but there are times when the distribution system that has grown up to serve the private owner of one car seems pretty cumbersome for fleets. (See Table 1.)

The fleet shop at the bottom of the list gets the generous-seeming discounts, and the purchasing agent feels a good deal has been made – and it is, so far as his immediate source is concerned. But the thought arises – what service do all those middlemen render to a fleet, especially when we order 100 items for a job and have a delay because 10 to 50 items are back-ordered up the list to the producer? The present system serves a purpose, but can't it be simplified and cost reduced at least to the larger fleets? Then we can scrap more assemblies and simply replace when trouble occurs on our preventive maintenance schedules.

Has Fast Replacement Been Tried?

Most rental companies in this country now replace passenger vehicles annually or at about 25,000 miles. Of course, appearance and new models have sales appeal, yet economic justification for fast replacement does not rest solely upon sales appeal.

Retirement of a sedan at approximately 20,000 to 25,000 miles means that maintenance consists principally of grease jobs and accident damage – no overhauls, no engine jobs, no valve jobs, no spark plugs, no clutch jobs, and no tires, in many cases. Maintenance cost of ½¢ per mile or less, excellent operating cost, too (for fuel and oil consumption is low for new vehicles), can be obtained with fast replacement.

A study of sedans sold early in our fleet (because of accident damage) and *without* a plan for early retirement shows a tire cost reduction of 60%, a fuel mileage gain of 3.9 mpg (from 19.7 for the group sold at 17,338 miles, up to 15.8 for a similar group sold at 67,000 miles), and a slight gain in maintenance. The group sold at 67,000 miles was in very high mileage service conducive to good costs. Comparison with sedan fleet averages shows a further gain in maintenance – 1.10¢ per mile compared to 1.24¢ per mile. Table 2 shows a cost estimate for light 4-door sedans on 20,000-mile (1.5 year) replacement, and on 60,000-mile (4.5 year) replacement. The figures improve where the 20,000 miles can be run in one year.

Note the 1.22¢ per mile savings shown for fast replacement!

Sometimes a fleet operation can lead to fitting up a complete shop to such an extent that eventually the fixed shop costs are so high that overhauls must be performed on and on to spread costs over a large volume. Or, a fleet makes a calculation showing overhauls are economical, and then lets time slide by and costs climb to a point that completely changes the picture. In 1932, with 35¢ per hr mechanics on a 50-hr week we could reclaim and salvage and rebuild – economically. Parts were higher than labor, but they *are not now.*

Government fleets (standard vehicles) are now replaced on a standards basis – sedans must be 6 years old or have 60,000 miles, and trucks (up to 2 tons) must be 6 years old or have 57,000 miles,

with a maximum of 25% of fleet that can be replaced in any one fiscal year. This, you will note, is a floor under replacements. This plan was set up by a committee of Government fleet supervisors – Interdepartmental Motor Equipment Committee -- sponsored at the time by the Bureau of Budget. Some of us who participated believe that a ceiling over which Government fleets *must* replace may be a worthy addition to the floor standards. Perhaps an average mileage and age standard would do the job best. The present standards-for-replacement plan is a big step in the right direction. We hope that it can be adjusted from time to time to fit current conditions, resale prices, purchase prices, maintenance costs, national economics, and so forth.

This idea serves one excellent purpose in planning retirements and replacements. It permits maintenance to be tailored to fit service life, and this is mandatory to lowest costs, whether it be fast or slow replacement. Unless the approximate time of replacement is known, unnecessary maintenance will be performed.

As proof that times do change and vehicle disposals vary, the TVA record is shown for representative types in Tables 3, 4, and 5.

You will note that fast replacement was indicated for trucks by the very high recovery values of 1948, 1949, and for some trucks in 1950. You can't fix a replacement plan this year and hold to it firmly for all years to come. Good fleet management reviews, studies cost data, cost trends, and revises replacement plan item by item as needed – all within the framework of the general policy.

When Is Fast Replacement Fast?

Questions and replies from eight varied fleets representing over 46,000 vehicles – 36,000 in the United Kingdom (British Transport and Victoria, Australia, Electricity Commissions) and over 10,000 vehicles in the United States (City of Los Angeles, Equitable Auto Co., Consolidated Freightways, Georgia Power Co., and Willett Co.) – indicate that fast replacement means replacement without an overhaul. A summary of this data is shown in Table 6.

Some fleets whose superintendents strongly favor fast replacement follow a relatively *slow* replacement schedule for reasons such as:

1. Failure to sell the story to all levels of management.
2. Long-period adherence to agreed-upon service lives – occasionally book values which have no reference to current overall costs (book value and depreciation being accounting tools for equitably spreading costs over period asset is used).
3. Unavailability of funds for purchase.
4. Effect upon current and future taxes, although with a constant upward trend the tax effect should favor fast replacement.

Our British friends mentioned purchase tax,

Table 2 – Sedan Cost Estimates

Operation	Sell at 20,000			Sell at 60,000		
Fuel	19.7 mpg	1.52¢		15.8 mpg	1.90¢	
Oil	800 mpg	0.04		450 mpg	0.07	
Miscellaneous services (antifreeze, wash, tire chains, and so forth)		0.60			0.60	
Tires (includes damage and repairs)	Replace ½ set	0.05		Replace 2½ sets	0.31	
Storage[a]		0.40			0.50	
Total			2.61¢			3.38¢
Maintenance						
Lube jobs	10 at $1.50	$15		30 at $1.50	$45	
Gear oil changes				5 at $2	10	
Initial service[b]		15			50	
Preventive maintenance service	1 at $45	45		At 10,000 miles	45	
				20,000	75	
				30,000	50	
				40,000	90	
				50,000	50	
Miscellaneous repairs (between each pm)		15			60	
Seat cover replacement					12	
Total		$90			$487	
Per mile			0.045¢			0.081¢
Depreciation						
Purchase cost		$1500			$1500	
Sales price		1250			500	
Actual		$250			$1000	
Per mile			1.25¢			1.86¢
Total cost			3.90¢			5.12¢

[a] Little outside storage for fast replacement.
[b] Seat covers, underbody spray, adjustments, marking, oil filters, and so forth.

Table 3 – Sedan Retirements – TVA

	Number of Units	Average Mileage	Average Age, months	Salvage Recovery	
				Cost, %	Replacements
1948	294	92,600	86	117	60
1949	208	86,458	90	98	52
1950	62	73,986	79	75	46
1951	97	56,745	37	61	54
1952	112	59,286	46	59	50

Table 4 – ½-Ton Truck Retirements – TVA

	Number of Units	Average Mileage	Average Age, months	Salvage (Recovery), %
1948	129	82,611	86	94
1949	195	68,161	86	73
1950	97	53,772	86	55
1951	38	45,793	46	43
1952	40	50,461	54	41

Table 5 – 2-Ton Truck Retirements – TVA

	Number of Units	Average Mileage	Average Age, months	Salvage (Recovery), %
1948	67	79,073	80	44
1949	68	67,229	82	39
1950	23	59,106	88	30
1951	—	—	—	—
1952	4	65,766	47	33

Table 6 – Data From United States and United Kingdom Fleets – Over 46,000 Vehicles

Type	Overhauls		Salvage	Average Age	
				Months	Miles
Sedans	0		40% (15–50)	52 (12–67)	53,000 (17,500–75,000)
Light trucks	0	(0–½)	23% (10–40)	55 (48–60)	49,000 (45,000–90,000)
Medium trucks	1	(0–2)	16% (10–30)	80 (48–120)	76,000 (50–126,000)
Heavy trucks	1¾	(½–2½)	10% (5–20)	108 (72–144)	137,000 (70–350,000)
Special heavy-duty trucks	2	(1½–2½)	10% (5–15)	133 (54–180)	162,000 (98–300,000)
Trailers	2	(1–3)	13% (5–15)	156 (96–180)	156,000 (100–300,000)

Fig. 1 – Diamond T truck modified by use of Hall-Scott engine, auxiliary transmission with power take-off, cab cooling, power steering, and other improvements

shortage of vehicles, and restraint on capital as reasons for not following a fast replacement plan, although they subscribe to it in these words:

"A new vehicle should normally be provided when the total operating costs per ton-mile payload of the old vehicle rise to a level just exceeding those for a new vehicle of similar carrying capacity. Total operating costs in this connection would include provision for depreciation and interest on the capital employed. It should be noted, however, that obsolescence in relation to current models may determine replacement before strictly necessary on economic grounds."

"As a general guide to United Kingdom practice, we would say that light mass-produced vehicles may be dealt with by substitution of new units or *units reconditioned by the makers* and available for substitution at moderate cost. Such vehicles would, however, be replaced when the stage is reached at which a general overhaul, over and above unit replacement, becomes unavoidable."

British operators would like to practice fast replacement but cannot, due (like TVA and others) to artificial restraints.

We may conclude that a replacement is *fast when it eliminates overhauls.*

What Do We Get In Overhauls?

The data from fleets questioned follows a trend that has been noted in discussions with fleet engineers. If the data in Table 6 is correct, fleet shop overhauls are amazingly good and cheap. (See Table 7.)

Table 7 – Fleet Opinions on Overhauls

	Overhauls		
Fleet	Restore Reliability, %	Cost in Per Cent of Replacement Cost	Life Restoration, %
1	100	10	—
2	90	16	—
3	93	50	—
4	75	25	—
5	75	20	—
6	60	25	50
7	90	40	90
Average	83 (75–100)	26 (10–50)	70 (30–50)

At the risk of provoking an argument from the people who cooperated so splendidly in furnishing this data *I question it.* Do we really get 83% restored reliability and 70% restored life for a cost of only 26% (of new vehicle cost)? If we do, why do we ever replace – appearance and technical advances aren't worth 74%.

The nearest to full-life restoration that we have reached in our fleet is in a group of war-surplus Diamond-T Model 981 tank-retriever 6 by 4 trucks. These were low-mileage trucks returned from overseas. We replaced the Hercules 895 cu in. diesel engines with Hall-Scott 1040-cu in. jobs, replaced Fuller 3-A-86 auxiliaries with Spicer 703 and 8031 top power take-off jobs, changed exhaust systems, aircooled the cabs, replaced the prime mover body, installed power-boost steering, and reworked other components as needed. Fig. 1 shows one of these trucks.

The total cost, including 45% overhead applied to labor, was $8,000 to $15,000, with an average of $13,000; and we got 90% service life and 90% reliability as compared to a comparable $16,500 new truck. This is a total cost of 80% of new cost, and the rehabilitation (and some modification) is 50% of new cost. The truck had many components which were practically new before this work began. At $15,000 no more such jobs will be approved.

Our 24,000 to 35,000-lb gvw trucks costing $8,000 sometimes have $3,000 to $4,000 overhaul costs – 40 to 50% of replacement cost with a life restoration of 75% and reliability of 75 to 80%.

It will pay to recheck total overhaul costs, actual life from such overhauls, and failures or troubles after overhauls.

Unit replacement by new or, as the British have, factory-rebuilt units, can help the overhaul and maintenance costs. Why don't we have more semi-mass-production-unit overhaul shops around the country equipped to handle truck and bus components – not just Ford, Chevrolet, and Plymouth high-volume components? Isn't the volume there for components of trucks above the 16,000-lb gvw class? The volume is there for *manufacturers* of units and parts – why not for regional rebuilders?

A quick study of transportation facilities and heavy-vehicle concentrations indicates excellent coverage by about 16 components rebuilders, and fair coverage by only four such shops, as shown in Fig. 2. It seems evident that the volume could be developed to support 4 to 20 component rebuilding shops with mass-production methods used to cut costs.

Fleets need, and as times get more competitive must have, help from heavy-vehicle manufacturers on either or all these:

1. Lower first cost – to encourage fast replacement.

2. More direct-parts distribution with lower prices – low enough to remove and throw away even semimajor components.

3. More component standardization – to encourage regional rebuilders.

4. Vehicle design with component life nearly equal – all components wearing out together.

Our United Kingdom correspondent expressed item 4 in this manner:

"As improved mass-production techniques tend to lower new vehicle costs relatively more than maintenance techniques lower overhaul costs, it would appear that we are heading for an era where vehicles will operate to the end of their useful lives with a minimum of maintenance and then be scrapped. Thus the ideal vehicle of the future might well be one which will operate without trouble until a stage is reached where so many parts are worn out that the purchase of a new vehicle will show clear economies as against the cost of repairs."

Some of us in fleet service have discussed this idea for several years. While we want long-life vehicles, we might even sacrifice a little life for a machine of balanced component design where *all components wear out together* or, to be practical, where many more than at present require replacement together. Component life varies with service conditions, but Messrs. Kunc, McArthur, and Moody[2] have reported that "engine life is one-half of car life." Every fleet has data showing that radiator life is less than engine, clutch less than engine – and they are hard to get at, too. Transmissions last about as long as engines, but we often have to open or remove them for one bearing or one synchronizer or one gear. These in-between failures are the expensive ones.

Why shouldn't a radiator be bolted to the engine as trouble free as the cylinder head? Why shouldn't fan and generators be driven with belts with lives equal to engine life? Why shouldn't we expect batteries to last as long as engine valves and without frequent service? Why should we have wiring failures 4 or 5 times in the life of a truck or bus? We know how to correct that now. Why shouldn't we expect distributor points and spark plugs to last to the first ring job – which should be an engine job anyway? Why should we have 40,000-mile valves in an 80,000-mile engine – in one case for want of $5 extra cost for valve rotators? Why 50,000-mile distributor points but 40,000-mile shaft bushings? Or 150,000-mile differential and 20,000-mile oil seals?

We need balanced component designing.

We have considered the aircraft progressive overhaul; but while it offers promise over the common preventive-maintenance plan for reducing time out of service, it is not the the whole answer.

[2] See SAE Transactions, Vol. 61, 1953, pp. 221-223; 230: "How Engines Wear," by J. F. Kunc, Jr., D. S. McArthur, and L. E. Moody.

[3] See *Commercial Car Journal*, Vol. 73, March, 1947, pp. 46-48: "Economies of Vehicle Replacements," by A. W. Bass, Jr.

Fig. 2 – Regional unit overhaul locations giving excellent coverage by 16 component rebuilders and fair coverage by four

We may conclude that we do not get all we should from overhauls – even for heavy vehicles where overhauls are economic. Fast replacement of components could give us more, but we need the help of the manufacturers.

Heavy Maintenance or Fast Replacement?

Like so many questions and like many engineering problems, there is no stock answer or cure-all for replacement of motor vehicles. Fleet conditions, current costs, cost trends, and other factors have a part in deciding between fast replacement or heavy maintenance.

Currently, few fleets can afford to pass up the savings inherent in annual or 20,000-mile replacement of sedans. Fleet discounts on new vehicle purchases will not eliminate the real depreciation for one year, but with good buying it should make real depreciation very low.

A. W. Bass, Jr., has offered a table[3] of vehicle market values which are probably low for current conditions and good fleet purchasing. Nevertheless, the data as quoted in Table 8, is interesting.

There are ways to improve fleet-vehicle sales

Table 8 – Market Values for Vehicles

(From Commercial Car Journal)

Years	Passenger	Light Truck	Medium Truck	Heavy Truck
1	64	60	58	55
2	52	50	48	45
3	41	40	38	35
4	32	30	28	25
5	24	21	19	17
6	17	14½	13	11
7	11	9½	8	7
8	8	6	5	4

Fig. 3 – Gradall truck which cannot be justified for quick retirement

Fig. 4 – A truck with aluminum body used for maintenance of electric transmission system – such a vehicle cannot be listed for automatic retirement

Fig. 5 – Earth drill on Marmon-Herrington truck, used for digging tower footing holes. Such a vehicle is not included in retirement listings

Fig. 6 – Bell helicopter used for spraying margins of reservoirs to control malaria and plant growth. This is a vehicle which is not retired according to a regular plan

Fig. 7 – Bell helicopter used in maintenance patrol of power lines is another vehicle that does not come under routine retirement plan

values. The Willett Co. set up a used-vehicle sales subsidiary. In TVA we have used newspaper ads, radio spots, personal sales effort, and the widest possible mailing list for bids (our sales are by competitive written bids submitted for action on a specific date). Some fleets use trade-in contract plans with dealers.

We can conclude that most fleets should retire sedans early – usually at 20,000 miles or within one year. Trucks present a specific problem for each class, for each fleet, and almost for each year. Fast replacement for trucks may be 5 years and 50,000 miles with no overhaul for ½ to 1½-ton classes, and up to 10 years and 500,000 miles for heavy trucks.

Table 9 shows a retirement plan that has been in use for over 5 years.

We get right back to the fact that fast replacement is economical for mass-produced vehicles – maybe it will prove economical for more and more truck classes. Also, that fast component replacement can offer fleet savings with a little encouragement from the industry. The fact remains, however, that we must justify each replacement on total cost basis – Dr. Dean's capital earnings method, or a plan that is proposed here as an economic justification method.

Figs. 3 through 7 show some types of special vehicles that cannot be justified for quick retirement.

Formula to Determine Economic Justification

All engineers like a formula to apply to problems. The one offered here is not, I'm sorry to say, a complex mathematical formula. More nearly a simple arithmetic thing, maybe too simple to call a formula.

This formula came into existence because vehicle replacement in TVA has been on an economic justification basis for many years. (The plan is covered in Appendix II of this paper.) Our people needed the plan resolved to simple arithmetic to facilitate application and lower-level supervisor determination – all within the retirement policy. In studying what and how we were handling each retirement it was found that this formula given in Table 10 seems to state the case.

An example of the formula is shown in Table 11.

The weaknesses of this plan are:

1. Good estimates are mandatory.

2. Appearance for advertising and driver pride is not included.

3. Past expenses are included – some authorities oppose use of original cost anld utilization to date.

4. Poor purchasing in the past can force holding a vehicle longer than with a lower purchase cost. (Is this bad? Shouldn't purchasing mistakes be emphasized to this extent?)

5. Extra capacity (or useful work) of new vehicles is not brought into the case.

6. Lost time of vehicles is not covered.

Table 9 – TVA Vehicle Retirement Plan

Type and Class	Miles		Hours		Years	
	Inspect	Retire	Inspect	Retire	Inspect	Retire
Carryalls, suburbans	36,000	50,000	—	—	—	—
Sedan delivery	36,000	60,000	—	—	—	—
½ and ¾-ton pickup, stakes	36,000	50,000	—	—	—	—
½ and ¾-ton panels, mounted body	36,000	60,000	—	—	—	—
1½ to 2-ton stakes	36,000	55,000	3500	5000	4	5
24,000 to 65,000-lb gvw stakes, winch, and so forth	54,000	75,000	3600	6000	6	10
1½ to 3 cu yd dumps	18,000	30,000	3600	5000	4	5
4 to 12 cu yd dumps	36,000	50,000	3600	5000	5	6

Some vehicle data are recorded by miles and others by operated hours – depending upon service.

The retirement is based upon conditions at time of inspection, but the figures are currently expected averages.

Table 10 – Suggested Formula for Replacement of Vehicles

$$\frac{a+b+f-c}{x} > \frac{a+e-d-w}{y} = \text{Sell}$$

When:

a = Original cost of old vehicle under consideration for disposal
b = Cost of repairs required to continue in service
c = Salvage value of old vehicle at end of extended service life (after repairs covered by b)
d = New vehicle salvage value at end of its normal life
e = New vehicle cost (current cost)
f = Difference in operation and maintenance cost of old and new vehicles for extended life (after repairs)
x = Total life of old vehicle with repairs (total life including life to date plus life extended by repairs)
y = Total life of old vehicle (if sold now) plus expected life of new vehicle (life to date of old plus normal life of new)
w = Current sales price or value of old vehicle

Note:

1. Life should be expressed in miles or hours although age can be used.
2. In some cases there may be no difference in operation and maintenance cost between the old vehicle after repair and the new vehicle, although it is normal to expect some difference. Repair or overhaul rarely restores complete efficiency and the newer or current vehicles usually have some improvements, representing a gain in cost of operation anp maintenance.

Table 11 - Example of Application of Retirement Formula

$$\frac{a+b+f-c}{x} > \frac{a+e-d-w}{y} = \text{Sell}$$

a = $2289 - Cost of vehicle being considered for disposal
b = $900 - Cost of repairs to extend life 46,000 miles
c = $250 - Estimate salvage value at 80,000 miles (current 34,000 plus extended 46,000)
d = $200
e = $1934
f = $230 – Extra cost for operation and maintenance of ½¢ a mile for 46,000 miles
x = 80,000 miles (current 34,000 plus 46,000 additional estimated with repairs)
y = 114,000 miles
w = $400

$0.0396 greater than $0.0318 = Sell

Advantages of the plan are:

1. Ease of application by even lower-level supervisors.

2. Forces individual consideration into an overall policy based upon averages useful for planning operations.

3. Includes real depreciation, current sales prices, replacement vehicle cost, and service life restoration as major factors.

4. Gets very close to the true cost – the great American measurement undestood by all – the dollar sign.

The plan as applied in the TVA fleet produced the depreciation rates shown in Appendix III of this paper.

Conclusion

From the foregoing it is hoped you will agree that:

1. Fast replacement (before an overhaul) is economical for fleet vehicles of the high volume, mass produced, light types.

2. Fleet sedans can often be replaced at one year or 20,000 miles (approximately) with substantial overall savings. Also, it releases vehicles to private users who can't buy new ones, with no real competition to dealers and no adverse effect on short-supply materials.

3. Fast replacement of components, even in heavier trucks, offers cost reduction promises. Also, it reduces total manpower required by the industry and lessens highly skilled craftsman requirements by using assembly line techniques.

4. An overall plan based upon averages from good records and good estimates of future costs will permit *tailored maintenance* and lowest costs.

5. Economic justification can be resolved to a simple formula.

6. Fleets need industry help in lower replacement component costs, lower purchase prices, and balanced design to reduce in-between maintenance and encourage production techniques in job-shop work. Shorter distribution to fleet supply lines may be the answer.

7. Fast replacements, both vehicles and components, reduce fleet investment in shop equipment and space.

8. Fast replacement may be 20,000 miles in some cases and 500,000 miles in others.

9. Economic justification is the soundest basis for vehicle replacement.

Additional Data

For those who wish to review the subject we suggest the following, in addition to the footnotes given in the paper:

1. "Proposal to Motor Vehicle Retirement Petroleum Distributor," by Gavin W. Laurie. Presented at SAE Annual Meeting, Detroit, Jan. 8. 1946. Digested in *SAE Journal,* Vol. 54, August, 1946, p. 18.

2. "Approaches to Vehicle Retirement," by D. K. Wilson. Presented at SAE Annual Meeting, Detroit, Jan. 8, 1946. Digested in *SAE Journal,* Vol. 54, August, 1946, p. 18.

3. "The 'For Hire' Carrier Looks at Depreciation," by Howard W. Willett, Jr. Presented at SAE Annual Meeting, Detroit, Jan. 8, 1946. Digested in *SAE Journal,* Vol. 54, April, 1946, pp. 50-51.

4. "Graph System Helps Locate Best Trade-In Point," by S. G. Page, in *Commercial Car Journal,* Vol. 73, July, 1947, pp. 64-65; 122.

5. "Transmission Bearings for Heavy-Duty Service," by E. J. Barth and Thomas Backus. Presented at SAE National West Coast Meeting, San Francisco, Aug. 12, 1952. Contains statement on fallacy of overdoing reclaiming in job shops; cites transmission case item on pages 1 and 2. Digested in *SAE Journal,* Vol. 60, October, 1952, pp. 107-08.

6. "Statistical Studies of Vehicular Retirement." Compiled and published by the University of Iowa Experiment Station in 1942.

7. The law of natural growth expressed by

$$y = ce^{kt}$$

where:

c = Constant
e = Natural logarithmic base
k = Proportionality constant
t = Time

8. Your own cost records.

APPENDIX I

Easily Overlooked Costs

News Item of September 7, 1952

"Fringe Benefits" 16% of Payrolls
Average $653 Annually Per
Worker in 109 Companies
Surveyed by Chamber

WASHINGTON, Sept. 7 (AP)—Workmen in 109 American companies draw an average $653 a year in "fringe benefits" – pensions, sick benefits and the like – which don't show up in their pay envelope as so many dollars.

This is the finding of the United States Chamber of Commerce, which has been surveying "fringe benefits" for more than three years. The Government's Bureau of Labor Statistics (BLS) says the C of C survey is "the most complete study that has been done in this country."

In 1949 C of C researchers sent questionnaires to 690 employers, large and medium-sized companies in all the major industries. They found that "hidden wages" accounted for an average 16% of payrolls of these 690 employers and that the average workman enjoyed welfare benefits that cost an average 23.7¢ an hour, $477 a year.

In 1951 the C of C went back over 109 of the companies. It found fringe benefits had increased to 16.2% of the 109 payrolls, the cost per hour to 31.1¢ per worker and the cost per year to $653.

APPENDIX II

TVA Vehicle Retirement Plan

Purpose – This plan is intended to provide an orderly retirement of vehicles on an economic basis, enable replacement planning, and permit the tailoring of maintenance to retirement. The plan follows these steps:

1. Predetermined service life averages – a staff function.

2. Inspection prior to end of average service life - a field function.

3. Determine remaining service life – a staff function based upon field inspection, cost data, and historical records.

4. Notification of retirement – a staff function utilizing memorandum to assignee and copy to garages.

5. Avoid repairs – a field function involving shop and driver personnel.

6. Plan replacement near retirement locations – a staff function.

7. Replace and sell – a staff and field function.

Normal Retirements – Vehicles are retired under normal conditions when "no longer suitable for efficient service." "Efficient service" includes factors such as reliability, cost of repairs, cost of operations and expected cost of replacement vehicles. Availability of replacements is of primary importance under abnormal conditions (cost, status of funds, acts of God, strikes, unusual market conditions, legislative restrictions, and so forth).

Light vehicles with standard bodies and equipment are normally retired when a complete overhaul is imminent. Such vehicles can have reliability and economic operation partially restored only by greatly increased maintenance cost (overhauls).

Retirement and sale when expensive repairs are imminent follows the policy generally used by private owners. Estimated retirement ages from a study of records and vehicle specifications results in predetermined inspection mileages, a final careful estimate of remaining mileage, and the fitting of maintenance to remaining life.

Inspection and Retirement Mileages – Until further notice inspection and retirement *average* mileages will be used as shown in Table 12.

Abnormal Retirements – When vehicles are damaged in an accident, badly worn through unusual service conditions, or in a very few cases prove to be "lemons," earlier inspections, reports, and possible retirements are in order. Foremen should feel free to offer Form 3070 reports and estimate of remaining service life on any vehicle at any time.

Estimate of Remaining Life – By study of historical data to insure against retirement just after expensive rehabilitation repairs, examination of cost data in order to arrive at a point just prior to expensive repair requirements, and use of the field inspection report which covers mechanical condition as well as estimated repair cost and estimated service life remaining, the central staff can fix economical retirement age (mileage or months). Retirement age for individual vehicles will, of course, be varied in order to meet replacement plans and other economic requirements such as: sales values, now and future; replacement vehicle purchase cost, now and future; and current level of repair cost.

Notification to Field – After the retirement age has been established on an individual vehicle basis, a memorandum will be written by the central staff to the assignee with copies to all garages having occasion to service the vehicle. This memorandum will request that repairs be kept at a minimum consistent with safe operation until the estimated remaining life has been extinguished.

Table 12 – Inspection and Retirement Schedule

Type and Class	Miles		Hours		Years	
	Inspect	Retire	Inspect	Retire	Inspect	Retire
Sedans (up to 1000 tag series)	54,000	70,000	—	—	—	—
Carryalls, personnel carriers, and so forth (in both 10,000 and 30,000 series)	36,000	50,000	—	—	—	—
Sedan deliveries (in 20,000 series)	36,000	60,000	—	—	—	—
½ and ¾-ton pickups, stakes, and so forth (in both 20,000 and 30,000 series)	36,000	50,000	—	—	—	—
½ and ¾-ton panels, SLM,[a] and so forth (in 20,000 and 30,000 series)	36,000	60,000	—	—	—	—
1 to 2-ton stakes, special, LM[b] (in 40,000, 41,000, 50,000, 51,000, and 56,000 series)	36,000	55,000	3600	5000	4	5
5-ton and up, cargo, special, and so forth (in 48,000, 49,000, 57,000, 58,000, and 59,000 series)	54,000	75,000	3600	6000	8	10
1½ to 3-yd dumps, 1½ to 4 ton (in 52,000 and 53,000 series)	18,000	30,000	3600	5000	4	5
4 to 12 yd dumps, 5 to 12 ton (in 53,500 and 54,000 series)	36,000	50,000	3600	5000	5	6

[a] Street-light maintenance.
[b] Line maintenance.

Replacement Plan – Replacement vehicles will be planned and scheduled for delivery at a point near the old vehicle assignment area. This keeps ferry mileage at a minimum.

Light vehicles (2-ton trucks and down) are normally obtainable upon request under an annual vehicle contract. Until automotive production returns to normal it may be necessary to schedule replacements at approximately the same period each year (within 30 to 60-day limit). Return to normal, however, will permit scheduling of replacements around the calendar.

Sale of Old Vehicles – Retired vehicles will be grouped at sales points convenient to the point of replacement. Consideration, however, will be given to the current level of used vehicle prices. The cost of ferries will be compared to the expected difference in prices between sales points. The final decision will be based upon all these economic factors and should result in the best overall net price.

Heavy Vehicles and Machines – This plan covers primarily light, medium, and heavy-duty vehicles. Heavy vehicles are handled on a more individual basis, but the same economic considerations are used in determining retirements.

At present the small number of construction machines in the fleet eliminates the need for published predetermined inspection ages, although the central staff actually uses predetermined retirement ages (estimated service life from depreciation rate studies).

Procedure – While it is not the main purpose of this outline to cover all points of procedure, it is perhaps well to mention a few. Flags will be used

on assigned cards to indicate remaining service life, imminent retirements, and vehicles already retired but in process of sale.

The sale of used vehicles will, of course, follow the accepted policies outlined by the sales section of the materials branch. All shops handling vehicle sales should keep in mind that "accessories on vehicles at the time of inspection must not be removed without sales section permission."

Form 3210 will be used to accumulate a condensed historical record and operating data such as mph, gph, actual mileage accumulation, repair cost per mile, and so forth. This and other record sheets used are standard TVA forms.

Form 3070 will be used by the field in making inspection reports. Reasonable estimates of the cost of repairs required to continue the vehicle in service for the estimated remaining life as indicated, should be made by the inspecting shop personnel. The estimated remaining life should be inserted on Form 3070 for any vehicles under retirement consideration.

Actual retirement should be effected on schedule. It is the responsibility of assignee, field personnel, and staff to retire as scheduled. The staff function will be to review historical records periodically and call attention to any vehicles at or very near the retirement age. The field function will be to watch for vehicles at or near retirement and notify the assignment clerk.

Conclusion – It is believed that this retirement plan follows sound economic principles. The plan utilizes historical data, estimated service life data, a study of price conditions, the restriction of maintenance expenditures near the end of vehicle life, and, in general, should result in the lowest net overall cost to TVA.

Cooperation of all groups is essential to successful operation of the plan.

APPENDIX III

TVA Standard Rates – July 1, 1952

License No.	Vehicle	Depreciation Rates Per Month	+ Per Mile
100	Passenger cars		$0.01
10000	Reconnaissance vehicles		0.018
20000	½-ton trucks		0.016
30000	¾-ton trucks		0.017
36000	Light 4 x 4 trucks (jeeps to 1-ton)		0.037
40000	Light cargo trucks		0.022
41000	Light special body trucks		0.04
49000	Heavy cargo trucks		0.083
			Per Hour
50000	Light LM trucks	$36	$0.09
50600	Special power take-off trucks	22	0.57
51000	Medium LM trucks	32	0.16
51500	Light tractor trucks	17	0.28
51600	Medium tractor trucks	21	0.10
51700	Heavy tractor trucks	31	0.18
52000	Light dump trucks	11	0.30
53000	Medium-heavy dump trucks	20	0.40
53500	Heavy dump trucks	70	0.80
57000	Medium LC trucks	20	0.55
59000	Special heavy winch trucks	49	0.46
61000	Medium LC crawler tractors	16	0.35
62000	Medium-heavy LC crawler tractors	19	0.35
62200	Heavy crawler tractors	49	0.38
62400	Extra-heavy crawler tractors	39	0.52
63000	Medium-wheeled scrapers	13	0.52
63100	Large-wheeled scrapers	30	0.53
64000	Small farm and industrial tractors	9	0.14
64200	Medium farm and industrial tractors	13	0.18
65100	Small air compressors and portable welders	4	0.18
65200	Medium air compressors	8	0.12
66300	Extra small mixers	2	0.13
66400	Small mixers	7	0.10
67000	Small road rollers	5	0.07
67100	Extra small cranes and shovels	21	0.19
67400	Medium small shovels	65	0.32
67500	Small power shovels	125	0.25
70100	Light dinkey trailers	2	0.02
70200	Medium dinkey trailers	1	0.02
70300	Medium pole trailers	3	0.04
71300	Light special-purpose trailers	1	0.02
71400	Medium special-purpose trailers	1.50	0.13
71500	Heavy special-purpose trailers	24	...
71800	Van special-purpose trailers	7	0.11
72500	Light platform trailers	5	0.13
72800	Medium platform trailers	14	0.32
72900	Heavy platform trailers	16	0.32
72950	Extra-heavy platform trailers	24	0.16
89000	Small fire engines	7.50	...
89100	Medium fire engines	30	...

DISCUSSION

Evaluates Major Factors in Vehicle Retirement Plan

– T. L. Preble
Tide Water Associated Oil Co.

THIS very useful paper is indeed thought-provoking. It should induce the fleet operator and his boss to re-examine their overall motor transportation cost data, and their procedure formula, if any, for vehicle retirement. It should prompt them to re-evaluate each element of such procedure, and try again to eliminate as much of the empirical as possible.

The author mentioned Dr. Dean's excellent paper presented five years ago. Dr. Dean consulted many people before producing this paper. I was one of them. In the course of this consultation I think I told him substantially this:

"Vehicle retirement deserves intensive study. It is a difficult problem, loaded with variables, which I doubt will ever be solved completely either by educated guess or by mathematics. The operator must use his soundest judgment in retirement studies (as in vehicle transfers) with overall economy in mind. That somewhat vague term – overall economy – refers specifically here to: (a) operating expense, (b) investment account, and (c) to a constant realization that both (a) and (b) are cash out of the corporate till."

Mr. Hudson necessarily deals with some painful variables in his final equation, particularly (c) salvage value of the old vehicle, and (d) new vehicle salvage value at the end of its normal life.

Values of the moment may be paramount in the rendering of a final decision.

I think this paper contains "take home pay" for many of us who are interested in transportation and maintenance activities, and should be of more than casual interest to our vehicle-design colleagues. But I think the latter may, with this discussor, raise a skeptical eyebrow over the author's expressed hope for vehicle designs with equal component life, so that all parts will wear out together.

Vehicle Evaluation Based On Labor-Saving Devices

– D. K. Wilson
Niagara-Mohawk Power Corp.

THE author has done a good job in analyzing a transportation situation. I find very little to quarrel with in his thoroughgoing presentation of the replacement versus maintenance problem.

However, I would like to point out one phase of the overall picture, which may militate against the adoption of a simplified formula, especially at this time. It probably applies particularly to public utility vehicles, where trucks are used principally as a means of moving auxiliary apparatus – winches, earth borers, compressors, aerial equipment, and so forth – from job to job. Many new labor-saving devices have recently appeared on the market that are adaptable to the various needs of utility operation. Savings in vehicle replacement accrue not alone in original investment in the vehicle, or its subsequent maintenance expense, but in the cost of the work the vehicle is to do, as well. In fact, many of the greatest savings effected by a competent automotive fleet engineer may not be truly reflected in actual transportation costs at all, but largely in the cost of the job on which the vehicle is working. Therefore, in competing with other segments of a corporation's operation for capital funds, it often obtains that the purchase of one piece of specialized equipment represents a more profitable overall internal investment than the equivalent dollar expenditure for several small cars or trucks. In such cases, which are quite frequent today, the author's definition of "extra capacity (or useful work) of new vehicles," a basic measure of capital productivity which it is difficult to incorporate into any formula, may easily become the deciding factor. Replacement of smaller vehicles, on a like-for-like basis, being lower in the scale of profitability from the standpoint of return on investment, cannot be justified in comparison with the purchase of a proved labor-saving type of special equipment.

We fully agree that, divorced from comparisons with other types of equipment, fast replacement of mass-produced light vehicles is sound. In a mixed fleet, ranging from mass-produced light vehicles to the most highly specialized types of vehicles and construction machinery, the problem becomes more complex. Retirement formulas, applied without regard for other than light vehicles, tend to limit one's thinking to a like-for-like replacement basis, often overlooking possible applications of new developments, new methods, and new work techniques.

Early Replacement of Light Vehicles Approved

– W. M. Kissam
Tide Water Associated Oil Co.

THE question of when to retire a vehicle and how to reach the right decision on this subject has been a matter of controversy as long as we have had cars and trucks, and will probably continue to be controversial as changes occur in the factors that affect the equation, probably the most variable at the moment being taxes and used car values. The nongovernment operator who trades often must consider the 8% Federal excise tax, and in New York City and the states of Connecticut and Rhode Island to my knowledge, there is also a 3% sales tax. An item of 8 to 11% could swing the decision the other way when the balance is close. However, I feel that the author has indeed made a strong case for frequent replacement of stock passenger cars, for his particular situation. His conclusions follow the line of most fleet-operating policies, that is, the lighter and cheaper the vehicle, the sooner it is replaced.

Mr. Hudson is to be commended for having put the economic factors into a highly simplified formula to expedite the work of the field supervisors as well as the home office staff, even though the author himself does not claim it to be perfect in every detail. On this latter point I agree with him, since it depends so much on estimates and on estimated projections that are little more than guesses.

I am inclined to think that the vehicles the author selected to illustrate full-life restoration were so heavy and so very special and were so completely changed and converted that they are not really typical of the type of overhaul and restoration job done in commercial highway truck fleets.

Frankly, I fear that, as long as fleet buyers have minds of their own on selection of components, and sales competition among truck builders is what it is, component standardization is still a dream.

Experiences in Fleet Operations Compared

– E. B. Ogden
Consolidated Freightways, Inc.

THE author has made an admirable presentation of a very difficult subject, and I might add, a subject that cannot be dealt with by the industry as a whole, since there are too many varied circumstances to allow any degree of complete accord or adaptation.

Also, some utopian thinking has been proposed in several places in the paper, but I fear that, unfortunately, it will not lead to practical results in the foreseeable future.

In the introduction the author states that while the actual practice of replacement versus maintenance varies widely, in his opinion there is uniformity of thought on the subject. I question even the uniformity of thought, since uniformity of thought would surely result in more uniformity in practice.

When I agreed to discuss this paper, I did not fully realize the trend of the subject. Otherwise I probably would have been reluctant to participate, since the bulk of our fleet is composed of over-the-road heavy-duty trucks and trailers, costing in the $25,000 class. These units are not conducive to fast replacement as referred to in this paper, since the resale or trade-in of this type of equipment is considerably more difficult than equipment in the $5,000-or-under class.

Insofar as sedans or passenger cars are concerned, they present no problem to us, since the percentage of these as compared to the whole is very slight. In passing, however, I can truthfully state that our policy of replacement at 60,000 miles, which is between two and three years, has proved very satisfactory, and is before the first overhaul.

I agree with the author in his calculation of job costs, and I was interested in the item on fringe benefits in Appendix I. Many of you might well take a good look at that $653 cost per employee which you must pay just because of the fact that he is on the payroll.

In some cases, due to other restrictions imposed by the work week, days off, and so forth, it might be better to pay overtime rather than carry so many employees on the payroll. To cite an example, in one of our work units, 480 man-hours were required per week on a 6-day basis, and the work could not be cut to 5 days. Also, there was the same amount of work to do each day. To staff this unit on a 40-hr per week basis required 12 men. To accomplish the same result on a 48-hr basis required only 10 men. Using $2 per hr straight-time pay, it figures like this: 40 hr $\times$ $2 = $80 + $12.56 per week fringe = $92.56 $\times$ 12 men = $1110.72 per week total. Again, using $2 straight-time pay, plus time-and-one-half for 8 hr, it figures like this: 52 hr $\times$ $2 = $104 + $12.56 fringe = $116.56 $\times$ 10 men = $1165.60. Thus, the increased cost (on paper only) is $54.88 for the entire unit.

Before you condemn this as unusable, due to increased cost, remember I said there was the same amount of work to do each day, and we had to cover a 6-day period. We have never found a way to stagger shifts to accomplish the desired result of having no overtime on the days the crew is light, without having idle time during the balance of the week when all men are on duty; so we feel we are well justified in reducing the crew by 2 men and permitting all men to work 8 hr at overtime rates. We probably will go further with this if it continues to prove desirable.

In the paper a reference is made to parts which are cheaper than labor. This may be true of parts for vehicles in the lower price class but it certainly is not true of many parts in the vehicles we are using, nor, I believe, is it true for most of the other vehicles designed for over-the-road operations. For example, a new worm shaft costs $148; the cost of reconditioning it runs from $10 to $15. A new brake cam costs $11.10; it can be reconditioned for $2.50 to $3. There are many more examples like these.

I agree, however, that any fleet superintendent must constantly review the practice used in the shop in regard to what is economical to salvage, and he must check this by reviewing actual average cost and not rely upon just what the shop man tells him he thinks it costs, because many shop superintendents are prone to salvage items which are far beyond the point of economy.

The author seems to question that we receive 83% restored reliability and 70% restored life for 26% of new cost. I don't think this is out of reason at all, and I feel it is probably reasonably true. I don't consider the Diamond T model 981, cited as an example, an overhaul at all, as it could more nearly be called a redesign, using the frame and wheels as an excuse to start. If we were to practice this type of rebuilding and redesigning, I am sure we would not come out with the 26% figure.

Our practice is to keep the vehicle as near to factory standards as possible for the entire time it remains in the fleet, be that 6 years or 12 years. By so doing, heavy expenditures such as outlined above on the Diamond T, do not occur.

If such practices were carried out in general, there would never need to be any replacements of equipment, since all advancements in design would be incorporated at the time the overhaul, or rebuild, was accomplished. Referring to the example of the 24,000 to 35,000-lb gvw trucks, to come out with a $4000 overhaul on an $8000 truck, it would seem that some redesign must have been worked in.

Some years ago we followed, to a large degree, the practice of trying to incorporate the latest developments into each old vehicle at the time of overhaul, but in so doing, we found that our costs resembled those on the Diamond T.

We feel the practice is unsound, so therefore there comes a time when replacement is necessary and desirable from the maintenance viewpoint, leaving all other factors such as taxes, book values, and so forth, out of the picture entirely.

There is probably considerable merit in the suggestion that large rebuilders of units for trucks have a place in the maintenance field, especially for operators with fleets too small to properly equip a shop of their own. However, for fleets of large size, I question that any mass rebuilder could compete in price for overhauled units with a well-equipped and well-managed company-operated shop, since a rebuilder has to show a profit on all labor, and company shops usually merely figure actual labor costs plus overhead. I certainly do not agree that the quality of workmanship should be any better.

Another extra cost that surely would enter into a large rebuilder's total price is the use of more new parts in a rebuilt unit, merely to safeguard against a possible failure.

I think most fleet superintendents will agree that 100% replacements of certain parts, merely to avoid an occasional failure, or the replacement of certain parts merely to improve the quietness of a unit, is a practice that large fleet operations cannot afford.

I agree that lower first cost is most desirable. As for lower prices and direct distribution of replacement parts, so far as I am concerned, the replacement part pricing system is downright shameful; there are at least two profits in every part that the fleet shop uses that, in my opinion, have no justification for existing.

When it is possible for large fleets to secure 60% discounts after going through all the channels outlined in Table 1, imagine what savings could be made in fleet maintenance if even two of these profits didn't exist. I am sure most of you know that many parts are actually shipped direct from parts manufacturing companies to the fleet shop, and each one on the list referred to on Table 1 takes a slice, although actually they never even see the part, so

there can't be much cost attached to merely making out an invoice.

Again, most of us know that many original units are sold at a very low margin of profit so as to get them in service, in order to enjoy the parts business. This whole practice needs revision.

More component standardization is most desirable from the standpoint of fleet shop maintenance, whether or not it encourages regional rebuilders. However, I do not hope for a far-reaching improvement in this channel since the sales of vehicles are largely dependent on so-called outstanding improvements in new models, which usually results in changes from a former so-called standard.

In principle I agree, however, that many standards can be set which will not restrict advancements or confine the use to any one unit. (For example, a flywheel for clutch mounting.) There is no reason why all flywheels for a given size of clutch should not have dimensions planned so that any one of several makes of clutches could be interchanged; many more examples could be worked out.

The utopian condition of having all components wear out together is the end we have been working toward for years. However, I don't think we have ever suggested that any part should be purposely made with less life, in order to balance it with some other part of the unit. Rather, we have been harassing the manufacturer to increase the life of the offending part. I fear if we use any other approach we might, in the end, have an inferior product with about the same price tag on it. No, let us not let the bars down to admit an inferior product, but rather let us find better materials and better designs to improve the short-life unit. In some cases, all that retards this at present is the cost of materials, and when we are buying on a competitive market, as most of us are, we may well look at our own policy to see if we are willing to pay that little extra cost for those valve rotators referred to in the paper.

In conclusion, I want to remark that the author has touched on many subjects that should not be taken lightly by anyone from top management on down to the shop foreman.

ORAL DISCUSSION

Reported by Henry Jennings
"Fleet Owner"

W. A. Taussig, Burlington Truck Lines: Lack of experience with arithmetical formulas and graphs disqualifies me from saying whether or not they are practical. My impression, however, is that whether you overhaul a vehicle or buy a new one depends upon a lot of circumstances peculiar to the time, including the cash position of the company.

I also have some reservations about the extremely high labor rates, or rather, about the amount that represents labor and the amount that represents overhead. Overhead costs are high in dealer shops. Whether we repair in the fleet shop or in the dealer shop depends largely on overhead costs. For example, we recently asked an outside shop to do a specialty job on an accident-damaged truck – a job that our shop was not equipped to do. Through a misunderstanding the outside shop completely rebuilt the truck. In the negotiations which followed relative to who was going to pay for the work done but not ordered, we offered to pay the actual cost of doing the work. The cost amounted to 50% of the original bill. Now this seems like a great deal of overhead, and I wonder what happens to it if overhauls are omitted.

Speaking of the reliability of old trucks, I have checked up on my records and find that the most unreliable trucks in the fleet are those which have had less than a month in service. Previous experience indicates that trucks are most unreliable during their first year in service. It often takes a year to get the bugs out of a new design. For that reason, vehicles may be more reliable after overhaul than before. Maybe that is why the statistics presented show the post-overhaul reliabilities they do.

Whether to retire or rebuild depends partly on company conditions. We had a case a year or so ago, for example, although it was a case of buying used vehicles and overhauling them or buying new vehicles. We wanted to experiment with running our own pickup and delivery service, instead of hiring contract draymen. The operation was a must, but the money to buy a new fleet was not available. Hearing of a group of used trucks that were available, we bought them for $25,000, fixed them up, and put them into operation. These trucks were of an age that would indicate replacement by any existing formulas. The operation is now working at a profit with these vintage trucks, and it would have cost us $500,000 for a fleet of new vehicles to do the job.

W. A. McCutcheon, U. S. Bureau of the Budget: Studies show that the Federal Government should operate fewer vehicles more miles, and should gear maintenance to early replacement. Thinking and practice are now tending along these lines.

A. W. Neumann, Willett Co.; We believe in relatively early replacement. It is important to keep replacement schedules up to date.

E. P. Gohn, Atlantic Refining Co.: Mr. Hudson's paper ought to wake us up to some economic facts. We don't get the same changes in fuel and tire costs with mileage as he shows in Table 2, but we do get about the same total costs. If we run a vehicle 20,000 miles, we get the cost per mile down just about as low as if we run it 60,000 miles. So why not sell at 20,000 miles?

Author's Closure To Discussion

REVIEW of the discussions indicates some misunderstanding of "when is fast replacement fast?" I have tried to point out that fast replacement may vary from 20,000 miles for light sedans to 500,000 for heavy trucks.

The term "overhaul" has a widely varying understanding and has, perhaps, confused some of us. Overhaul for my purpose is "general tear-down, rehabilitation, and reassembly, over and above unit replacement or unit overhaul." This is the overhaul for fleets to eliminate – in most cases – certainly for mass-produced, low-cost, high-resale vehicles!

The formula offered lends itself to consideration of purchase taxes, added productivity, and varying resale values (economic trends or local sales prices) for light and heavy vehicles. The weakness of the formula lies in the accuracy and honesty of estimates, but don't we make these estimates sometime anyway? Why not make them at purchase and fix our general replacement plan for all to know and work toward (tailored maintenance) – with each replacement rechecked by the formula when estimated life is extinguished or before, if conditions have undergone radical changes? As Ted Preble said, "Values of the moment may be paramount in the final decision." I add that they should be! The formula merely attempts to bring together all the factors of the replacement problem except these noneconomic factors such as personal preference, more or less profits this year, desire to have a particular make or model, drivers' pressure, and so forth – the emotional factors which should be eliminated in an economic engineering operation.

Measuring Rate of Fuel Injection

DIESEL-ENGINE combustion studies in the past have been handicapped for lack of information on some basic factors regarding fuel-injector performance. These primary unknown factors are the rate at which the fuel is introduced into the engine cylinder and the quantity of fuel available for combustion at any instant.

Satisfactory methods have been developed by bench tests to determine rate of fuel injection, such as the circulating collector pack method. However, it was the aim of this investigation to measure fuel-injector performance in a firing engine simultaneously with obtaining other pertinent engine data such as the cycle pressure-time curve. This paper describes the method and instrumentation developed to give three characteristics of injector performance: (1) rate of fuel injection, (2) fuel injection timing, and (3) injection pressures.

A cross-sectional view of the unit injector used in this development is shown in Fig. 1. The pri-

UNKNOWN factors regarding fuel-injector performance have handicapped diesel-engine combustion studies in the past. Rate at which fuel is introduced into the engine cylinder and quantity of fuel available for combustion at any instant were two basic unknowns hindering advance.

This paper presents results of an investigation aimed at measuring fuel-injector performance in a firing engine, simultaneously with obtaining pressure-time curves and other needed data. Method and instrumentation developed covers three characteristics of injector performance:

1. **Rate of fuel injection.**
2. **Fuel-injection timing.**
3. **Injection pressures.**

The method results in obtaining more comprehensive data, but has the disadvantage of added complexities requiring skilled handling.

The Authors

R. J. WEHRMAN (J '52) is a research engineer in mechanical engineering department No. 7 of the General Motors Research Laboratories Division. He joined the organization in 1950, after obtaining his B.S. and M.S. degrees in mechanical engineering from Ohio State University.

H. R. MITCHELL (M '48) is a senior engineer in mechanical engineering department No. 7 of the General Motors Reserach Laboratories Division. He received his B.S. degree in mechanical engineering from Purdue University.

W. A. TURUNEN (M '47) is the department head of mechanical engineering department No. 7 of the General Motors Research Laboratories Division. He received his B.S. in mechanical engineering from the Michigan College of Mining and Technology, and his M.S. degree from Columbia University.

in an Operating Diesel Engine

R. J. Wehrman, H. R. Mitchell, and W. A. Turunen,
Research Laboratories Division, GMC

This paper was presented at the SAE Annual Meeting, Detroit, Jan. 15, 1953.

mary parts are the body, plunger spring, bushing, rack, plunger, check valve and spray tip. In operation the fuel is introduced in the inlet where it follows the passages down around the plunger bushing. Fuel enters the cylinder ahead of the plunger through the two inlet ports. When the plunger is depressed by a cam mechanism, the plunger passes by the two inlet ports, sealing the chamber. The downward movement of the plunger forces the fuel past the upper check valve, past the spring-loaded check valve, and into the spray tip and out the metering holes.

Basically, our method of measuring rate of fuel injection treated the spray-tip holes as a flow-metering element. The injection pressure versus time curve was measured by means of a strain gage for a representative cycle, from which the fuel-injection rates were obtained using a suitable pressure versus fuel flow-rate calibration.

Measurement of Injection Pressure – The pressure-sensing device used for determination of injection pressures was an electrical resistance-type strain gage mounted on the injector spray tip. A change in pressure inside the spray tip produces a corresponding change in the strain of the spray-tip walls and thereby causes a resulting change in the strain-gage resistance. A modified d-c bridge circuit was used to convert this resistance change into a corresponding voltage variation. This voltage signal was then amplified and fed into a cathode-ray oscilloscope. The pressure-time trace produced was recorded photographically.

To provide a means of calibrating the pressure-time trace on the screen of the oscilloscope, a chopped square wave was superimposed on the trace by making and breaking a shunt across one of the ballast resistors in the bridge circuit.

The determination of the pressure equivalents of the superimposed chops was accomplished by statically calibrating the strain-gage voltage change with pressure. Fig. 2 shows three of these chops superimposed on typical injection pressure-time traces. A selection of five chopped square waves of different magnitudes was available in the bridge circuit for this calibration purpose.

A time calibration was provided on the pressure-time signal on the oscilloscope by two vertical pips located on the trace exactly 30 crank-angle deg apart. These pips (Fig. 2) were superimposed on the trace by means of a magnetic pickup and special circuit to be described later.

Flow-Rate Calibration – Since this method of measuring rate of fuel injection is based on using the spray-tip holes as an orifice-type restriction of a flow meter, an accurate relationship had to be found between the pressure difference across the metering element and the corresponding rate of fuel flow.

It was recognized at the outset of this work that the flow through the spray-tip holes of an injector is of a pulsating nature. However, originally it was accepted that any inferential head meter would measure flow correctly if the differential pressures are measured accurately. It was assumed then that if the strain gage could correctly measure the pressure drop across the spray-tip holes, the instantaneous-flow rates could be obtained from an experimentally determined steady-state pressure versus flow-rate calibration. On this premise a set of injection pressure data were taken and time versus rate of fuel injection curves determined. To check the accuracy of this thinking these curves were mechanically integrated to ob-

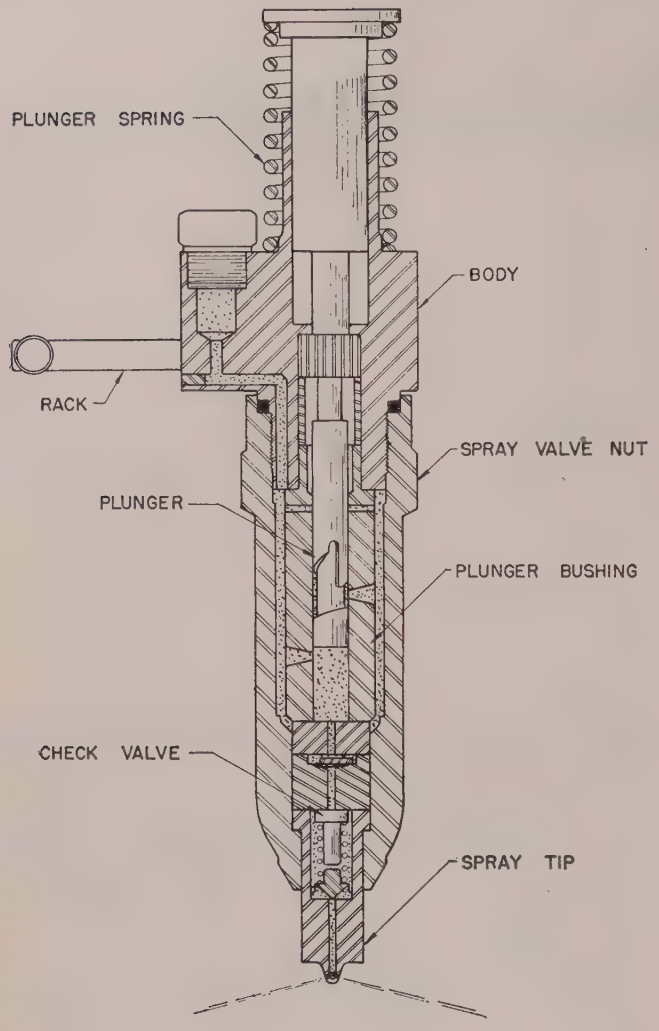


Fig. 1 – Schematic drawing of fuel injector

tain the total amount of fuel injected per cycle. The values thus obtained differed from the values determined by actual volume measurements by as much as 50%.

It was concluded that the errors found in this work were probably the result of the application of a steady-state flow calibration to a pulsating-flow condition. Therefore, it was deemed necessary to develop some method of deriving a pulsating-flow calibration which would accurately correlate the instantaneous pressure-flow rate phenomena occurring in the injector spray tip. This would permit accurate determination of the desired time versus rate of fuel injection curves.

The physical conditions existing inside the injector spray tip during injection are not of such a nature as to make a mathematical analysis practical. The amplitude, length, shape, and frequency of the injection pressure wave depend on the cam velocity and acceleration diagrams, engine speed and throttle conditions, and the physical properties of the fuel oil used. To attempt to resolve mathematically the interdependence of these variables would be at best a tedious, and extremely cumbersome procedure. Hence, it was felt that the desired pulsating-flow rate calibration should be derived in some manner from experimental data.

The time element involved in measuring coexisting pressures and flow rates complicates the determination of a pulsating-flow calibration. The coexistent pressures and flow rates must be known or measured during an infinitely small increment of time during the injection period. The strain-gage instrumentation is capable of providing the pressure-time relationship for an injection cycle. By this means the pressure corresponding to any point of time during the injection period is known. If, then, it is possible to associate a flow rate with some point of time during the injection cycle, one point on a pressure versus flow-rate calibration could be determined. Repetition of this process through a range of conditions would provide the desired calibration.

The approach which the authors used was to select an arbitrary flow rate and determine the time in the injection cycle at which that flow rate occurred. If it is initially assumed that the flow rate is proportional to the square root of pressure, as is true in the case of steady flow, then the time during the injection period when the instantaneous-flow rate is numerically equal to the mean effective flow rate for the injection period may be found. The validity of the above assumption is later checked by the accuracy of the resulting measurements. Plotting the square roots of the instantaneous pressures as ordinates with time (preferably in crank-angle degrees) as abscissas will result in a curve which, by the assumption stated, is proportional to the time versus rate of fuel flow curve. Fig. 3 shows such a curve. This curve can be integrated by mechanical methods and a mean effective ordinate determined as indicated in the figure. At the time when the ordinate of the curve is numerically equal to the mean effective ordinate, the actual instantaneous-flow rate through the spray-tip holes will be numerically equal to the mean effective flow rate for the injection period. This mean effective flow rate may be readily calculated from volumetric measurement using a burette, revolutions counter data, and the length of the injection period as determined by the strain-gage data. Hence, by this procedure, the time corresponding to a selected arbitrary flow rate (judiciously chosen as that flow rate numerically equal to the mean effective flow rate for the injection period) is determined. The pressure existing at that time may be found easily from the pressure-time curve or by squaring the mean effective ordinate of the square root of the pressure versus time curve. Thus, by this pro-

cedure, one point of the desired pulsating-pressure flow-rate calibration is found.

A typical pulsating-flow rate calibration as defined by experimental data using the foregoing method is shown in Fig. 4. As indicated in the figure these points are determined through a range of engine speed and throttle settings, thus providing a range of variation in the pressure and flow-rate conditions. When the relationship thus defined is plotted on logarithmic coordinate paper, the result is a straight line. Fig. 5 shows a typical flow calibration of an injector spray tip for steady-state and pulsating flow plotted in this manner.

Rate of fuel injection curves were then obtained from the strain-gage data using this pulsating-flow calibration. These curves were mechanically integrated and the total amount of fuel injected per cycle determined. The values thus obtained were compared with similar quantities obtained by burette measurements. This check showed an accuracy within 5%. Since this was believed to be approximately the accuracy of the strain-gage instrumentation, the use of the assumption employed in the derivation of the pulsating-flow calibration was considered to be justified.

It will be noted in Fig. 5 that the pulsating-flow curve is not exactly proportional to the square root of the pressure, but is actually proportional to the pressure raised to a power slightly less than 0.5. This suggests that to achieve a slight increase in accuracy the derivation should be repeated, using the proportionality indicated by slope of the calibration curve obtained on the first try. Such a procedure nets an increase in accuracy of a fraction of 1%, which is considerably less than the error inherent in the strain-gage instrumentation. Because of the small gain which might be obtained at the expense of a considerable amount of tedious labor, this procedure was not adopted in the work. Nevertheless it is recognized that the possibility of such a refinement exists.

It may be noted from Fig. 4 that the pulsating-flow calibration has been extrapolated slightly beyond the limits of the data. The possibility of such an extrapolation is suggested by the fact that the calibration plotted as a straight line on a logarithmic coordinate paper. The use of such an extrapolation was made necessary by the fact that the maximum injection pressures encountered exceeded the highest pressure which could be correlated with a flow rate in the process of establishing the pulsating-flow rate calibration. No difficulty was encountered due to this procedure.

It should be recognized at this point that the foregoing treatment of pulsating-flow rate measurement is not necessarily a general solution to this problem which would be applicable for all cases. Rather it is presented as an empirical method which yielded satisfactory results under the particular conditions encountered in this work. The effects on the pulsating-flow calibration of gen-

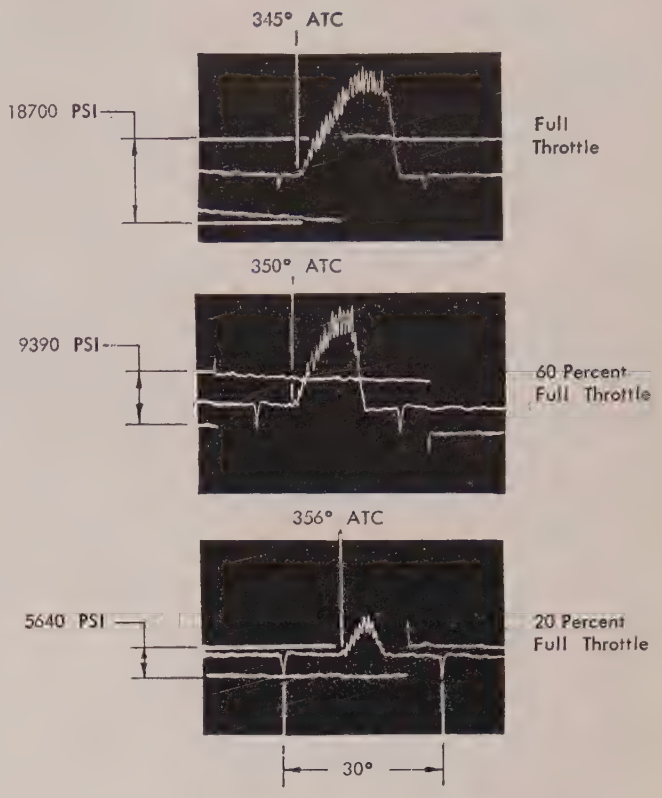


Fig. 2 – Photographs of injection pressure-time curves

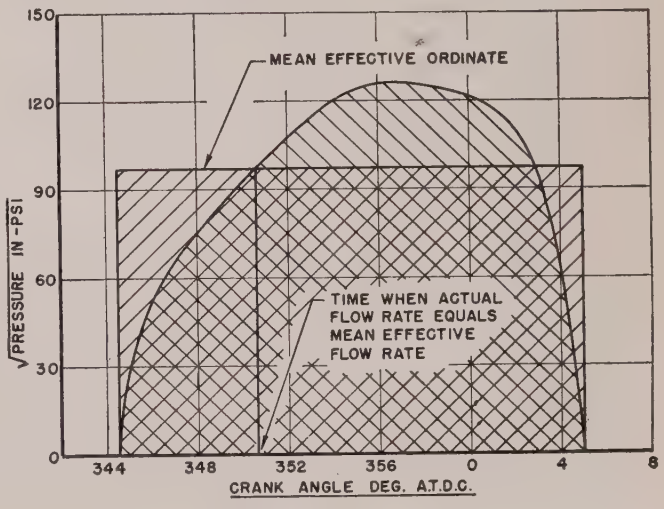


Fig. 3 – Diagram showing method of finding time when flow rate is equal to mean effective flow rate

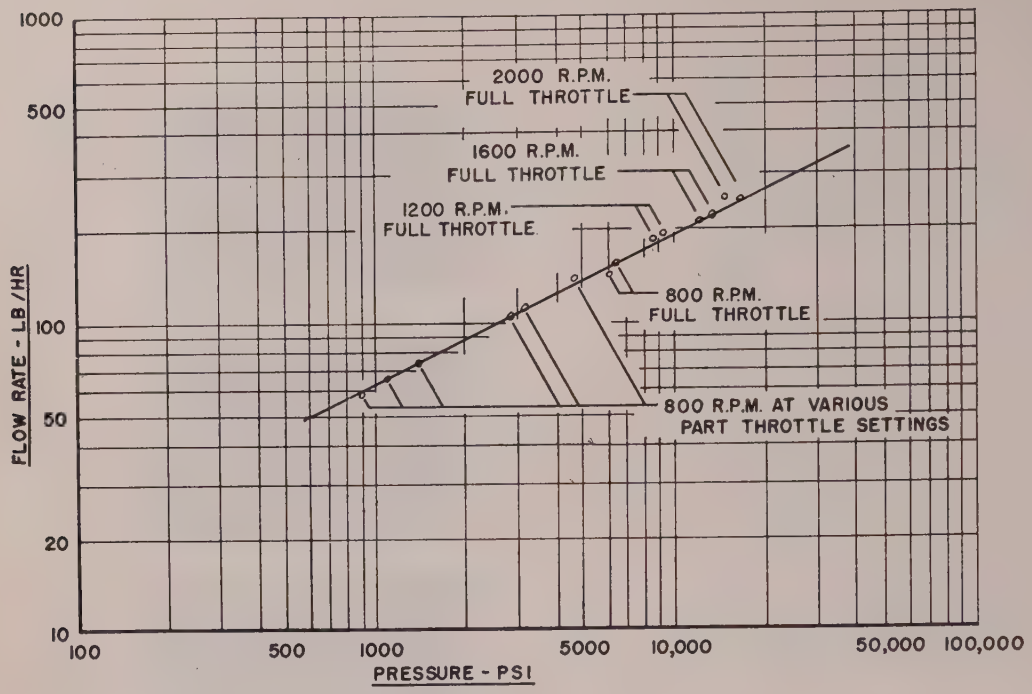


Fig. 4 – Typical pulsating-flow calibration of an injector spray tip

eral variation in frequency, amplitude, and harmonic content of the pressure variations, density, viscosity, and bulk modulus of the fuel oil, orifice size, approach conditions, and so forth, were not investigated.

Strain-Gage Instrumentation – In order to obtain accurate rate of flow determinations it is required that the strain gage indicates the pressure immediately ahead of the spray-tip orifices. For this reason the strain gage was mounted on the spray-tip walls, in spite of the resulting complications in the lead wire arrangement. Other locations for the strain gage such as the rocker components were considered, but tests to determine the possibilities of such locations indicated that the pressure drop due to intervening restrictions were objectionable.

To successfully accomplish its purpose the strain-gage construction must meet some rather severe requirements. The construction must be such that it senses the pressure inside the spray tip, but remains relatively insensitive to combustion-chamber pressures. A linear relationship must be maintained between injection pressure and the strain-gage signal at all times. The gage construction should remain stable under the adverse conditions of pressure and temperature encountered and should be reasonably durable.

The strain-gage construction developed to meet these requirements is shown in Fig. 6. An annular recess was ground into the spray-tip wall so that the remaining wall thickness was 0.046 in. This wall thickness was chosen as a compromise between providing the maximum amount of strain-gage sensitivity and still leave a reasonable cross-section of metal to conduct heat away from the tip. A smaller cross-section of metal would result in a hotter tip, which in turn would complicate the strain-gage construction. The strain-gage wire, which was 1-mil Advance wire, was wound into this recess and bonded with bakelite cement. One end of the winding was grounded by spot welding, while the other end was brought out by means of a short intermediate copper lead wire which was provided between the strain-gage wire and the lead wire from the bridge circuit. To minimize the sensitivity of the strain gage to the variations in combustion-chamber pressures encountered in the engine, provision was made to press a steel sleeve over the gage in such a manner that an air space was left between the gage and the sleeve. This technique successfully reduced the sensitivity of the strain gage to combustion-chamber pressure variation to an unobjectionable value.

Engine tests with an earlier spray-tip strain-gage construction, in which the gage wire was spirally wound into the recess provided, showed pickup of an extraneous signal. This signal was traced to small magnetic field changes in the cylinder caused by the motion of the cast-iron pistons. The presence of an inductive strain-gage winding in this varying magnetic field caused an extraneous voltage signal to be induced in the windings. Be-

cause of this the final construction shown in Fig. 6 features noninductive, bifilar-type windings, which successfully eliminated this source of difficulty. Fig. 7 shows an instrumented injector as used in an engine.

The resistance changes in the strain gage induced by corresponding pressure variations were converted to corresponding voltage signals by means of the circuit shown in Fig. 8. However, any suitable bridge circuit should give equally satisfactory results. This circuit is described as a potential divider circuit with a means for providing a pressure-calibration signal on the oscilloscope trace incorporated. This is done by shunting part of the ballast resistance. A selection of five shunt resistors is available for this purpose, depending on the magnitude of the calibrating signal desired. In order to be able to amplify the small d-c voltage changes caused by this shunting resistor with an a-c amplifier, the shunt was chopped by means of the breaker. Thus the pressure-calibrating signal appeared on the trace as a square wave. This square wave was statically calibrated for pressure by means of a high-pressure pump capable of developing over 20,000 psi pressure.

Test Apparatus – The test work was conducted on a General Motors model 3-71 diesel engine, which was instrumented with a special electrical timing device to provide the time calibration on the oscilloscope trace. This device consisted of a magnetic pickup attachment mounted on the end of the crankshaft which, in conjunction with a special strobolight trigger circuit, superimposes the two timing pips on the trace exactly 30 crank-angle deg apart. The pickup unit can be rotated with an indexed head, thus providing a means of moving the pips back and forth horizontally on the injection-pressure trace. By moving the left pip, whose position is calibrated with the indexed head in crank-angle degrees, to coincide with the beginning of injection on the trace, the time of the beginning of injection may be read. The same process can be used to locate the end of the injection.

A separate fuel system was provided for the instrumented injector. Incorporated into this fuel system was a burette for volumetric determination of the fuel consumption by the instrumented injector, and a system of pumps, pressure regulators, and heat exchangers for controlling the pressure and temperature of the fuel. For most of the test work these pressures were set at 25 psi on the inlet and 15 psi on the outlet of the injector. The other two injectors were supplied from the regular engine fuel system.

In the initial stages of the development of this method much preliminary work was done on a bench test setup in which the injector was actuated in a manner similar to its operation in an engine. By this means the added complications arising from the pressure and temperature conditions existing in the engine combustion chamber were avoided. The same systems of metering fuel to the

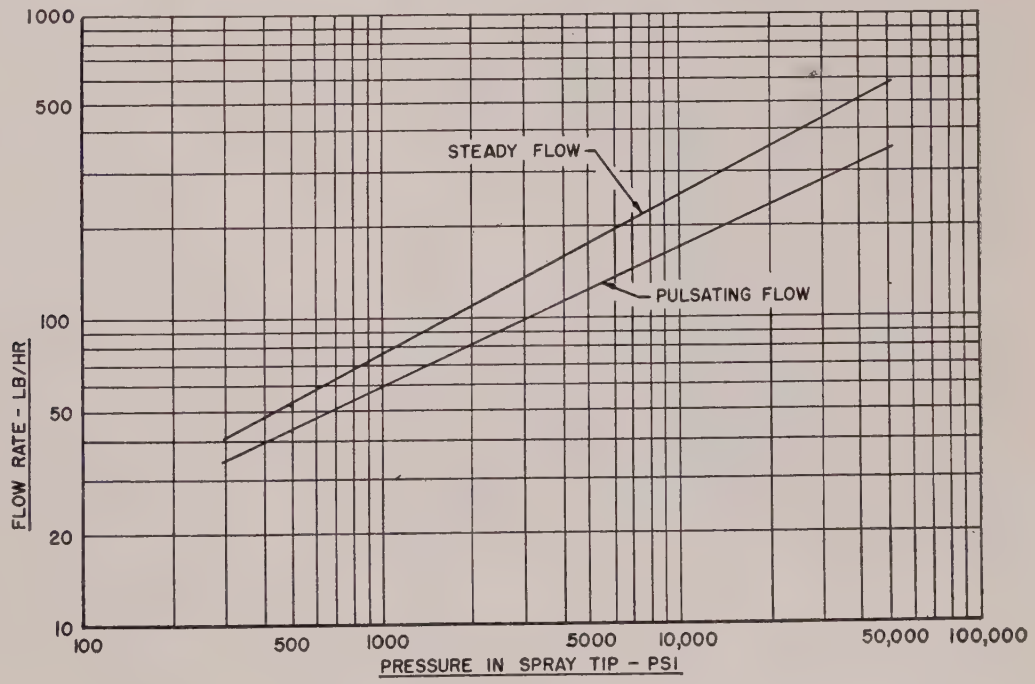


Fig. 5 – Flow-rate calibration of injector spray tip

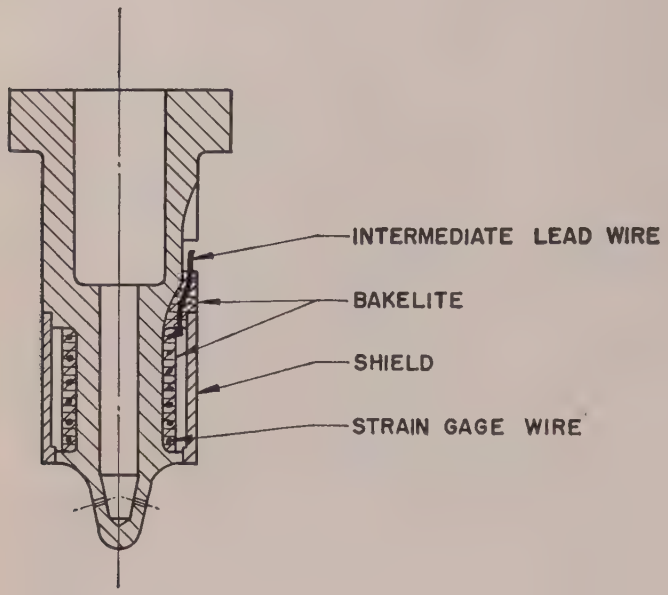


Fig. 6 – Strain-gage installation on injector spray tip

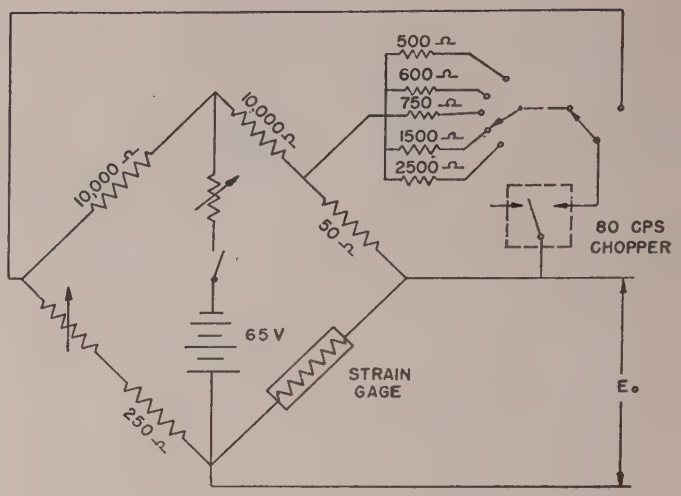


Fig. 8 – Strain-gage control-circuit diagram

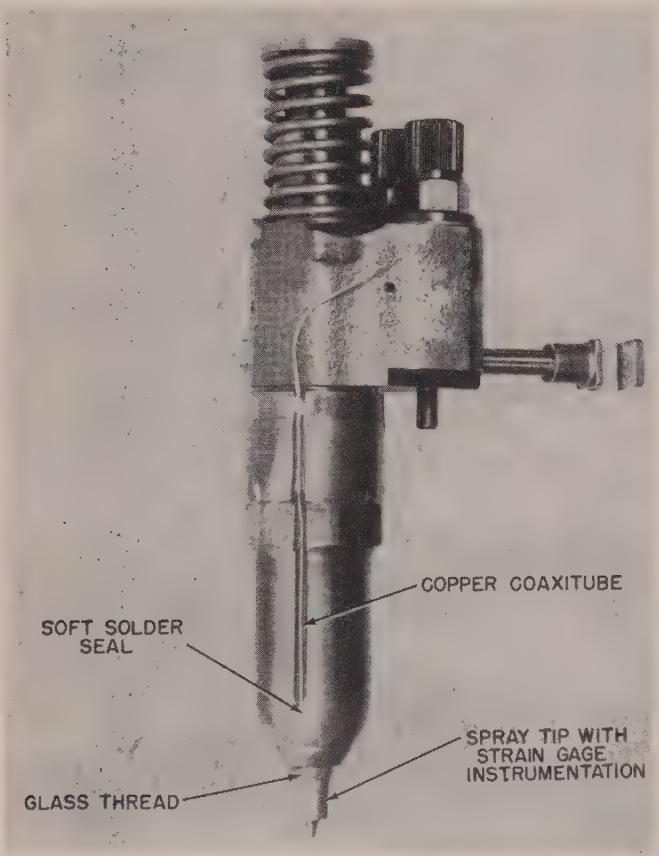


Fig. 7 – Instrumented injector assembly used for engine test

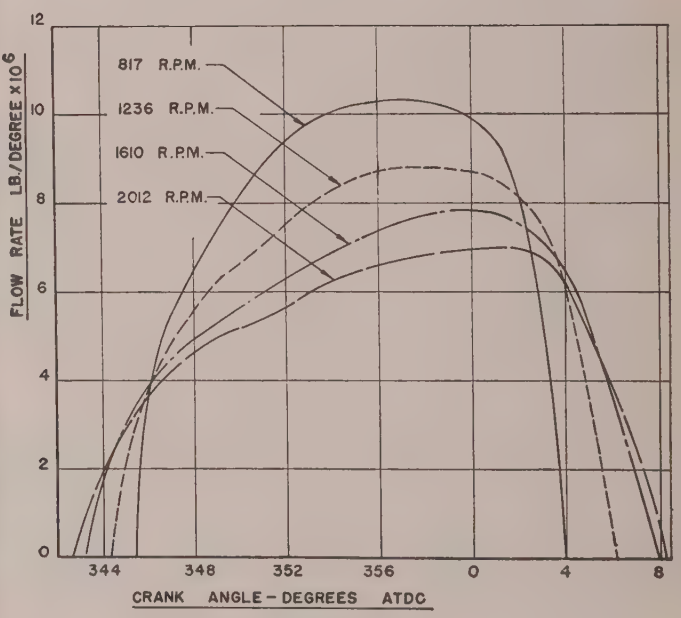


Fig. 9 – Rate of fuel flow during injection at full throttle

injector and providing a time calibration on the oscilloscope trace were used on this setup as was provided for the engine work.

Typical Injector Data – In order to supplement the description of the method and instrumentation used some model 71 injector data are included to show the type of information that may be obtained through the application of this method.

A family of rate of fuel injection curves observed at full throttle and various speed conditions are shown in Fig. 9. It is especially interesting to note from this figure that an increase in engine speed causes a decrease in the quantity of fuel injected per degree of crank-angle rotation. This is a direct result of the higher injection pressures induced by the higher plunger velocities as the speed is increased. These higher pressure levels cause increased fuel compression and increased leakage. Since the rate of fuel injection must equal the rate of plunger displacement minus the rates of leakage and volume decrease due to fuel compression, the higher pressure levels at the increased speeds result in lower rates of fuel injection. If these rates had been plotted in quantity injected per unit of time, then the curves would have exhibited an increase in rate of fuel injection with increase of engine speed.

Another significant change is that injection begins slightly sooner and ends slightly later as speed is increased. The earlier beginning of injection is probably due to the fact that the injection pressure buildup begins not when the plunger helix cuts off the spill ports, but when the restriction at this point is such that the rate of plunger displacement is greater than the rate at which the fuel is spilled. The effect of this restriction is more pronounced at the higher speeds. The increasing delay of the end of injection can be explained by similar reasoning.

The variation in rate of fuel injection with change in throttle setting is shown on Fig. 10. As would logically be expected, these curves indicate lower rates of fuel injection at the lower throttle settings.

In order to determine to some extent the effect of engine-cylinder conditions on injection characteristics, rate of fuel injection using the same injector assembly were run in the engine and on the bench test setup with atmospheric conditions in the receiver. Typical results of this work are shown in Figs. 11 and 12. A marked decrease in the length of the injection period, and increase in flow rates in the main portion of the injection period, was noted. This is believed to be due to the effect of the combustion-chamber pressures on the action of the spring-loaded spray-tip check valve and the tendency of the injector to function as a constant volume pump.

Typical oscillograph traces of injection pressures from the engine test work are shown in Fig. 13. The two traces shown represent the extreme operating conditions tested. One trace was photographically

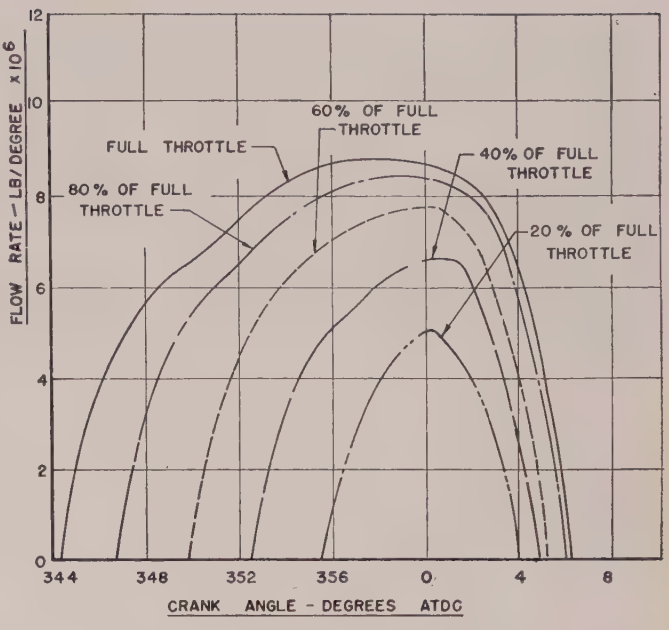


Fig. 10 – Rate of fuel flow during injection at 1200 rpm

recorded at 800 rpm and no engine load. The other trace represents the injection pressure picture at 2000 rpm and full-load conditions. Differences in oscilloscope gain settings used preclude a direct visual comparison of the injection pressures obtained at these conditions.

It will be noted that the traces show a high-frequency pressure vibration superimposed on the injection pressure-time curve. These high-frequency pressure variations proved to be the result of the check valve vibrating. This was concluded by observing the effects of changing the mass of the check valve or the spring constant of the check-valve spring.

Having established the cause of these vibrations it remained to determine the location of the effective injection pressure curve. For this purpose a specially designed dash-pot check valve and stop was used to successfully eliminate the check-valve vibrations without affecting the release pressure of the check valve. Injection pressures obtained with this arrangement were compared with pressures obtained by plotting injection pressure curves midway between the extremities of the vibration pressure waves. This comparison showed good agreement, indicating that the injection-pressure curve lies midway between the extremities of the vibration waves.

Since the flow rate through the spray-tip holes is a function of injection pressure, one might more fully appreciate the reasons why time-rate of injection curves take their particular shapes by studying the causal relationships which exist between injection pressure and the sequence of events which occur when the plunger helices close and open the

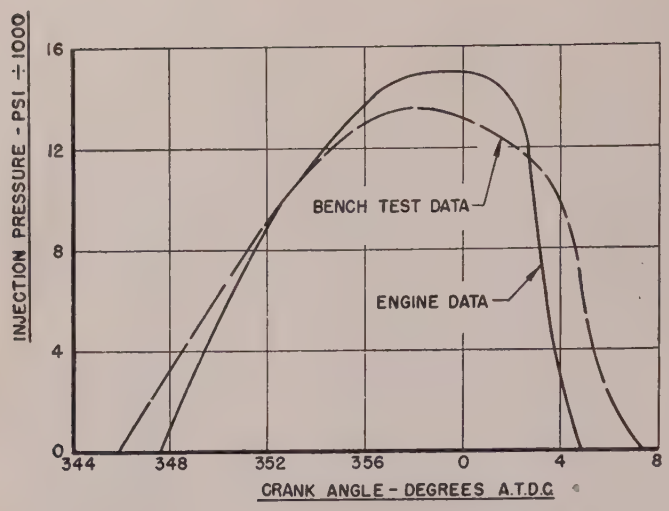


Fig. 11 – Injection pressures as measured in an engine and on bench test apparatus

spill ports. In a paper by Green[1] the pressure rise during the injection is discussed. It is pointed out that the pressure rise would continue until the discharge velocity is sufficient to pass fuel at a rate equal to the rate of the plunger displacement. In practice this condition is not normally attained. Only a part of the plunger displacement represents injected fuel; while a remaining part serves to further compress the trapped fuel. If the plunger velocity is great enough to maintain a rising pressure, its instantaneous velocity and the exact form of the cam profile are of secondary importance.

[1] See "Fuel Injection in Modern Oil Engines," by W. A. Green, Paper S210, Jan. 19, 1950, p. 24. Published by the Diesel-Engine Users Association, 56 Victoria Street, London, England.

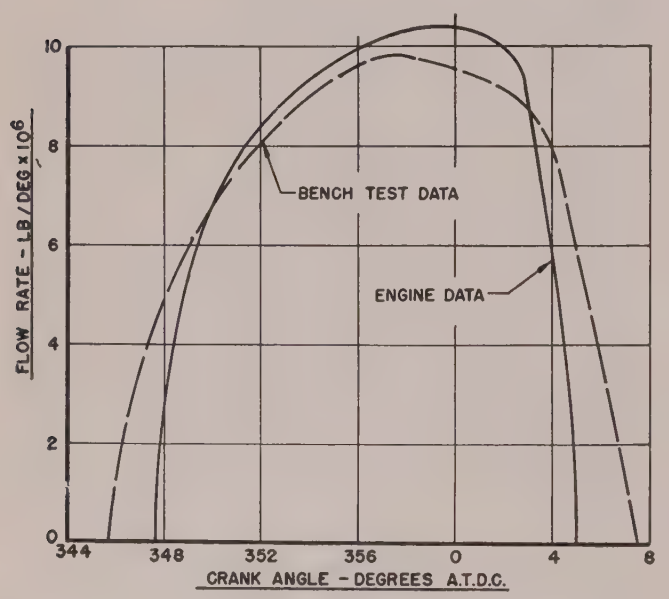


Fig. 12 – Rate of fuel injection as measured in an engine and on bench test apparatus

Pressure rise increases in this manner until spill occurs due to the uncovering of a spill port by the plunger helix. The effectiveness of the spill is small at first, but increases in effect rapidly during succeeding degrees, causing the pressure to fall correspondingly.

In light of these considerations it is of interest to examine some of the characteristics of the pressure-time curves in Fig. 14. It will be noted that the initial slopes of all the curves are quite similar indicating that the rate of pressure rise may depend on speed, but not throttle setting. The injection interval during which this relatively constant rate of pressure rise occurs is probably the interval during which the spill ports are completely shut off by the plunger. As the spill port is uncovered by the helical plunger surface, the rate of pressure rise decreases and becomes negative where the pressure falls off. The amount of the decay of the rate of pressure rise may be taken as an indication of the effectiveness of the spill-port opening.

It is beyond the scope of this paper to attempt a thorough treatment of fuel-injection phenomena as such. The foregoing discussion and the data presented were included only to show some of the relationships that may be explored with the aid of the method and instrumentation described.

Flow Characteristics of Spray Tips – Since the method of measuring rate of fuel injection is based on the treatment of the spray-tip holes as the orifice of a flow meter, it would be well to consider some of the flow characteristics of these holes. It will be readily understood that the small size of these holes (0.006 in.) complicates the determination of their physical dimensions as well as such characteristics as the coefficient of discharge.

In considering the physical shape of the holes it seems somewhat illogical to call them orifices in the usual sense of the word. They are actually more like tubular passages. The length/diameter ratio of model 71 injector spray tip is approximately 4.2. Similar length/diameter ratios would probably be encountered in the injector nozzles of other makes since this parameter is usually determined from considerations of practical mechanical design, fuel penetration, and atomization in the cylinder. Metal failure at the metering holes, due to the high injection pressures, would also preclude the use of a hole shape approaching that of a sharp edge orifice.

The difficulty of accurately measuring the physical and flow characteristics of these small metering holes means that each tip must be calibrated experimentally. In this work steady-flow calibrations were obtained by means of a high-pressure pump. When plotted on logarithmic coordinate paper with pressures as abscissa and flow rate as ordinate, the flow calibration is a straight line with a slope of one half which indicates that flow rate is proportional to the square root of the pressures as expected.

In order to investigate the effects of downstream pressure on the steady-flow calibrations, a model

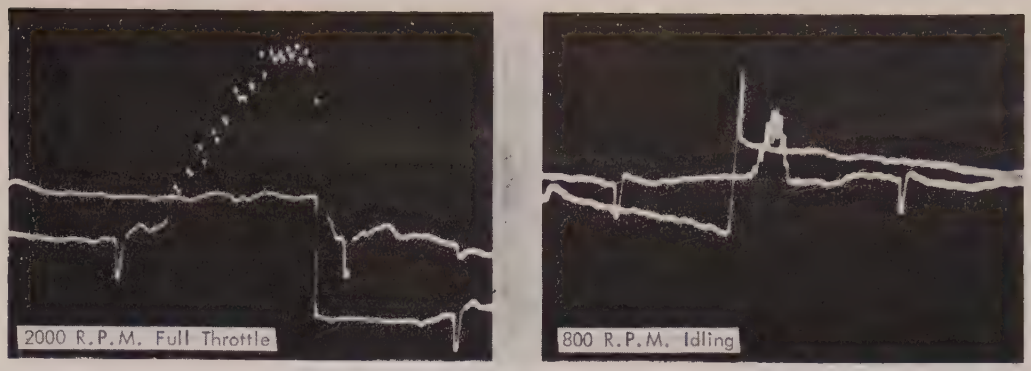


Fig. 13 – Injection pressure-time traces from engine tests

71 injector spray tip was calibrated for flow at a number of downstream pressure levels. The results of the work are shown in Fig. 15. It may be noted that the calibration at high back-pressures all blend into a common flow curve when the ratio of upstream to downstream pressure exceeds a certain value. For the particular tip and fuel tested in this application this ratio had a value of about three. Thus when the ratio of upstream/downstream pressure exceeds this value the flow rate becomes independent of the downstream pressures and is a function of the upstream conditions only. These results were checked using different spray tips. Similar results were obtained in all cases.

Previously established orifice equations describe the flow rate as being a function of the pressure difference across the restricting element. The concept that flow rate depends on the upstream pressure only after a certain ratio of upstream to downstream pressure is exceeded is not readily explainable by these relations. It appears from this that some other type of analysis is required to describe adequately the flow phenomena through small holes such as the ones in injector spray tips.

The lack of sufficient information on flow through small orifices has been recognized by other investigators. Studies of this type of flow phenomena were conducted for the National Advisory Committee for Aeronautics by Joachim[2] and Gelalles[3]. Some of the results of these studies were similar in character to the results shown in Fig. 15. The peculiarities in flow behavior reported by these investigators were interpreted as being due to variation in the coefficient of discharge.

It might appear from these results that combustion-chamber pressures in an engine would have little effect on injection since the ratio of upstream/downstream pressure exceeds three for nearly all of the injection period. It should be remembered, however, that these results were obtained for steady-flow conditions and hence are not necessarily applicable to the pulsating-flow conditions in an operating injector. Subsequent work has shown that the pulsating-flow calibration is appreciably affected by back pressure.

Spray-Tip Temperatures

Because electrical resistance strain gages are sensitive to temperature variations, a study was made of spray-tip temperatures and their cyclic variations. This was done by means of a thermocouple welded to the spray tip. To approximate better the heat transfer and temperature conditions the strain gage would encounter, this tip was recessed in the same manner as the instrumented tips were. It was found that the spray-tip temperature increases with speed and throttle setting. The maximum temperature observed was 470 F at 2000 rpm and full throttle.

To observe the degree of temperature cycling experienced by the spray tip, the output of the ther-

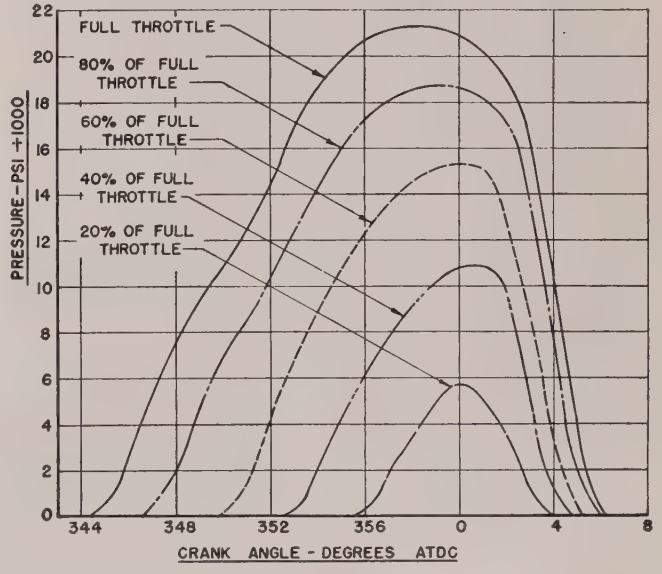


Fig. 14 – Injector-tip pressure during injection period for various throttle settings

[2] NACA Technical Report No. 224 (1926): "An Investigation of Coefficient of Discharge of Liquids through Small Round Orifices," by W. F. Joachim.

[3] NACA Technical Report No. 373 (1931): "Coefficients of Discharge of Fuel Injection Nozzles for Compression-Ignition Engines," by A. G. Gelalles.

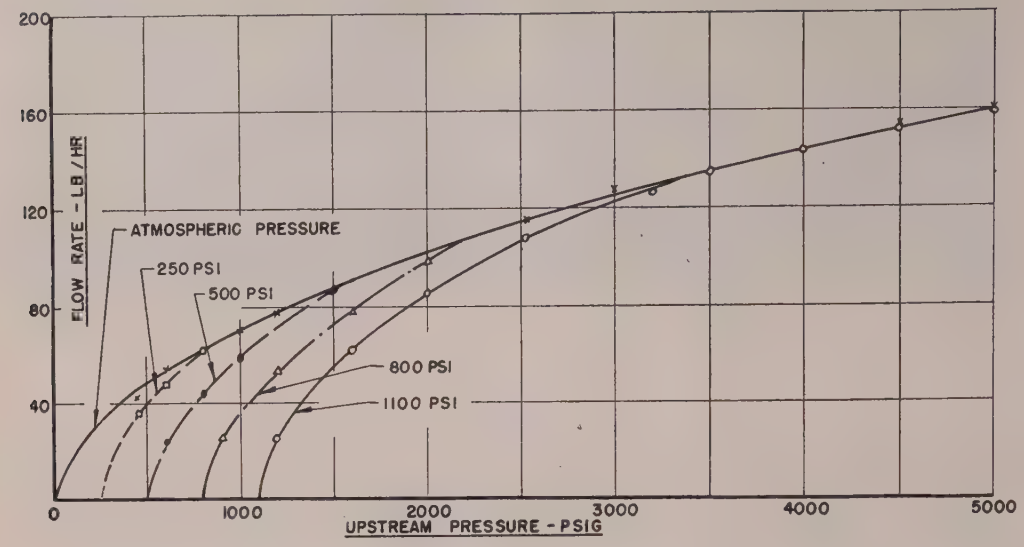


Fig. 15 – Calibration of spray tip for flow versus pressure with various back pressures

mocouple was fed through an amplifier and then into an oscilloscope. Since the mass of the thermocouple was small relative to the mass of the tip, the response rate of the thermocouple should be sufficiently fast to follow any temperature cycling of the spray tip. When the nominal gain characteristics of the amplifier and oscilloscope were used, it was possible to observe any temperature cycling greater than 2 deg.

It was found that no temperature cycling could be detected in the normal test range of 800 to 2000 rpm. However, when the speed was reduced to 600 rpm at full-throttle conditions, a temperature cycling of 3 or 4 deg F was observed. Thus it appears that at these conditions the variation in combustion-gas temperature was slow enough that sufficient time was allowed for the heat transfer process to cause a detectable temperature change. At the higher speeds the time interval between gas temperature variations was decreased so that the heat transfer processes were less effective in varying the tip temperature.

Durability of Strain-Gage Construction

Although only a limited amount of testing time has been accumulated, this work has indicated that the durability of this installation is satisfactory. Two injector spray tips have been instrumented in this manner to date. One of these tips has been in service for 2 hr at various engine operating conditions. The other tip has been in use for about 3 hr of test work at various engine operating conditions plus 2 hr of the following cycle of operation: 5 min at 800 rpm and no load, 5 min at 800 rpm and full load, and 5 min at 2000 rpm and full load. No change in the quality of the signal produced after this time was evident.

Features of Strain-Gage Method of Measuring Rate of Fuel Injection – The method of measuring fuel-injection rates described has several advantages and disadvantages when compared with the rotating receiver method in which rate of fuel injection is measured by passing a series of collecting cells under an injector nozzle during the injection period. The apparent advantages of measuring fuel-injection rates by the method presented are as follows:

1. More comprehensive data are obtainable. In addition to fuel-injection data, information may also be obtained on injection pressures, injection timing, and check-valve vibrations. The cause and effect relationships existing between these variables may be studied.
2. It can be used in an operating engine or on bench test setup.
3. Transient pressure changes may be observed and studied.
4. No changes in the fuel-flow passages of a standard injector is required.

The disadvantages of this method are listed below:

1. It is more complex and less direct than the rotating receiver method.
2. It requires the use of a high-pressure pump if injection-pressure data are desired.
3. Installation of the strain gage requires skilled personnel.

Acknowledgment

The authors wish to express their appreciation to Wayne Chapman of the GMC research laboratories, who designed the strain-gage bridge circuit and assisted in problems connected with auxiliary electronic instrumentation.

DISCUSSION

Rotating Collector Measures Injection Rate

– Paul G. Burman
American Bosch Corp.

THIS paper is a welcome addition to the meager literature available on this subject, particularly with the growing demand for more precise control over the injection process, as diesel engine speeds and power outputs are being upped. It is now recognized that the combustion efficiency of a diesel engine is dependent not only on such factors as combustion-chamber design, and on the duration, dispersion, and atomization of the fuel spray, but also on the rate at which the fuel is injected into the engine cylinder. Some of the progress which has been made in achieving the high efficiency and outstanding performance of the modern diesel engine can be attributed to the continual improvements which have been made in the injection equipment to satisfy the increasingly stricter requirements.

Because of the complexity of the fuel-injection process, it is not possible to predict with scientific precision what the exact injection characteristics will be for equipment selected on a design basis, but it is necessary instead to rely on the results of bench and engine tests for final approval of the injection equipment for application to a specific make of engine. As manufacturers of injection equipment we have, therefore, a considerable investment in test equipment and instrumentation, and we are always on the lookout for new techniques of testing and analysis This paper is, therefore, of considerable interest to us.

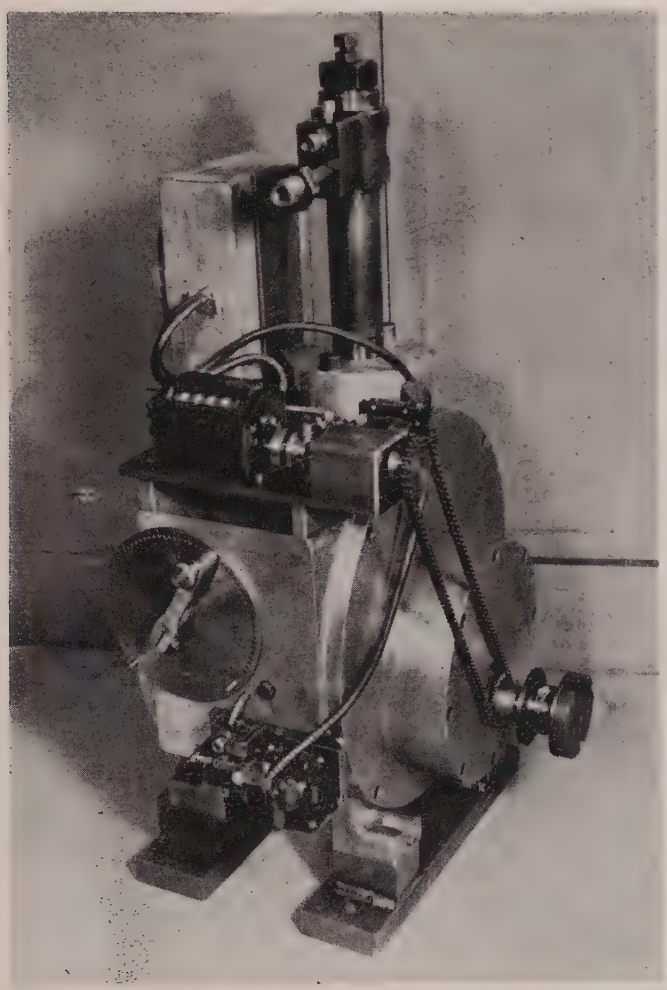


Fig. A – Slotted disc rate of discharge apparatus

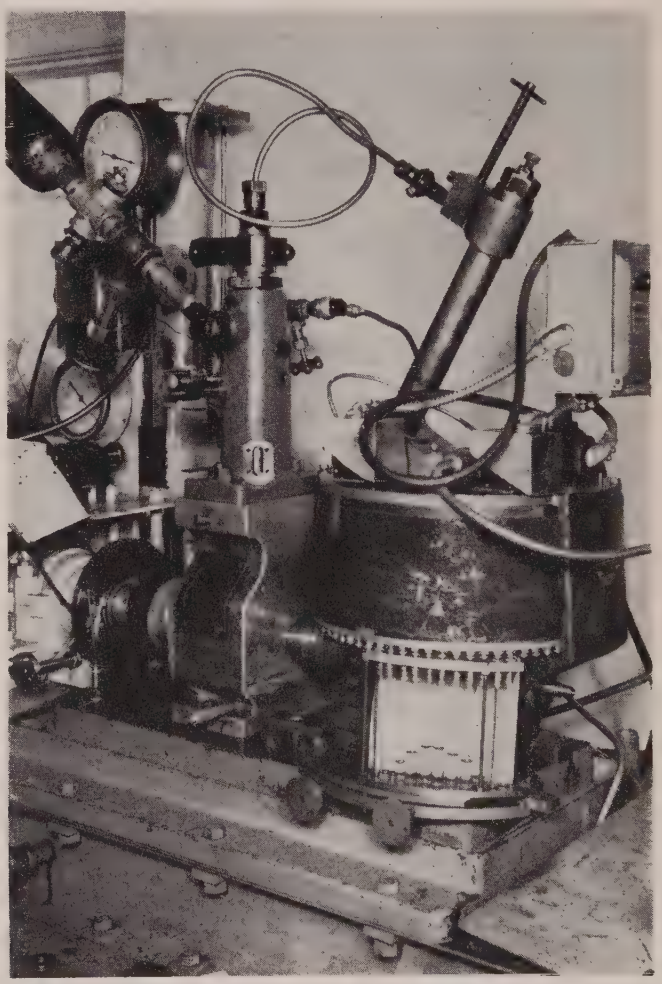


Fig. B – Rotating collector rate of discharge apparatus

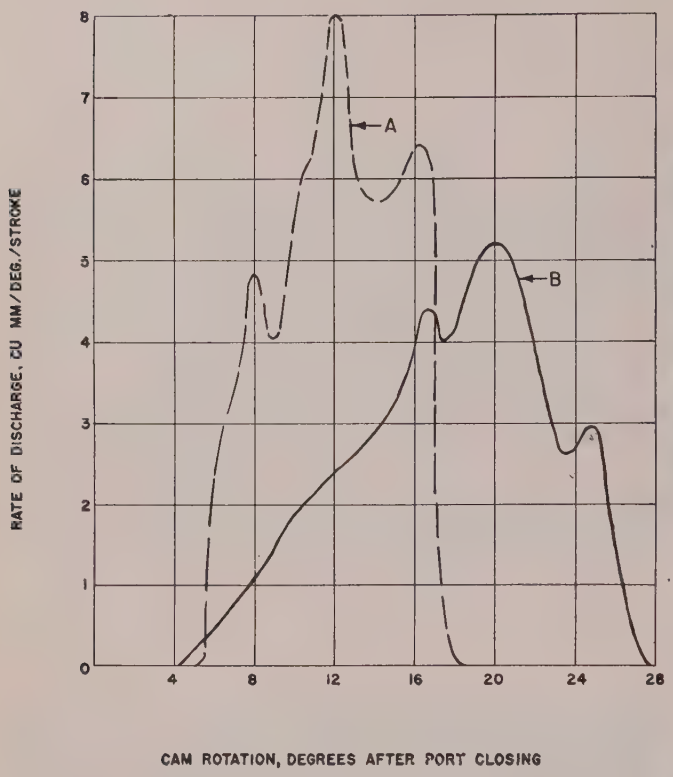


Fig. C – Comparative rates of discharge for 4⅜ x 5 in. automotive diesel engine – fuel quantity 67 cu mm per stroke at 1000 pump rpm

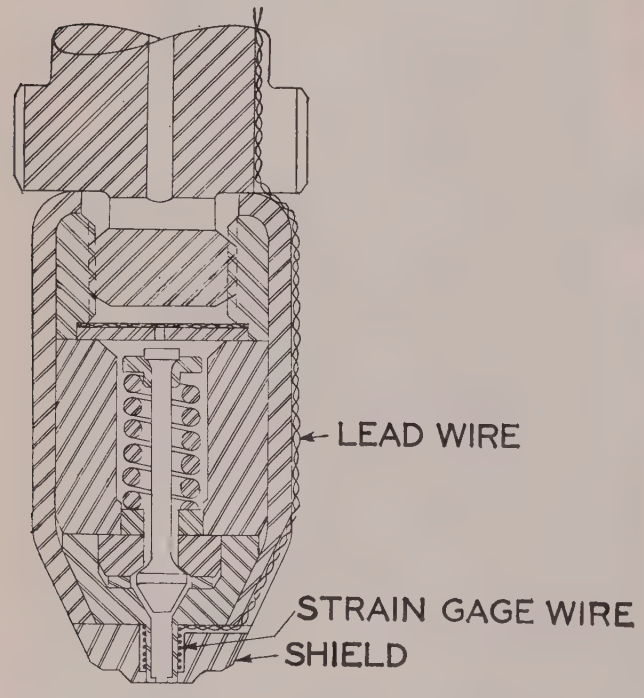


Fig. D – Standard Caterpillar fuel valve nozzle assembly reworked to include a strain gage

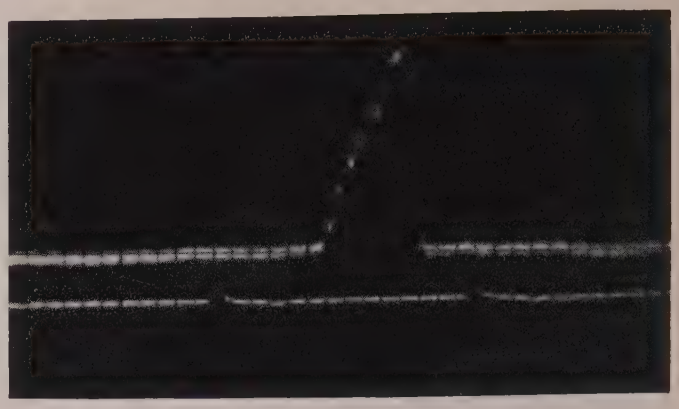


Fig. E – Injection pressure-time traces – part load at 1800 rpm

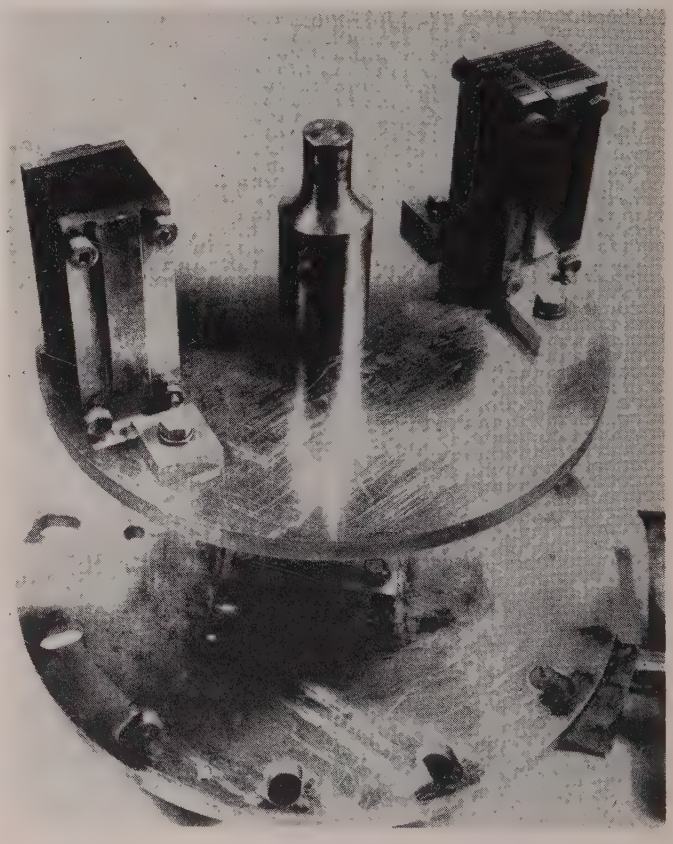


Fig. F – Turntable with collector packs mounted

Fuel-injection test methods have been steadily improved. Twenty years or more ago the spray duration, as well as the relative rate of discharge, was determined by measuring the length and depth of the groove cut by a nozzle spray in a strip of blotter paper attached to the engine flywheel. Shortly thereafter the introduction of commercial stroboscopic lamps permitted more convenient and accurate determination of the nozzle spray pattern and duration. Various optical indicators were also devised for measuring nozzle valve lifts and instantaneous fuel pressures, but they were rather inconvenient to use. For measuring rates of fuel discharge from the nozzle, simple slotted disc stroboscopes, Fig. A, and later rotating collectors, Fig. B, were then developed. These instruments are generally accepted as providing the best method for obtaining direct quantitative rates of discharge, despite their limitations in giving average values for numerous cycles instead of those for a single injection. With the commercial development of the cathode-ray oscillograph various methods of determining rates of discharge were tried, such as using photoelectric cells to measure the change in density of a light beam passing through a fuel spray or in measuring the impact of the fuel jet striking a diaphragm, but primarily it has been used for pressure measurements.

The first pressure pickups used with the cathode-ray oscillographs were of the piezo-crystal type, then of the magnetic type, but it is only recently that temperature-compensated strain-gage pickups with suitable amplifiers have become available that could be relied upon for quantitative pressure measurements. Because of their small size and adaptability these pickups often permit pressure measurements in such locations in the injection system as would not otherwise be possible. Our first application of strain gages for pressure measurement was made by necking down the diameter of a portion of the discharge tubing and winding the strain gage around it. The ingenious application of a strain gage in a nozzle tip as described in this paper is an example of its versatility.

Now that it is possible to accurately measure the instantaneous-injection pressures in the nozzle tip, it is a logical step to try to calculate the rate of discharge from these pressures and the steady-pressure flow characteristics of the nozzle. We also tried this and found the results in error though not to the extent mentioned in this paper, probably because of the lower injection pressure in our case. The method described in this paper has been tried with satisfactory results, but for our purpose the more direct and quicker method of measuring the rate of injection with a rotating collector is preferred. Since this paper shows fairly close similarity between the rates of injection as determined in an engine and on the bench except that the duration is slightly shorter in the engine, which our tests also confirm, we believe that bench tests of rate of injection will generally suffice. Moreover, when a variable-orifice-area nozzle such as the pintle type is used it adds so much complication to this method of calculating the rate of discharge as to preclude its use in this case altogether.

Because of the similarity of the pressure-time and rate of discharge versus time curves, and since we are usually interested in trends rather than absolute values, it is generally sufficient to assume that the injection-rate diagram is the same as the pressure diagram. As the final objective in determining the rate of discharge is ultimately to control combustion, it would be of interest to learn what correlation the authors have found between engine performance and rate of fuel injection into the combustion chamber. Fig. C shows comparative rates of fuel discharge which we obtained with two variations of injection equipment for an open-chamber engine. The same power and fuel consumption was obtained with both, but the steeper and shorter rate A gave more combustion noise and exhaust smoke than the injection equipment with rate B.

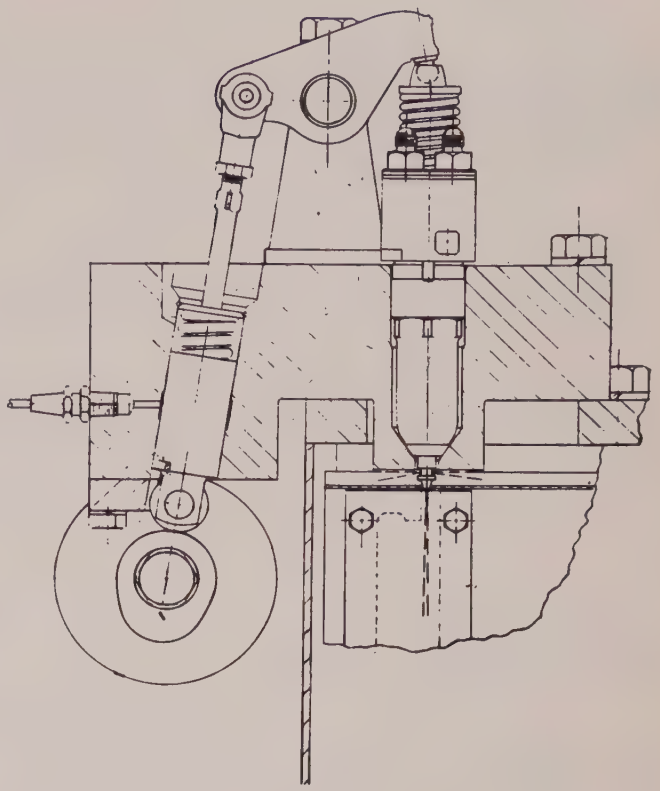


Fig. G – Injector mounting for rate of fuel-injection fixture

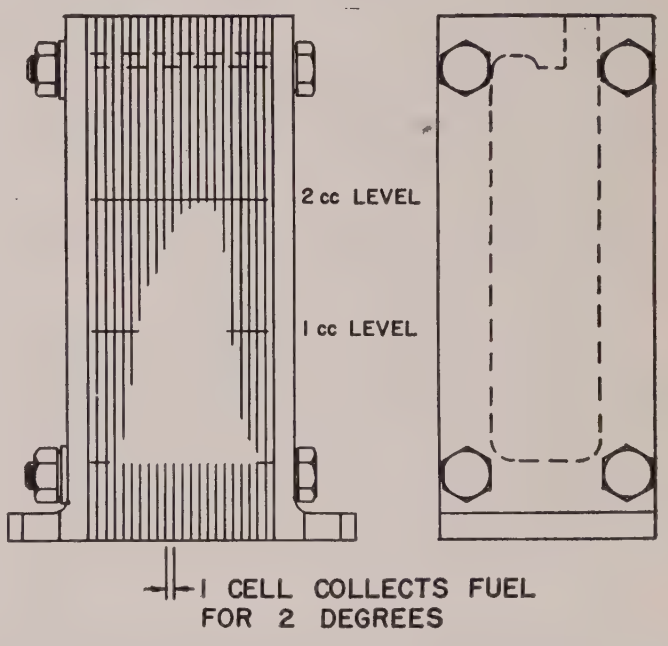


Fig. H – Collector-cell assembly – 1 cell collects fuel for 2 deg.

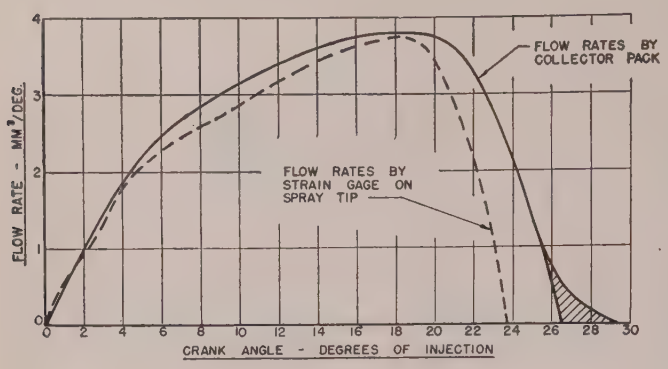


Fig. I – Rates of fuel injection as measured by strain gage on spray tip and by collector pack – GMC model 71-60 injector, 2000 rpm, full throttle

Diesel Combustion Studies Aided by Advanced Methods

– Stanley J. Kranc
Caterpillar Tractor Co.

IT is evident that the development of the method described for measuring rates of fuel injection resulted from a great deal of very careful work. In our laboratory work, we have made somewhat similar measurements. Fig. D shows a cross-section of a standard Caterpillar fuel-valve nozzle assembly, reworked to include a strain gage around the fuel passage, upstream from the orifice. In comparison to the 0.046-in. wall thickness mentioned by the authors, the wall thickness in this case is only 0.008 in. In this nozzle assembly, the fuel pressure rise is not as high as indicated in the paper.

Special and Standard Injectors

Fig. E shows the pressure-time diagrams obtained. We also attribute the high-frequency pressure fluctuations to check-valve vibration.

Referring to the paper, perhaps one more disadvantage could be listed. The necessity of using a special, rather than a standard, injector would be undesirable in some types of investigational work.

Possible Use of Method

And another advantage might be added. There is the possibility of using this method in a bench test setup, with the fuel being injected into a chamber pressurized at a constant level with an inert gas. This might provide a better correlation of bench test data with engine data.

With the strain gage wound on a wall thickness of 0.046 in., it would appear that obtaining the desired sensitivity at the low idle condition might be a problem.

System Can Be Modified

With the development of a reliable method for measuring rate of fuel injection in an operating engine, there is the opportunity to more accurately determine what the engine wants in the way of injection rates. Then, the fuel-injection system can be modified in an effort to satisfy those wants.

Diesel-engine combustion studies in the future should be much less handicaped than those in the past, because advanced methods of observing fuel-injection system characteristics, such as the one described in the paper, are being developed.

Injection Rates Measured By Collecting-Cell Method

– William S. Shade
Diesel Equipment Division, GMC

AFTER careful consideration of this paper, I wish to discuss briefly the collecting-cell method of measuring rate of fuel injection, as designed and developed by the Diesel Equipment Division.

These data were requested by several diesel-engine manufacturers to see if they could improve combustion if the rate of injection from a unit injector was known. After considerable time and money had been expended, a machine was devised to measure the rate of any GMC unit injector just by changing the method of locating and driving each model. With the aid of illustrations I will describe this method as used by our division.

Use of Collector Packs

Fig. F shows the turntable of the fixture with the collector packs. This collector assembly is designed to rotate at one-half of the camshaft speed so that one of the packs of collecting cells is passed underneath the injector spray tip hole each time an injection period occurs. Using two collector packs allows the turntable to remain in balance as fuel is collected and one pack serves as a check against the other.

Special Spray Tips

Fig. G shows the method of mounting and driving the injector over the turntable. This shows a standard model 71 injector, except for the spray tip, driven by model 71 engine parts. This special spray tip has one hole which ejects fuel downwards, with the remaining holes equally spaced around the periphery of an annular recess in the tip wall. These spray tips were calibrated and found to have the same flow characteristics as conventional tips. Only the fuel ejected from the lower hole is collected, with the fuel from the side holes being deflected by the plate between the injector and the collector.

Two Methods Compared

Fig. H shows the collector-cell assembly which consists of layers of transparent lucite separated by thin steel shim stock. Each individual cell collects for 2 deg of camshaft rotation. This illustration shows a typical collection from one of the test runs.

Fig. I shows a comparison between the rate of injection as measured by the collector packs and as measured by the strain gage on the spray tip, as reported in this paper. A close comparison of all the speeds and throttle settings checked points to the conclusion that if the tests had been run with the same injector and driving mechanism the two methods of measurement would check each other.

Both Methods Accurate

We feel that the rate of fuel injection can be measured accurately by either using the strain gage on the spray tip or the collecting-cell method. For bench test work the collecting cell is more simple and direct but for fuel-injection studies in which the effects of engine-cylinder conditions are to be included it is necessary to use the strain-gage method.

Engine Cylinder Pressure Measurements

J. D. McCullough, Ethyl Corp.

This paper was presented at the SAE Annual Meeting, Detroit, Jan. 15, 1953.

NOVELTY or improvements occasionally happen in a burst of genius, but most often they are the result of long effort and study. The transportation industry can be cited as an example, where the hard row of research and development has been assiduously cultivated.

The improvements in engines and chassis have been the result of years of searching for better ways to utilize mechanical power. It is generally accepted that maximum utilization of any means requires a rather complete understanding of the principles involved and the effects of various design variables. Consequently, mechanical and chemical engineers have long sought a more complete understanding of the combustion process. These men have found a common interest in the study of petroleum fuels being burned in an engine. The chemical engineer is concerned with the problem of providing chemical energy in a useful form, while the mechanical engineer is concerned with the problem of converting chemical energy into heat energy from which he can obtain useful work.

In piston engines, this energy conversion takes place in a confined space, causing pressure differences which can be harnessed. These same pressure differences are also a primary source of data for the study of energy changes in the internal-combustion engine. Such information is useful for understanding the burning process and the roles played by those parameters over which the engine designer has some control. Data from a plot of combustion-chamber pressure versus crank angle degrees can give information for evaluation of combustion-chamber shapes, mechanical loads caused by gas pressures, adiabatic exponents of the air cycle, fuel characteristics, carburetor and manifold design, valve timing sequences, and other design considerations. Consequently, engine cylinder pressure diagrams are assuming increased importance in laboratories devoted to engine research and development.

On the other hand, petroleum refiners use combustion-chamber pressure diagrams and related techniques, to study the behavior of fuels in en-

ALTHOUGH there is no substitute for practical experience, this paper will provide useful guidance for those who are contemplating engine cylinder pressure measurements for the first time.

A frank discussion of the dangers inherent in some common assumptions and short cuts illustrates the need for continuous appraisal of test methods even by veteran users of engine indicator equipment. The new techniques and new transducers described may stimulate rapid advances in the art.

J. D. McCULLOUGH **is instrument engineer and assistant head of the instrument section in the research laboratories of Ethyl Corp.** He joined the organization in 1946. During the four previous years he was instrument engineer at Chrysler Corp. He received his B.S. degree in electrical engineering from the University of Cincinnati in 1942. In 1944 he received his master's degree in automotive engineering from the Chrysler Graduate School.

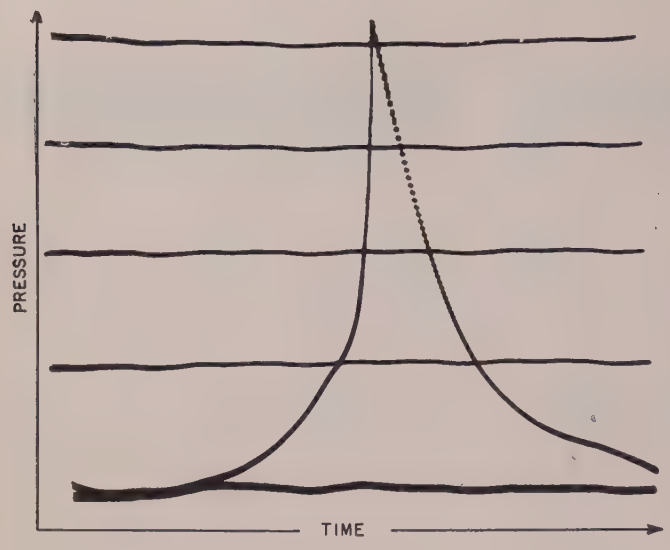


Fig. 1 – Continuous indicators require only one explosion to show cylinder pressure as a function of crank angle

gines. Data on ignitability, ignition delay, rate of fuel burning, precombustion reactions, and antiknock quality afford the refiner a means of evaluating his product in relation to customer requirements and to competition. Such data also provide a systematic means of relating inspection data and bench tests to actual field use in engines. To the refiner this sort of knowledge could lead to changes in refinery processes which would yield a more satisfactory product.

Recognizing the potential value of data from combustion-chamber pressure measurements, many laboratories have searched for suitable methods of obtaining such data. As is usually the case, there is no one technique superior in all respects to other techniques. Therefore, the user of such equipment is faced with the problem of selecting equipment and techniques most suited to producing the desired results. To a large extent, the success of the work to be undertaken depends upon careful analysis of the alternatives available.

There are three important factors governing the choice of equipment and techniques for an individual problem. These factors are:

A. Purpose of test.
B. Precision desired of test results.
C. Configuration of combustion-chamber and engine.

The purpose of the test may be to make an initial survey of a relatively new field where a lot of background data are desired in a short time. This type of test places less emphasis on rigid control of accuracy and refinement of technique in favor of broad conclusions from the trends observed.

On the other hand, the purpose of the test may be to obtain the best quantitative data possible to reach a particular conclusion. Such a test requires the greatest precision obtainable with contemporary equipment and techniques. A detailed analysis of the possible sources of error is essential if the resultant data are to be considered meaningful.

In addition to the purpose of the test and the precision desired of the test results, the physical configuration of the engine may constitute a third factor modifying the choice of equipment and technique. Despite the user's ingenuity, limitations due to space, temperature, or other environmental conditions may preclude the use of the most desirable types of equipment. Less satisfactory instruments or techniques may offer the only alternative. In such cases the test results must be interpreted by someone fully aware of the sacrifices made.

Which Technique? – The initial decision to be reached involves the choice of an operating technique which will best serve the purpose of a particular test. The alternatives available may be divided into two categories. These categories separate the techniques into *instantaneous* pressure-recording versus *point-by-point* pressure-recording. By instantaneous recording is meant the recording of a complete history of engine cylinder pressure (as plotted against crank angle or time) for each revolution of a 2-cycle engine or for each two revolutions of a 4-cycle engine. This curve is usually displayed continuously on a cathode-ray oscillograph. In contrast, point-by-point recording measures the cylinder pressure at one particular crank angle for each revolution of a 2-cycle engine, or each two revolutions of a 4-cycle engine. Consequently, it may require thousands of point-by-point measurements to delineate the same curve shape as was recorded by the instantaneous indicator.

In order to make a logical decision as to which of these alternatives to choose, the relative performance of contemporary instruments should be examined.

From a practical viewpoint, the accuracies obtainable when using an instantaneous indicator and a cathode-ray oscillograph are disappointing. If it is permissible to have only one auxiliary hole for pressure measurements in a given combustion-chamber head, then the indicator output must be interpreted in the light of previous calibrations. In the case of a good indicator used carefully, cumulative errors may easily reach values of 50 psi when using a 2000-psi maximum range instrument. Consequently, when used alone the instantaneous indicator is not well-suited to obtaining quantitative data.

On the other hand, pressure differences may be measured to within accuracies of 1%, if some known pressures can be indicated simultaneously with the output of the instantaneous indicator. For example, if two auxiliary holes are available in a given combustion chamber, both an instantaneous and a point-by-point indicator may be used simultaneously. If the point-by-point indicator back-

pressure is adjusted to known values, in effect a dynamic calibration point is obtained for the instantaneous indicator. The accuracy obtainable is now a function of the accuracy of the point-by-point indicator, which might possibly be of the order of $\pm$ 2 psi.

The question which immediately arises is: If the point-by-point measurements are so much more accurate, why use the instantaneous indicator at all? Figs. 1 and 2 illustrate the essential differences between the data obtained from the two systems. Fig. 1 illustrates engine cylinder pressure versus time for one engine cycle, showing rapid pressure variations occurring near peak pressures. This information was recorded in approximately 1/10 sec.

In contrast, Fig. 2 depicts a pressure versus crank angle diagram obtained with a balanced-pressure indicator. Depending upon the technique employed, such diagrams require from 5 to 30 min and represent hundreds of engine cycles. Consequently, cycle-to-cycle variation causes an appreciable scattering of individual points recorded. Since high-frequency pressure transients are not repetitive, it is obvious that such transients are lost when using the point-by-point technique. Such a loss is intolerable when studying detonation phenomena.

The essential differences between the instantaneous indicator and the point-by-point indicator then lead to the following conclusions:

A. The instantaneous indicator is desirable where rapid accumulation of qualitative data is wanted, or when rapid changes in pressure are to be measured, as in the presence of detonation.

B. Where precision is of greatest importance, and other factors of secondary interest, the point-by-point system should be used.

Which Particular Instrument? – Once the question of which technique to use has been decided, the user is confronted with the problem of choosing a particular instrument.

That the choice of suitable equipment is not an easy task may be illustrated by reference to the many commercial instruments from which to choose. There are continuously reading indicators suitable for displaying a curve of instantaneous cylinder pressure versus time on the screen of a cathode-ray oscillograph. These may be of such diverse types as piezoelectric, magnetic reluctance, magnetostrictive, mutual impedance, variable resistance, photoelectric, or variable capacitance. The general classification of point-by-point pressure indicators offers such intriguing subclassifications as intermittent sampling valves of internal needle or external poppet designs, and the balanced-pressure indicators of the contact-making, velocity, or displacement types. The balanced-pressure types may be purchased with either clamped or free diaphragms. All of these are available as recording or as just indicating. How, then, can an intelligent choice be made? Some criteria must be available which would describe the attributes desirable in a cylinder pressure recording instrument. Conversely, the characteristics of each particular instrument should be measurable against these criteria so that relative merits may be assessed.

In order to devise a criterion suitable for evaluation of this type of equipment, some appreciation of the environment in which the equipment must work is necessary. Of course, ideally, a pressure indicator would be affected by pressure alone. As a practical matter, however, complete independence of environment is almost impossible. Since internal-combustion engines are heat engines, the instrumentation should be unaffected by extremes in temperature. The instrumentation should faithfully measure rapid changes in pressure, and yet be unaffected by engine vibrations caused by explosive combustion. When corrosive combustion products are present, neither the life nor the reliability of the instrument should be affected. In spite of the ignition-system voltages and the magnetic fields near an engine, the instrumentation should remain unperturbed. Finally, engines are designed with little thought to providing space or conveniences for research tools; hence, overall dimensions may be of critical importance.

It might then be said that an ideal indicator should be easy to calibrate, and should faithfully measure pressure throughout the range to be found in the engine regardless of time, temperature, vibrations, corrosive gases, and electrical disturbances. Of course, the instrument should leave the engine unaffected by its presence.

Presumably, the most reliable method of making a choice of a particular instrument to use would be to devise engine and bench tests to measure each commercial instrument against the criteria listed. In most cases, however, such an investigation would require a much greater expenditure of time and energy than would be warranted by the proposed research program. Consequently, the user of

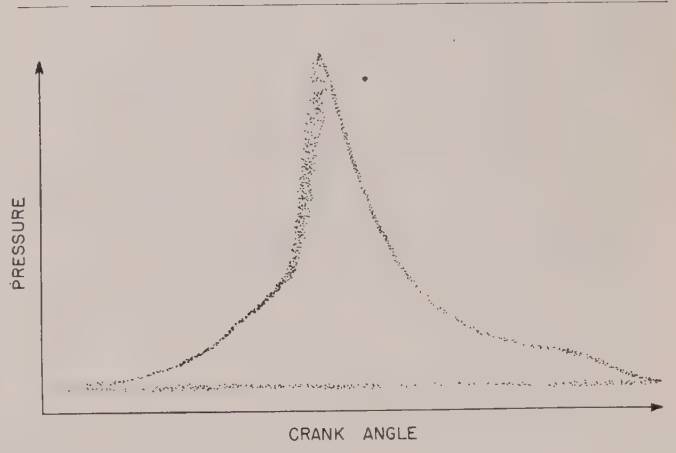


Fig. 2 – Point-by-point indicators require as many explosions as there are points recorded

Table 1 – Relative Performance Characteristics of Instantaneous Indicators

Type	Signal	Static Calibration	Frequency Response	Freedom from Vibration Error	Freedom from Temperature Error	Freedom from Electrical Interference Error	Reliability
Variable resistance (catenary type)	Pressure	Yes	Good	Good	Good	Good	Good
Variable capacitance (carrier type)	Pressure	Yes	Fair	Fair	Poor	Good	Good
Photoelectric	Pressure	Yes	Fair	Poor	Fair	Poor	?
Piezoelectric	Pressure	No	Fair	Poor	Poor	Poor	Fair
Magnetic (carrier)	Pressure	Yes	Poor	Fair	Poor	Fair	?
Magnetic reluctance	*DP/DT*	No	Fair	Poor	Fair	Good	Good
Magnetostrictive	*DP/DT*	No	Fair	Poor	Fair	Good	Good

such equipment is forced to rely on his common sense and the practical experience of others who have done such evaluation work. Some laboratories have felt the subject to be of sufficient importance to devote the time and effort required.

Continuous Indicators – The following examples will serve to illustrate the kind of testing procedures used to evaluate the performance of various continuously indicating pressure transducers.

1. Accuracy of calibration was determined by plotting indicator readings against known applied pressures as obtained from a deadweight type of tester. The resulting plot represented the static calibration. Plots obtained from increasing and decreasing pressures showed any hysteresis present. Dynamic calibration by means of rapidly varying pressures was obtained by operating the indicators in an engine under motoring and firing conditions. Errors were determined by comparing the pressures, as read on the indicator, with point-by-point data taken under favorable circumstances.

2. Freedom from spurious signals due to engine vibration was determined by measuring the indicator output signal when the indicator was subjected to engine vibration, but not to engine cylinder pressures. This was accomplished by first noting the indicator output signal in an engine under conditions of severe detonation. Then, leaving all other conditions the same, the indicator was removed from the engine cylinder head. A solid-bottom adapter was then inserted in the cylinder-head hole, and the indicator screwed into the adapter. Under these conditions, the solid adapter transmitted the cylinder-head vibrations to the indicator but the indicator diaphragm was exposed to atmospheric pressure only. Now, with the engine severely detonating, any indicator output could be regarded as false and due primarily to vibration response. The relative magnitude of the error due to vibration was indicated by comparing the false signals with the signal generated from the cylinder pressure plus vibration.

3. The effect of engine temperatures on the calibration of engine cylinder pressure indicators was studied qualitatively in an engine by noting the shift of the calibration ordinate as the engine warmed up from a cold start. Quantitative measurements of the effect of changes in cylinder-head temperatures were made by mounting various indicators in a dummy cylinder head. The cylinder head was mounted on a hot plate and the temperature noted by means of a spark-plug gasket-type thermocouple. Those indicators capable of static calibration showed a change in output even though the diaphragm pressure remained constant.

4. The effect of electrical accessories on the various indicators was observed by alternately operating and stopping each engine accessory. Relative values of modulation of the pressure curve were thus assigned to each electrical component.

5. Faithful response to high-frequency pressure transients is largely a function of the mechanical construction of the indicator. With the mechanical damping coefficients usually found in practice, it was reasonable to expect faithful response up to one-fourth the natural period of the mechanical system of the indicator. Since claims of manufacturers as to the natural period of individual indicators were difficult to check, a qualitative measure was obtained by mechanically shocking the indicator in the same direction as the pressure force. The resulting damped oscillation was photorecorded from an oscillograph. Frequencies were measured by comparison with known standards.

6. A qualitative measure of the utility of an indicator may be deduced from its overall dimensions. Other things being equal, a smaller indicator can be used on more engine styles than a larger one.

Coupling the results of these engine and bench tests with common sense, the ratings of Table 1 were compiled.

Point-by-Point Indicators – It has been inferred that best accuracies can be obtained by means of point-by-point pressure measurements. This is true only if certain conditions are met. An understanding of the essential nature of point-by-point pressure measurements will reveal what these conditional requirements might be.

Point-by-point pressure diagrams are obtained by measuring cylinder pressures in an intermittent fashion. A continuous curve joining these individual measurements then represents the plot of pressure versus crank angle for a given set of engine conditions. The equipment used is ordinarily considered simple to construct and to operate. However, in many cases, undeserved credence is given

point-by-point measurements because of the apparent, but unreal, simplicity of the equipment. To illustrate the nature of the difficulties encountered, let us examine these techniques more closely.

One of the first point-by-point techniques involved intermittent operation of an electrically actuated valve connecting the combustion chamber with a pressure gage. If the valve opened for a short time at the same crank angle each engine cycle, a steady pressure-gage reading would eventually result. The crank angle and the pressure reading were noted and a new crank angle selected and its pressure reading noted. By this means a curve of pressure versus crank angle could be drawn through the points thus obtained. Later developments produced an instrument for automatically recording this same information.

In spite of the apparent simplicity of this system, the fundamental assumption of pressure balance between the gage and cylinder during the time the valve is open, is not justified. Usual durations of valve open time range from one to two thousandths of a second. This represents a period of 6 deg at 1000 rpm or 18 deg at 3000 rpm. During combustion, rates of pressure rise of 50 psi per crank-angle deg are not uncommon. Thus, during the 6 to 18 deg that the valve is open, the cylinder pressure may change drastically. Consequently, a true balance of pressures cannot occur. It can readily be seen that two cylinder pressure diagrams could be identical when measured on such an arrangement, yet actually have widely different distribution of pressure dependence upon crank angle. For these and other mechanical reasons, such techniques have been abandoned by most research laboratories in favor of the balanced-pressure method. Fig. 3 depicts the essential elements of a balanced-pressure indicator.

The balanced-pressure indicator utilizes a comparison technique whereby a known pressure, as measured by precision gages, is compared to the pressure in the combustion chamber. A signal is given at the time these two pressures are equal. This action is usually accomplished by the motion of a compliant diaphragm separating the known and unknown pressures. As the diaphragm is forced against mechanical stops by the greater of the two pressures, it alternately makes and breaks a switch circuit. As the switch circuit is operated, some stroboscopic means is energized to indicate or record the crank angle at which pressure balance occurs. The known pressure is varied and the crank angle for balance read. A plot of pressure versus crank angle then yields the diagram desired.

Careful design and good practice can give results within ± 5 psi. However, as will be shown later, unrecognized errors may accumulate so that most results cannot be justified to better than 15 psi.

From this brief description of the operation of the balanced-pressure indicator, it could be concluded that a good design should exhibit the qualities of (a) low differential pressure necessary to cause an output signal; (b) independence of phenomena other than differential pressure; (c) good corrosion resistance; (d) ease of maintenance; and (e) reliability of operation.

Having selected the technique and the particular instrument to use, practical operating procedures must be then devised. Regardless of which method of obtaining data has been selected, continuous or point-by-point, poor auxiliary equipment or poor techniques can nullify the benefits of a good indicator.

Facts versus Fallacies

Facts often become confused with fallacies when applying instrumentation to an engine problem. Certain unjustifiable short cuts have become common practice, much to the detriment of test results. A few of these fallacies are easily exposed.

Fallacy No. 1 – "Standard plumbing practices may be used when connecting the indicator to the combustion chamber."

Fact No. 1 – Resorting to adapters or communicating passages of any kind penalizes the accuracy of the indicator. Accuracy is sacrificed by any arrangement other than locating the diaphragm flush with the combustion-chamber wall. Full exercise of ingenuity should be employed to get the indicator diaphragm flush with the chamber wall. If it is impossible to obtain flush mounting, the recorded data must be qualified by statements to the effect that certain characteristics are of doubtful nature. Figs. 4 through 7 illustrate the effect of the ratio of path length to diameter of a communicating

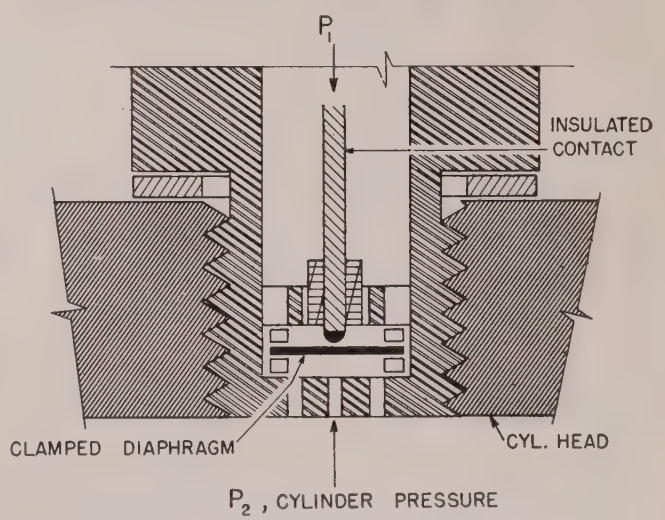


Fig. 3 – Typical construction of contact-making balanced-pressure indicator. Motion of center of diaphragm makes or breaks contact with center electrode, depending upon which pressure is greater

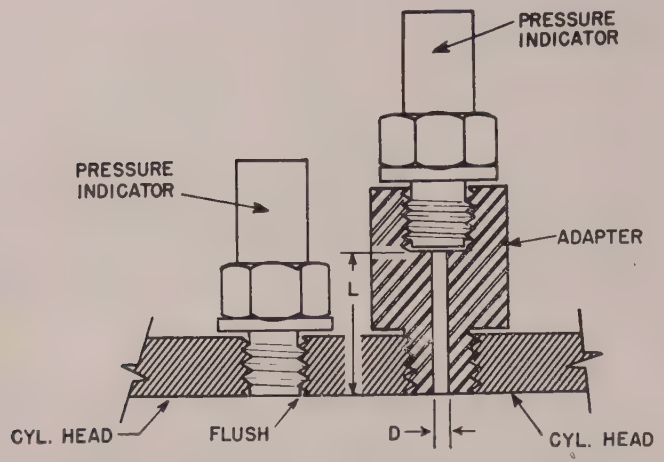


Fig. 4 – L-head engine fitted with two adjacent 18-mm threaded holes. Identical 18-mm indicators were mounted as shown. Simultaneous recording of one engine explosion showed effect of passageway of dimensions L and D

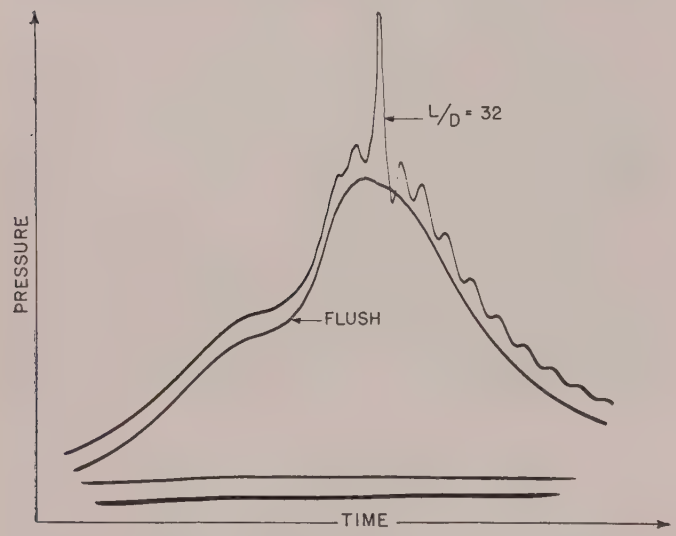


Fig. 5 – Simultaneous recordings of oscillograph traces showing errors introduced when indicator is connected by passage of length L and diameter D

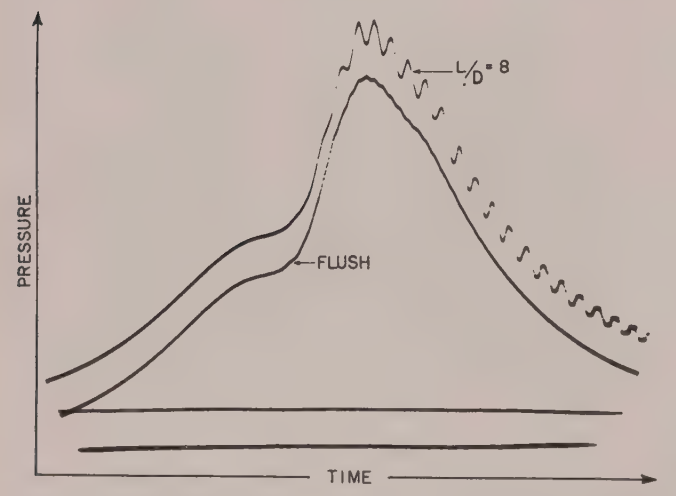


Fig. 6 – Errors introduced when indicator is connected by passage of length L and diameter D, as shown in simultaneous recordings of oscillograph traces

passage between the indicator and combustion chamber. These oscillograms depict simultaneous pressure-time records by identical indicators side-by-side in the same combustion chamber. The lower curve shows a flush-mounted indicator – the upper curve an indicator with an adapter of the stated length to diameter ratio.

Fallacy No. 2 – "Pressure versus time diagrams may be scaled directly from the oscillograph screen."

Fact No. 2 – Commercial oscilloscopes introduce such large errors that inches on the oscillograph screen may not represent pounds per square inch, or time, to accuracies better than 3 to 15%. Equipment manufacturers are continuously striving to produce error-free oscillographs, but the fact remains that for quantitative data either of the following alternatives must be employed.

Alternative A – This involves individual calibration of each component in the pressure-measuring system; that is, the indicator, the preamplifiers, the oscilloscope amplifiers, the cathode-ray tube deflection sensitivity at all points on the fluorescent screen, the camera distortion, and film shrinkage. Some overall transfer function must then be computed to relate inches of deflection on the oscilloscope screen to pounds per square inch pressure in the combustion chamber.

Alternative B – This may be called the interpolation or comparison technique whereby known quantities representing pressure and time (or crank angle) are recorded simultaneously with the pressure-time curve. Scaling these recorded data then becomes a process of interpolation between known values of pressure and time. Since the known and unknown data are recorded at the same time, it can be assumed that most phenomena affecting one affect the other. In order to record such data simultaneously, a multiple-channel oscilloscope may be required. Caution should be exercised to make certain that the traces of the multiple-channel oscillograph exhibit identical characteristics when the same signal is applied to the various channels. Particular care should be taken that coordinates of deflection are at right angles at each position on the fluorescent screen.

Fallacy No. 3 – "High-frequency 'wiggles' on the expansion portion of the diagram indicate the presence of detonation."

Fact No. 3 – High-frequency components of the pressure-time trace only indicate that something is vibrating. For example, some shock-excited mechanical or electrical system may have been caused to oscillate. Figs. 4 through 7 illustrated the effect of gas column vibrations. Mechanical vibrations of indicator parts, connectors, cables, oscilloscope table, and airborne vibrations coupled to the oscilloscope cabinet have all contributed to false wiggles on the pressure trace. Large errors may accumulate as each of these phenomenon contributes its part. While it is true that detonation may cause the combustion pressure to vary at the

indicator, it should be so interpreted only after every other possibility has been eliminated.

Fallacy No. 4 – "One good pressure curve is worth a thousand opinions."

Fact No. 4 – One good pressure-time curve may lead to costly policy decisions no matter how accurately it was made. Due to the large cycle-to-cycle variation in combustion pressures, the observer has no assurance that any one curve is representative of the majority of such curves obtainable. Fig. 8 illustrates approximately 10 consecutive cycles in a typical engine. Who is to say which of these 10 curves is typical? Enough data must be accumulated to furnish statistical support for any particular curve to be labeled as average or typical. Multiple-exposure photographic records such as Fig. 8 may delineate the extreme limits of maximum and minimum pressures reached, but such confusion exists near the typical curve as to be of no value. Many individual exposures statistically treated offer the only factual security.

Fallacy No. 5 – "Balanced-pressure indicators are practically self-calibrating."

Fact No. 5 – Large dynamic errors may easily occur when using a balanced-pressure indicator. The magnitude of these errors is not often recognized. Of course obvious precautions such as the use of leak-tight connections and accurate pressure-measuring equipment are usually observed. However, it is seldom realized that the overall accuracy can be drastically affected by such factors as differential operating pressure, contact point action, thermal bowing of the diaphragm, mounting of the indicator in the combustion chamber, recording time lags, and crank-angle alignment. As simple as it seems, the contact-making balanced-diaphragm indicator holds many pitfalls for the unwary. The usual arrangement is to apply dry nitrogen to one side of a compliant diaphragm, and engine cylinder pressure to the other side. Motion of the diaphragm between stops makes and breaks a contact in an electrical circuit causing some stroboscopic means to indicate or record pressures. If these contacts were to switch large currents, arcing at the contact points would cause metal transfer from one contact to the other, making a cratered instead of smooth surface. Since the maximum diaphragm motion is only a few thousandths of an inch, such cratered surfaces could easily nullify the initial contact spacing adjustment. Consequently, very minute currents are switched and electronic amplifiers employed to actuate the indicating or recording device. Although techniques have been devised to compensate for errors due to the recording devices, even larger errors may occur in the indicator before the contact mechanism does its work. Reference is made to the sequence of events preceding establishment of contact as engine cylinder pressure increases. In order to establish contact, a small amount of gas must flow through the lower set of holes, establish a force unbalance in favor of the cylinder pressure and accelerate the diaphragm from its lower stop to its upper stop. While the time required to accomplish this action is admittedly small, it is still significant when it is remembered that each millisecond is equivalent to 18 crank-angle deg at 3000 rpm and one crank-angle deg may represent from 50 to 100 psi pressure change. The time required just to

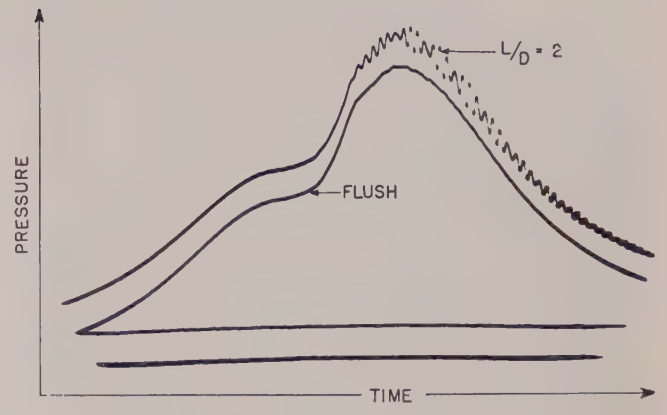


Fig. 7 – Another graph similar to those shown in Figs. 5 and 6. Simultaneous recordings of oscillograph traces showing errors introduced when indicator is connected by passage of length L and diameter D

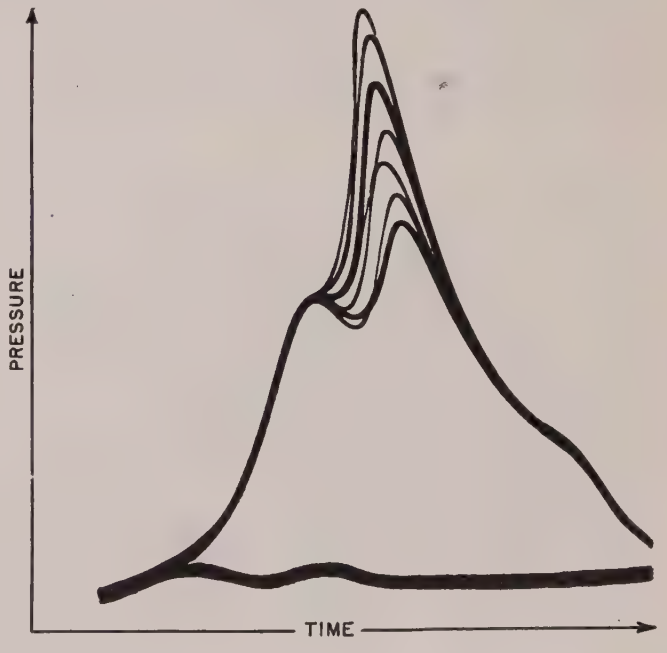


Fig. 8 – Variation of engine pressures during 10 consecutive cycles illustrates danger of basing conclusions on an analysis of one particular cycle

Fig. 9 – Experimental balanced-pressure indicator minimizes dynamic errors and is capable of indicating pressure differences greater than 0.06 in. of water

travel the few thousandths of an inch between stops has been measured to values greater than 3 deg, which can cause a significant error. In addition to this aforementioned error, change of the diaphragm characteristics due to thermal stress can introduce large errors. The mechanical forces generated can cause appreciable changes in the differential pressure required to establish contact. A balanced-pressure indicator design which offers a solution to some of these problems will be described in a later paragraph.

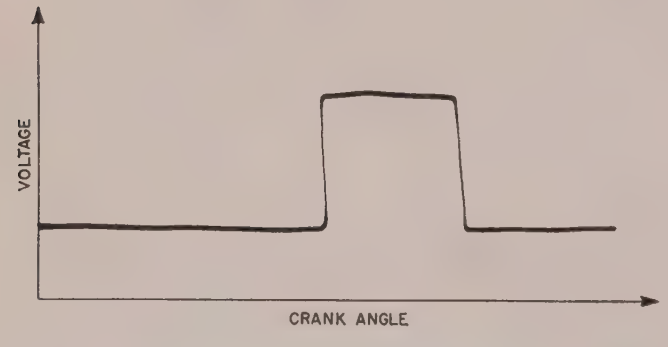


Fig. 10 – Oscillogram depicts balanced-pressure indicator diaphragm motion during one explosion. Total motion of diaphragm is 0.003 in.

New Developments – Recent developments in indicator equipment may serve to improve the accuracy and convenience of cylinder pressure measurements. The experimental designs to be described show enough promise to be of interest to those using such equipment.

One method of attack on the problem of time delays in balanced-pressure indicators is illustrated in Fig. 9. The objective of this design is to indicate the first instant of diaphragm motion, and thus to eliminate errors incurred by moving the diaphragm between its limiting stops. This objective is accomplished by using an unclamped diaphragm as one electrode of a variable capacitance in such a way that any motion of the diaphragm changes the capacitance of the system. An electronic circuit produces an output voltage proportional to the displacement of the diaphragm. The output voltage takes the form of a rectangular wave as shown in Fig. 10. This voltage may be used in any convenient form to actuate stroboscopic devices or to be displayed on an oscillograph. The most interesting performance characteristics of this indicator are its low differential operating pressure, 0.06 in. of water, and its quality of showing the first instant of diaphragm motion in either direction. The associated electronics consists of a 1-tube radio-frequency oscillator, link-coupled to the indicator so that the indicator may be used at a location remote from the oscillator. (The special oscillator circuit was used through the courtesy of Wesley Erwin of the General Motors laboratories.) Since there are no electrical contacts, cleaning may be very infrequent. However, the ease with which cleaning may be accomplished is illustrated in Fig. 11.

As mentioned previously, one of the difficulties facing the user of pressure-indicating equipment is the problem of providing another hole in the combustion-chamber head to accommodate the pressure transducer. Obtaining enough space to insert the usual indicator often taxes the user's ingenuity. Fig. 12 illustrates a strain-gage indicator of ½ in. maximum diameter requiring a 10-mm thread in the cylinder head. This indicator has made possible some measurements where other indicators could not be used. Some sacrifice in temperature immunity was accepted in order to keep the external dimensions small.

More convenient than this "pint-size" indicator is the combination of a spark plug and indicator described in Fig. 13. This ingenious indicator allows the cylinder pressure measurements to be made without requiring an auxiliary hole in the combustion chamber. A standard 14-mm spark plug thread permits immediate use in most automotive engines. This indicator is also of the strain-gage catenary-diaphragm family. Some of the design problems may be appreciated when it is realized that in addition to providing the functions of spark plug and indicator in the same space as

a spark plug, it also must shield the low-level strain-gage circuit from the immediately adjacent ignition voltages. That this is accomplished with fewer errors due to temperature and vibration than many commercial indicators is truly remarkable. However, it is still short of providing performance equal to the best of the single-purpose transducers. Further development of such an indicator could be extremely helpful to the industry.

Associated Apparatus – Once the factors involved in a particular type of engine research program have been weighed and a suitable choice of pressure-recording method and pressure transducer made, the auxiliary indicating, recording, and calibrating equipment must be assembled. If the user is to obtain quantitive data, it is imperative that the auxiliary equipment be of such quality as to fully exploit the potentialities of the indicator. Fig. 14 shows an integrated system designed to permit rapid recording of engine cylinder pressure diagrams. Since it was designed for use with a strain-gage type of engine indicator, it may also be used for other strain-gage applications in hydraulic, pneumatic, and mechanical force studies. It may be wheeled from room to room and quickly be made a part of the instrumentation for widely different kinds of tests.

At the upper right-hand panel may be seen the strain-gage bridge circuit with its balancing and calibration controls. The second panel from the top houses a variable-frequency audio oscillator for providing time reference marks on the cathode-ray oscillograph. The third panel from the top contains pulse-shaping circuits for introducing reference marks relating engine phenomena to the pressure diagram. Such reference marks might include known crank angles, the start of injection, ignition timing, and so forth. This panel also contains an electronic switch which permits time-sharing of one cathode-ray tube trace so as to effectively provide three channels of information to be recorded. This same panel also contains an oscillator circuit such as was described in the paragraph on balanced-pressure indicators. The lower panel houses an electronically regulated power supply providing direct current for the bridge circuit. The lower left-hand panel is a balanced wide-band d-c amplifier to multiply the bridge output voltage to a level suitable to operate the double-beam oscillograph mounted in the upper left-hand side of this portable console.

Groups of five plugs at each panel provide a patch-cord arrangement for flexible interconnection of the various components. Not shown are integral cooling fans and line-voltage stabilizers. More than 6000 individual pressure-time records have been obtained on such an instrument.

A companion device for the measurement of the known pressures used with the variable capacitance balanced-pressure indicator is shown in Fig. 15. Three sides of this cabinet house three precision pressure gages accurate to one part per

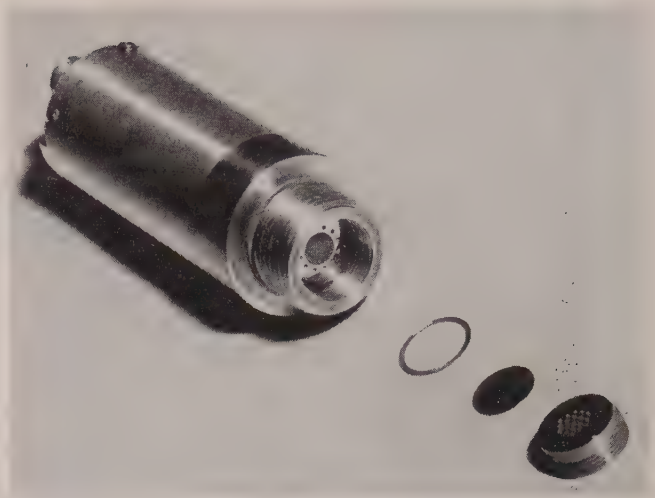


Fig. 11—Exploded view shows simplicity of construction and accessibility of vital parts of variable-capacitance balanced-pressure indicator

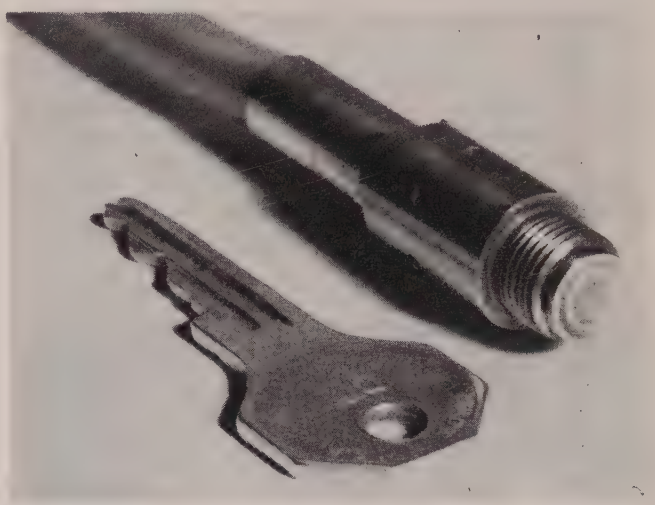


Fig. 12 – Ignition key shows relative size of this indicator. Maximum diameter is ½ in. Threaded end is 10 mm

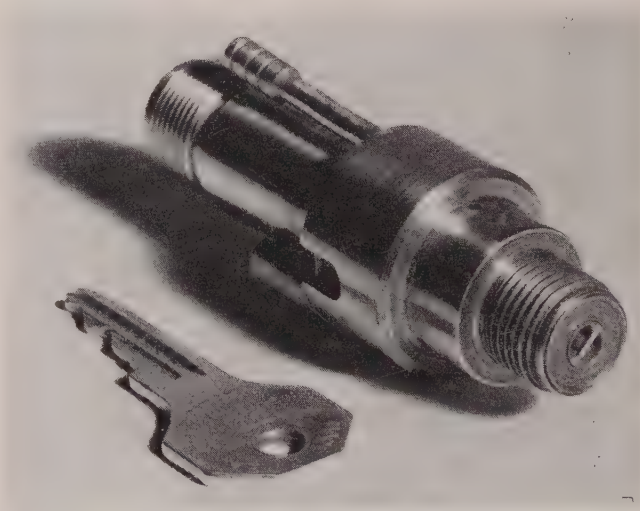


Fig. 13 – No auxiliary holes are required when using this indicator. It replaces standard 14-mm spark plug thus providing both ignition and cylinder-pressure measurements

Fig. 14 – Rack and panel design together with patch-cord connections permit flexible usage of components housed in this console. Use of a double-beam cathode-ray oscillograph permits simultaneous recording of cylinder pressure known crank angles, ignition timing, known frequencies, and known pressure calibration points

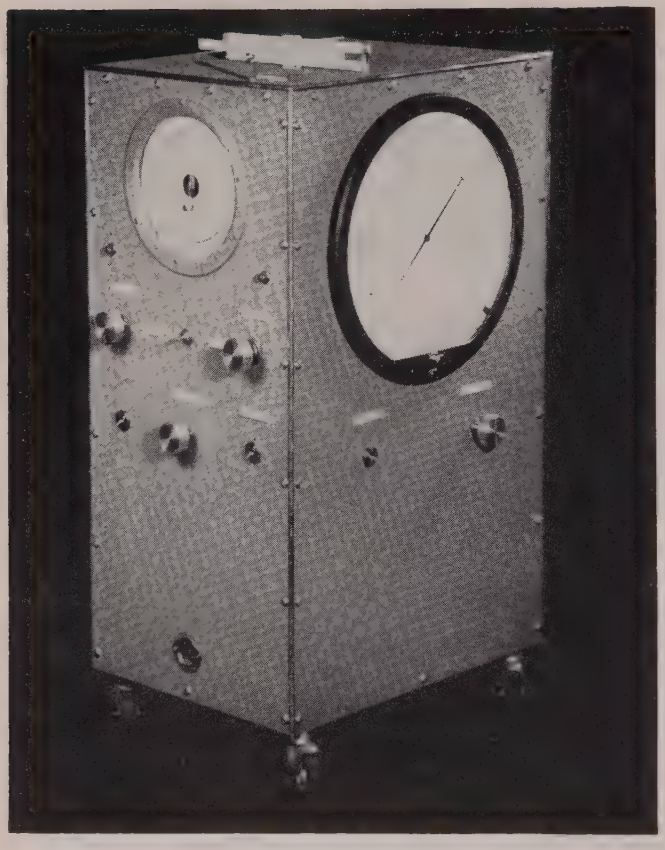


Fig. 15 – Engine cylinder pressures are read from these precision gages which measure known pressures applied to balanced-pressure indicators

thousand. A nitrogen tank supplies the source pressure, while an integral vacuum pump extends the operating range below atmospheric pressure. These instruments may be used together to present visual displays of engine cylinder pressures versus time. In order to make a permanent record of the data thus displayed, some sort of recording equipment is required.

Recording Data – Permanent records of engine cylinder pressures may be made in such varied ways as: photographic records of cathode-ray tube traces, sensitized paper records of magnetic oscillograph traces, spark records on engine-driven recording drums, and so forth. In each case, the means of recording has an end product suitable for storage of information subject to later analysis. The permanent records so obtained are usually filed on film or paper, both of which are subject to dimensional changes. Consequently, it becomes desirable to eliminate the possibility of erroneous data arising from the scaling constants changing with respect to time. Most workers attempt to include as much calibration data as possible on the record at the time the engine cylinder pressure was recorded. In this fashion the data may be scaled by interpolation between known values regardless of changes in the physical medium employed for recording.

Where pressure-time records are desired from the oscillograph screen, several practical techniques may be employed to improve the scaling accuracy and ease of recording data. Fig. 16 illustrates a recording camera designed to utilize these techniques. For ease of operation the fixed-focus camera is mounted above the oscillograph out of the way of the operator. This type of design permits mounting on the oscillograph in such a fashion that locking detents assure proper focusing. Daylight loading of the film is accomplished without disturbing the camera by using 12-exposure film packs. Film size was chosen as 4 by 5 in. to give images large enough for direct scaling without resorting to projections or enlargements. An F1.5 lens assures adequate recording speed. A means of identifying each particular film is essential to prevent confusion among many similar exposures. This feature is accomplished by exposing the numbers of a Veeder-Root counter directly on the emulsion. Pushing a button indexes two counters one digit and exposes the internal counter to the film. The external counter permits crossfiling the same number on the engine logbook. It is particularly difficult to scale oscillograms containing more than one pressure record due to crisscrossing of lines. Consequently, an effort has been made to assure that only one complete engine cycle appears on each film.

To be certain that only one engine cycle is displayed, the camera shutter must open at the beginning of each thermodynamic cycle and close at the end of that cycle. In order to center the most interesting portion of the pressure trace and

to record each thermodynamic cycle uniquely, the shutter should open at the bdc position of the engine, stay open for one complete cycle and close again at bdc. At high engine speeds it is almost impossible to synchronize mechanical shutters to the degree necessary. Consequently, an alternative arrangement has been employed. Such an alternative might be called an electronic shutter, since an electronic means is employed to perform the function of the usual mechanical shutter. Exposure is actually accomplished by a beam-blanking circuit which keeps the cathode-ray tube dark except for the instant desired for recording. When a picture is to be made, pushing a button causes the trace to brighten at the time the engine reaches bdc and to remain bright for two revolutions when using a four-cycle engine. By means of such equipment, large, easy-to-read negatives are obtained. Detail is sufficient so that standard full-sized duplicating means may be employed for illustrating reports, slides, and so forth.

Fig. 16 – Single explosion recording of pressure versus time is accomplished with this automatic camera and an electronic shutter

Summary

Assembling equipment to handle almost any type of engine cylinder-pressure measurement involves several different kinds of apparatus. Minimum requirements would be:

Instantaneous indicator.
Cathode-ray oscillograph.
Recording camera.
Source of known frequencies or crank angles.
Balanced-pressure indicator.
Precision-pressure source.
Stroboscopic indicator or recorder for balanced-pressure indicator.

To assemble a well-integrated system of such varied components requires the careful matching of the performance characteristics of such equipment as is commercially available. Thorough study of catalogue data and individual testing, if possible, is required. In order to guide the would-be users of such equipment, the factors influencing the choice of methods and components have been summarized. The requirements to be met by the instrumentation have been listed. Common fallacies have been exposed and alternative practices proposed. A hint of what may be available in the future has been included to stimulate further interest in this field. The road to successful application of pressure-recording techniques to development problems is not an easy one. Some guide posts have been pointed out herein and some of the dangers outlined. The importance of good equipment has been stressed and the value of sound techniques has been emphasized. However, the ultimate worth of any data rests in the hands of the person evaluating the results in the light of known errors and discrepancies. With intelligent and understanding usage of such equipment the industry is certain to gain more knowledge of the combustion process.

The author is deeply indebted to his coworkers for their helpful ideas and contributions, and particularly to Harold Stevenson, electrical engineer at Ethyl Corp.

References

1. *Automobile Engineer,* Vol. 41, September, 1951, pp. 341-343: "Engine Indicating – the C. A. V. – Ricardo Photoelectric Pressure and Movement Gage."

2. *The Engineer,* Vol. 186, Aug. 13, 1948, pp. 152-155; Aug. 20, 1948, pp. 178-179; Aug. 27, 1948, pp. 208-209: "Pressure Indicating for Internal-Combustion Engines," by R. Stansfield and J. G. Withers.

3. NACA Technical Note No. 1896, June, 1949: "Improvement of Accuracy of Balanced-Pressure Indicators and Development of an Indicator Calibrating Machine," by J. C. Livengood.

4. *Engineering,* Vol. 172, Dec. 28, 1951, pp. 806-810: "Obtaining Pressure Diagrams of Internal-Combustion Engines," by A. J. Cornish and W. P. Mansfield.

5. *Journal of American Society of Naval Engineers,* Vol. 61, November, 1949, pp. 967-974: "The Effect of Phase Errors in Time-Base Indicator Diagrams," by T. D. Walshaw.

6. *Journal of Aeronautical Sciences,* Vol. 16, October, 1949, pp. 593-610: "A New High-Performance Engine Indicator of the Strain-Gage Type," by C. S. Draper and Y. T. Li.

7. "Design of Strain-Gage Pressure Indicators," by Y. T. Li. Paper presented at Seventh National Instrument Conference and Exhibit of the Instrument Society of America, Sept. 8, 1952.

Discussion on following page——→

DISCUSSION

Effect of Gas Vibrations Shown on Pressure Trace

– P. S. Myers and O. A. Uyehara
University of Wisconsin

THIS paper serves the useful purpose of reminding all who, for one reason or another, are trying to measure rapidly varying pressures that, as usual, there are many pitfalls to be avoided and compromises to be made before useful pressure data are obtained. The paper presents an excellent discussion of these pitfalls and compromises, and should be of assistance in determining what equipment to use as well as in properly evaluating the data obtained. The new type of balanced diaphragm pickup that is described seems to offer definite advantages both in determining absolute pressures more accurately and in further evaluating the pitfalls and errors associated with a balanced diaphragm pressure pickup.

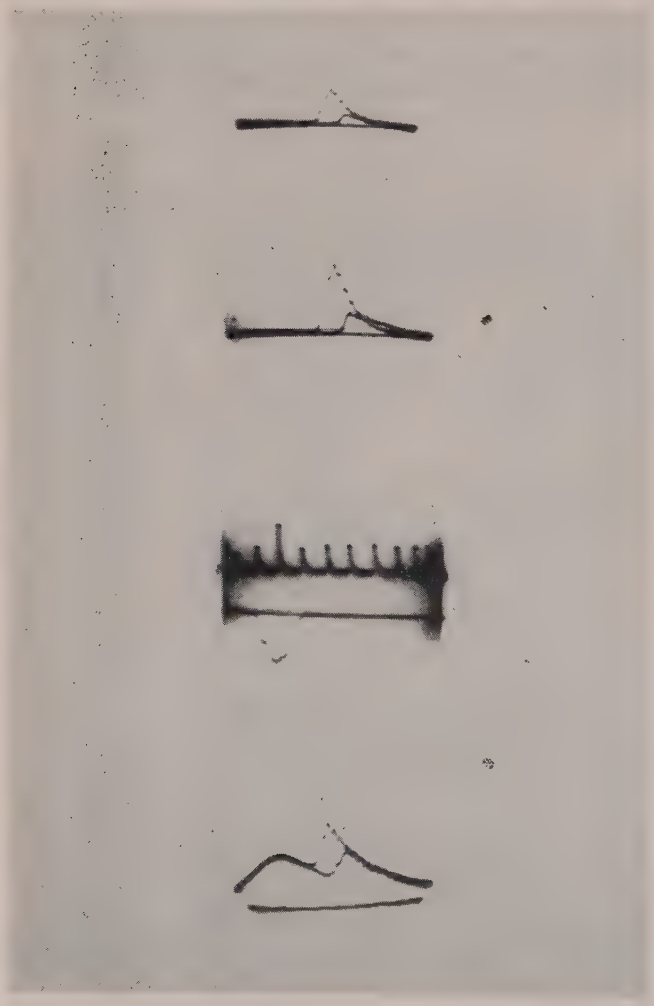


Fig. A – Oscillogram obtained from precombustion engine under severe operating conditions, showing fluctuations in both pressure and temperature traces

The fact versus fallacy section of the paper is well done. The conclusion that too often "wiggles" on the pressure trace are interpreted as gas vibrations without proper verification is concurred with by the discussors. There is experimental evidence, however, to show that these gas vibrations do exist. For example, Fig. A shows an oscillogram taken on a precombustion-chamber diesel engine under severe operating conditions. The two upper traces in Fig. A yield temperature values in the precombustion chamber. The third trace represents crank-position markers, the fourth trace represents opening of the injection nozzle, and the fifth trace is proportional to pressure in the precombustion chamber. It will be noted in Fig. A that both the temperature and pressure traces indicate gas vibrations of approximately the same frequency.

Speaking of gas vibrations the author presents data in the fact-fallacy section which presumably show the effect of the length/diameter ratio of a passage connecting the pressure pickup and combustion chamber. While the discussions are in thorough agreement that any connecting passage is undesirable, they are somewhat loath to attribute only to the L/D ratio (even by inference) the results shown. For example, the volume present at the pickup diaphragm, the connecting passage, and the combustion chamber, form two Helmboltz resonators with the connecting passage common to both resonators. In this case the volume of the space at the diaphragm of the connecting passage, or of the combustion chamber, might possibly affect the frequency of oscillation as well as the L/D ratio.

Several other pitfalls might be mentioned. For example, in a precombustion-chamber engine it obviously cannot be assumed that the pressure is uniform between the main and prechambers. Thus data obtained in the prechamber should not be used for calculations pertinent to the charges as a whole. Also when using photographic recording, the oscillographic trace width is of importance, particularly with small deflections. This trace width is affected by the choice of oscillograph tube and operating conditions, by the optical system used, and by the choice of film and its development. Another detail that requires attention is the correlation of piston position with the pressure measured. For example, if pressure rise rates of 50 psi per deg are present, and quantitative data are desired to ± 2 lb then the piston position data must be accurate to 0.08 of a crank-angle deg. Also if absolute pressure values are desired, the determination of the point in the cycle at which atmospheric pressure exists becomes of importance.

Figs. B, C, D, E, and F are presented to illustrate the effect of uncertainties that are present when using what might egotistically be called good technique. Fig. B presents an oscillogram taken on a 2-cycle open-chamber engine. Actually two oscillograms are shown – a hot motored cycle and a firing cycle. The two oscillograms were taken in sequence – the firing cycle first and the nonfiring cycle next, with a solenoid putting the pump rack to the no-fuel position in less than one engine revolution. A variable resistance (catenary type) pickup with a direct-coupled amplifier was used. To provide marks every 2 deg, 180 narrow radial slots were milled into the outer edge of a dish approximately 3 ft in diameter, and rigidly bolted to the flywheel. A special drum camera, rotated at engine speed by a pair of Selsyn motors and using one roll of No. 120 film, photographed the oscillographs and provided a time base. Horizontal alignment of the traces was assured by momentarily blanking out all oscillograph traces approximately 90 deg btc. Every attempt was made both electronically and photographically to obtain minimum line width. The resulting films were analyzed on a comparator which was capable of measuring to

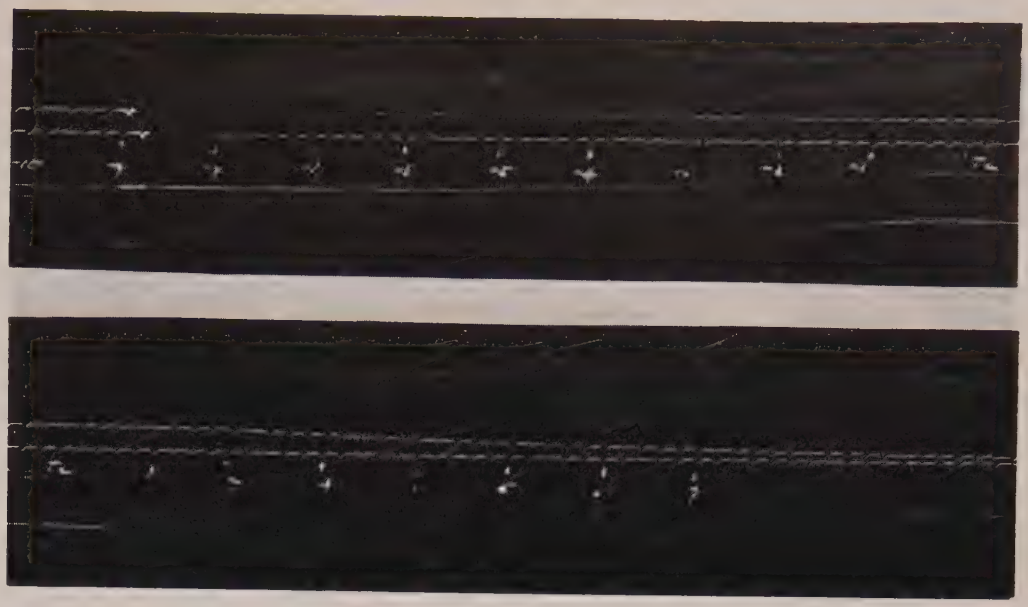


Fig. B – Typical oscillogram from open-chamber engine showing hot motored diagram and portion of fired diagram

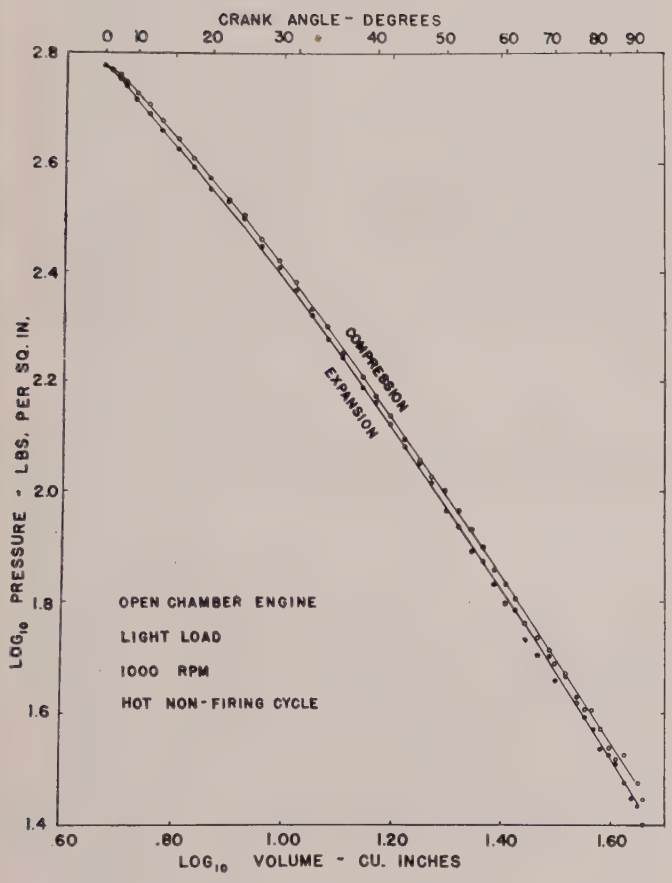


Fig. C – Logarithmic pressure-volume coordinates showing data as scaled from oscillogram in Fig. B

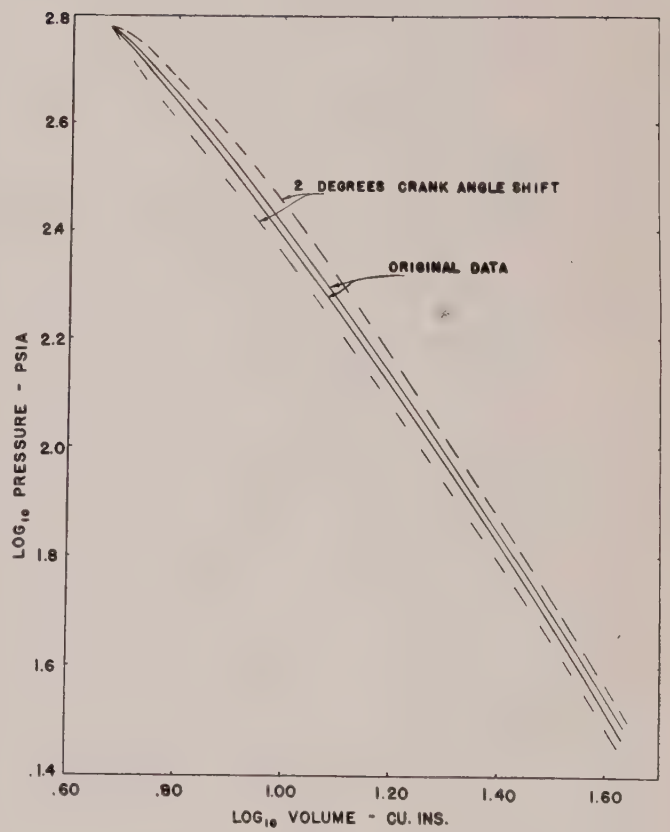


Fig. D – Diagram in Fig. C showing effect of shifting crank angle to indicate 2-deg error

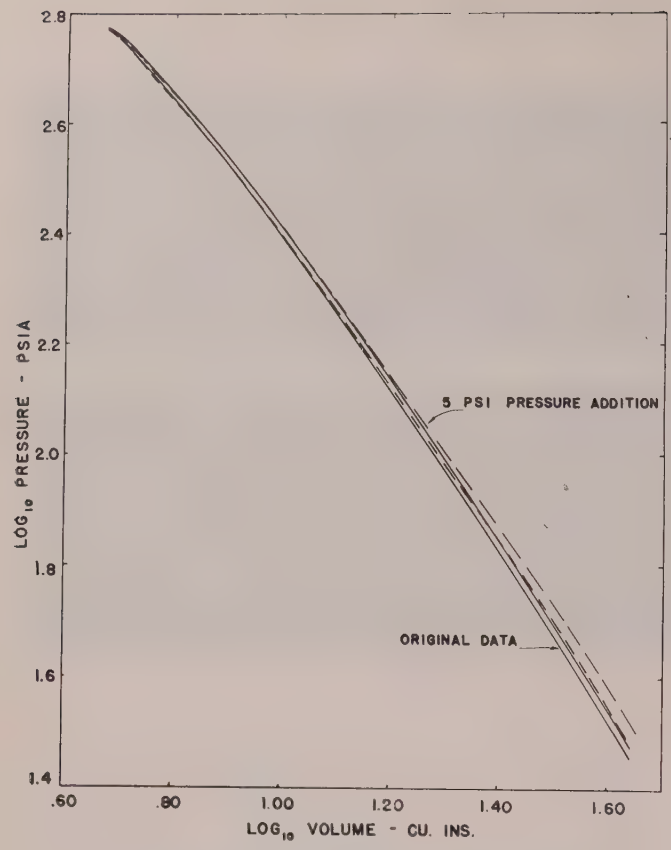


Fig. E – Diagram in Fig. C showing effect of 5-psi error

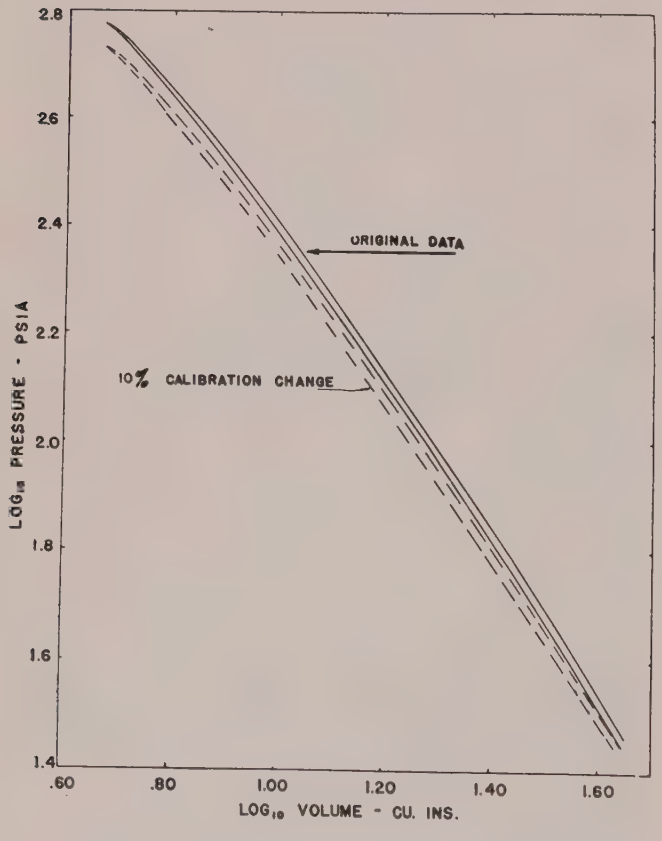


Fig. F – Diagram in Fig. D showing effect of 10% change in calibration pressure

0.0001 in. in the vertical direction and 0.001 in. in the horizontal direction. Using this instrument, scaling accuracy in both the horizontal and vertical direction was limited only by the ability to correctly judge the center of the trace on the film.

Fig. C presents, on logarithmic *PV* coordinates, the data as it was scaled from the film. While the information will not be presented here in detail, close examination of the slopes of the compression and expansion curves will reveal some inconsistencies. Accordingly, it was decided to determine the effect on the slope of possible errors in the pressure determination.

While the technique used enables relative values of crank angle to be determined to a few hundredths of a degree, the absolute value of the crank angle as referred to actual position of the top of the piston cannot be determined nearly so precisely. Accordingly, Fig. D compares the original diagram with the diagram that would have been obtained if a 2-deg error had been made in determining top center. The effect is surprisingly large and emphasizes the need for correct absolute values of crank angle.

When it is desired to obtain absolute pressure values, using an instantaneous pressure indicator, it is necessary to use one of the following procedures:

1. Determine the lowest pressure in the cycle, refer all measurements to this point, and add this pressure to each point.
2. Determine at what point in the cycle atmospheric pressure occurs, refer all measurements to this point, and add atmospheric pressure to each point.
3. Or, if using a direct-current amplifier, determine the location of the oscillograph trace when the engine it at rest and atmospheric pressure is present in the cylinder and, assuming the point does not change, refer all measurements to this point when the engine is running.

Since none of these procedures are free from possible error, Fig. E shows the effect if a 5-psi error were made in assigning a pressure value to the reference pressure. As would be expected, this effect is most pronounced at lower pressures.

Fig. F presents data showing the effect of a 10% error in the calibration of the pickup. Under these circumstances the largest error in psi occurs at the highest pressure. It is interesting to note that the 2-deg shift in crank angle (an error of 1 out of 180) had about as much effect as an error in calibration of 10 out of 100. It would seem that correct absolute values of crank angle would be of considerable importance in any pressure studies where precise data are desired.

Small Design Modifications Reflected in Performance

– Simon K. Chen
International Harvester Co.

IN the development of the modern high-speed engine, we are looking for small improvements in engine performance which reflect small design modifications. The author has presented an excellent paper describing methods of pressure-time measurement. However, I would like to point out some problems that are commonly encountered in pressure-time measurements.

Fig. G shows the different components or stages of the engine pressure-time measurement. Both the instrumentation or the operation of any one of these stages might introduce intolerable error, thus the accuracy of the final data depends on the accuracy of every single component of the whole process.

To obtain a useful and reliable pressure-time diagram, the faithful indication of the crank-angle signal is as important as that of the pressure signal. For example, if we are shooting for 1% accuracy out of a range of 2000 psi, the pressure measurement has to be accurate to 20 psi, and

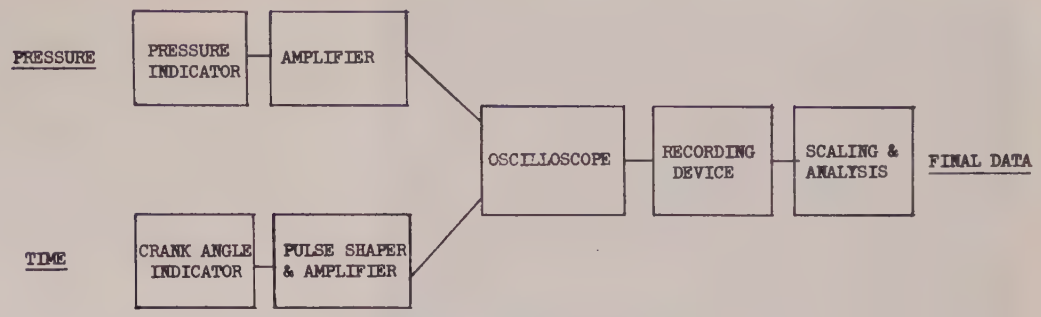


Fig. G – Components of pressure-time measurement

the crank-angle measurement has to be accurate to 4/10 deg crank-angle at the time axis, in assuming that the pressure rate is around 50 psi per degree. For instance, if the torsional vibration of the crankshaft goes as high as 1 deg at the free end of the shaft where the marker is connected, a maximum error of 50 psi will be introduced due to time measurement alone.

One of the commercially available crank-angle indicators is a magnetic pickup marker. In this particular installation with which we encountered trouble, a flexible coupling was used in connecting this marker unit to the engine crankshaft; at a certain speed we found that the out-of-phase of the marker is sometimes as large as 5 deg crank angle. This will shift the pressure curve 5 deg with respect to the time axis, or an error of about 150 psi at tdc, since the pressure rate is around 30 psi per deg.

To minimize an error of this sort, the marker should be rigidly connected to the crankshaft, assuming that the crankshaft is not subject to torsional vibration itself.

The paper has covered very well the different types of pressure indicators, problems of installation, and of calibration. This represents a tremendous amount of knowledge and hard work. The table of relative performance characteristics of instantaneous indicators is also very informative.

One other property of the pickup, although not listed in the table, which we found quite important, is the linearity of the pressure sensitivity of the indicator. A linear pickup will simplify tremendously the calibration and the analysis of the data. Unfortunately, some of the indicators on the market are not linear throughout the whole operating pressure range.

Fig. H shows the difference in linearity of the two types of indicators; the upper trace is obtained by using a strain-gage indicator while the lower one by using a capacitance indicator. It is noted that the capacitance pickup is not linear below around 200 psi for the particular size of diaphragm used, the strain-gage pickup is linear over the entire range.

Another item we would like to know more about is the indicator's temperature characteristics. If the indicator will hold its sensitivity over the range from room temperature to the operating temperature on the engine, the static calibration method as suggested by the strain-gage indicator manufacturer will be valid. I hope the author will tell us a little more about his temperature experiment. A curve of change in sensitivity, as well as a curve of zero drift against mounting temperature, would be quite useful.

Our farm tractor division has also performed some experimental work on the effect of passageways on pressure measurement. In addition to the pressure curve, they use a rate of pressure curve to illustrate the high-frequency oscillation on the pressure trace.

The upper curve in Fig. I is the pressure trace with the indicator mounted in a 4.5-in. passageway of a 3/32-in. hole. The lower curves are with a 6-in. passageway. Both of these curves are taken from a gasoline engine running at 1650 rpm. Also note the rate of pressure curve on the diagram.

Using three different lengths of passageway, that is, of

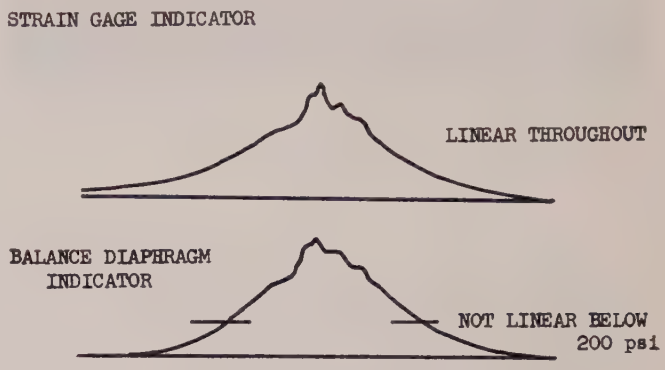


Fig. H – Difference in linearity in two types of indicators

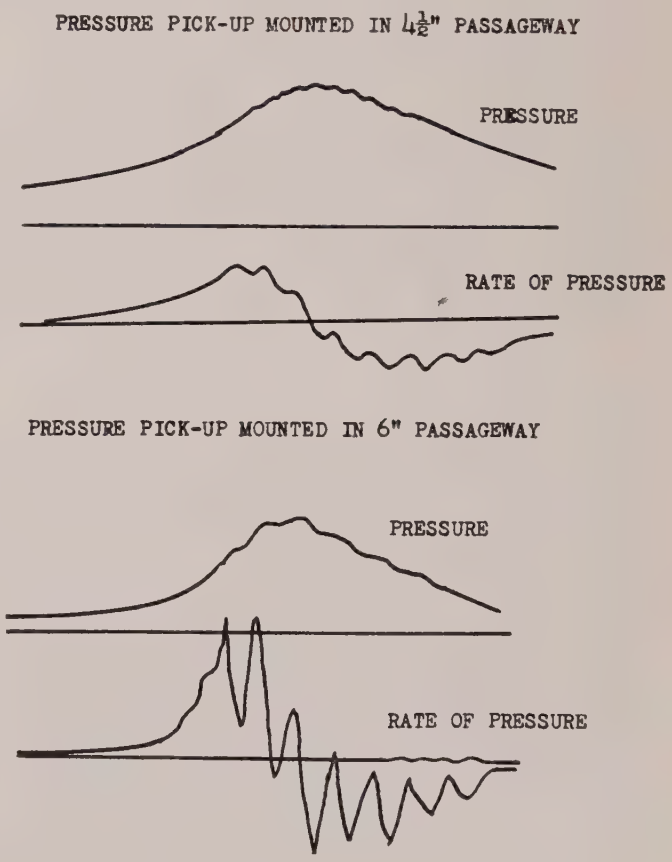


Fig. I – Effects recorded as result of different lengths of passageway

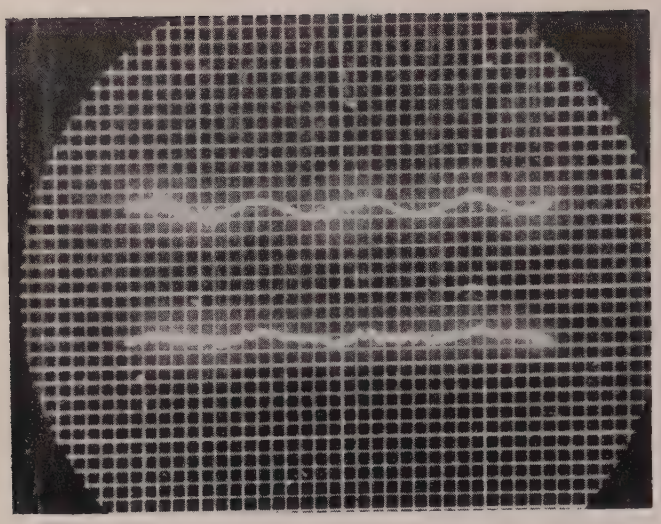


Fig. J – Distortion can be minimized by inserting illuminated calibrated scale on surface of scope.

2 in., 4½ in., and 6 in., respectively, it was found that the frequency of the oscillation is approximately inversely proportional to the length of the passageway. In the 2-in. passageway the frequency of the oscillation on the pressure trace is 2800 cps, in the 4½-in. passageway the frequency is 1600 cps, and on the 6-in. passageway, the frequency is around 1200 cps. These values, furthermore, do not change with a change in the engine rpm.

Using this frequency data, I roughly calculated the temperature of the medium in the passageway, in assuming the wave travels at sonic speed across the passageway; the result was around 3500 F maximum, which is also reasonable.

The paper has covered very well the selection of the amplifier and the oscilloscope, and the importance of electronics. The next component on my chart is the recording device. Mr. McCullough is perfectly right in favoring the use of a larger film. For practical purposes 4 x 5 film will give better resolution than a 35 mm film.

If even better accuracy is desired on the photographic technique, a drum camera or a moving-film camera should be used. Dr. Uyehara and Dr. Myers, at the University of Wisconsin, had used such a camera to obtain accurate measurements that cannot be obtained otherwise.

A properly designed drum camera seems to have the following advantages:

1. The y-axis distortion of the signal due to the cathode-ray tube, if any, is practically eliminated by the use of vertical axis amplifier of the scope only.

2. Time scale can be expanded to ¼-in. per 5 deg for a film strip 3 ft long, in comparison with about ¼-in. per 180 deg for the 35 mm film. The greatly expanded time base reduces the scaling error.

3. The drum rpm can be made to synchronize with the engine rpm.

4. Film of larger size can be used. Linagraph film of 2¼ in. has been used at the University of Wisconsin and there is no reason why 4-in. film cannot be used. A larger film will minimize the effect of trace width.

When an ordinary recording camera is used the distortion due to photographing, projection, or enlargement can be made minimum by inserting an illuminated calibrated scale on the surface of the scope, as shown in Fig. J. The distortion can be thus detected and be calibrated out. If there is any distortion it will show up on the calibrated scale.

The last step of the pressure measurement is the scaling and the analysis of the diagram. In using the ordinary method of scaling out the pressure trace by a divider and a scale, the error is sometimes objectionable, and different people will usually give different values. One of the reasons for error here is due to the inherent width of the trace. The width of the trace can represent some 20 psi on the 35 mm film, thus it is possible to introduce an error due to the inability to consistently locate the center of the lines for scaling. An additional error is caused by the scaling of the time axis.

To minimize the scaling error, an optical comparator capable of measuring one-tenth of a thousandth of one inch can be used. The instrument will eliminate most of the human error in scaling.

In using a drum camera of 4-in. film together with a comparator, the error due to the photographic recording and the scaling can be reduced to a minimum amount.

Mechanical Card-Drawing Indicators Discussed

– J. A. Stein
Bacharach Industrial Instrument Co.

THE title of this paper is much broader than is warranted by the contents. The author limits his discussion to electrical (omitting mechanical) pressure-time card-drawing (omitting nonrecording) laboratory (omitting field service) indicators for spark-ignition (omitting diesel) engines.

There are numerous nonrecording maximum pressure indicators on the market, which enjoy wide distribution. They are used chiefly (but not entirely) for maintenance checking of cylinder pressures.

The high-speed mechanical card-drawing engine indicator has a respectable history which goes back several decades. It is used in this country and abroad, in laboratories and in the field, and really merits an honorable mention. To give it less would seem an undeserved slight.

Those who are "contemplating engine cylinder-pressure measurements for the first time" (as stated in the digest) might well consider the simpler instruments just mentioned.

Passageway Effect on Indicator Reading Shown

– Frederick B. Kruger
Bacharach Industrial Instrument Co.

ONE of the highly commendable points of this paper is the work which has been done to show the effect of the connecting passageway on the indicator reading. However, due care should be used in interpreting or attempting to apply these curves to any specific installation, for there are several other factors which need to be considered since they also have an appreciable effect contributing to the resonating conditions depicted in Figs. 5, 6, and 7. Some of these factors are engine speeds, gas temperatures, and rates-of-pressure rise. It is this latter factor which, as much as any other, governs the proper selection of both the general type of indicator as well as the particular techniques to be used in engine cylinder-pressure measurements. Other aspects would also include such variables as the turns (which are the rule more than the exception on large engines) and the fittings which comprise the indicator-passage geometry.

Furthermore, it might be observed that the "novice to engine cylinder-pressure measurements" who elects to use electronic equipment to measure these pressures may find himself forced to learn simultaneously two highly technical skills. The use of electronic equipment to the extent of being familiar with and able to compensate for inherent equipment errors requires the services of a very able and skilled person; to ask this together with the application of this equipment in a field where there is no approved standard method of measurement seems to be a formidable assignment.

Seeks Further Details of Development and Test Work

– G. M. Rassweiler
Research Laboratories Division, GMC

THE story as presented in this paper is so simple and straightforward that it should be required reading for anyone using or developing engine pressure indicators.

I have followed Mr. McCullough's work for some years, so I appreciate the large amount of work he has done in this indicator field and how much more he could tell us if space permitted. In fact my only real criticism of his paper is that I wish he had somehow been able to include more detail of his development and test work. For example, I wish he had given us more specific information on his new condenser-type balanced-pressure unit, which appears very promising.

It would also be very helpful if we could see a great deal more information with regard to the table where the virtues of various instantaneous indicators are compared. Probably because of space limitations, this very important table is presented with essentially no experimental evidence to support the conclusions which it suggests. More test data would be very welcome and there are many other questions which naturally arise in one's mind with regard to this table. For example, do the ratings in the table refer to the pickup elements only, or do they include the associated electronic and the recording units? Were the tests limited to commercially available instruments? If so, were the commercial instruments modified by Ethyl and what were these modifications?

I would like to illustrate the necessity of being more specific in describing the indicators under test if the reader is to use the table as a guide for the selection of the best indicator for a given job. The table lists, as one type of indicator, a variable capacitance unit. Fig. K shows three records, all taken with capacitance pickups. The data in this figure were taken as part of a program to reduce temperature drift. Each curve was obtained by suddenly applying heat to an indicator, allowing its temperature to rise and observing the resulting change in the output signal. Time is represented on the horizontal axis and change in output on the vertical axis. These curves show that by making modifications in the design of this capacitance pickup the temperature drift may be radically changed. It may have a large positive value, it may go from negative to positive, or it may be reduced by suitable design changes to a very small amount. It is important to note that all three of these indicators were of the variable capacitance type and would thus fall into the same category in the author's table.

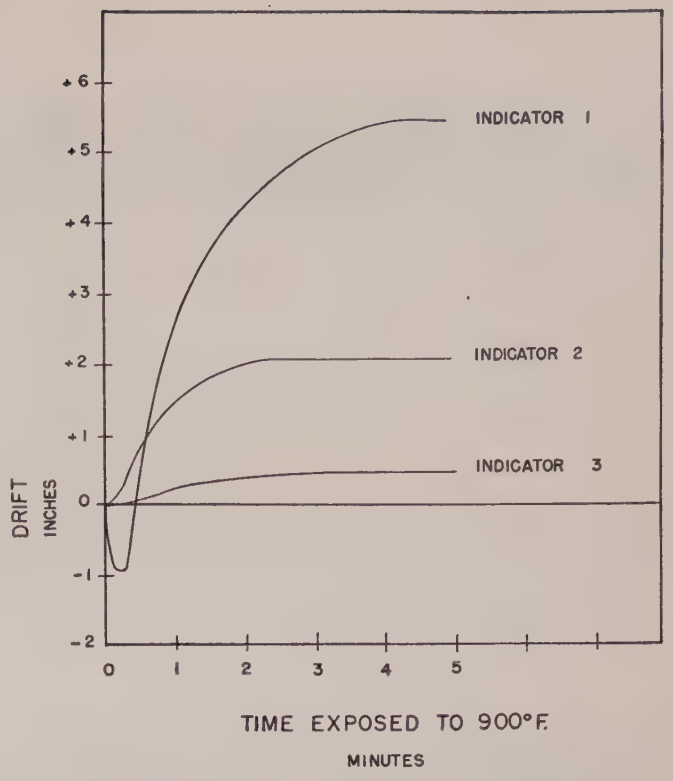


Fig. K – Three condenser indicators, differing only slightly in design details, show widely different temperature drifts

Fig. K thus shows that there are large variations between indicators in one of these categories. There are variable capacitance indicators much better (and some, I might add, are much worse) than the ones which the author probably tested. Very likely the same can be said of the other indicators in his table. So unless we have further information with regard to the indicators tested and the data obtained, we might actually be misled by his relative performance table.

My last point is with regard to recording instruments for engine indicator cards. Years ago, we commonly used the magnetic oscillograph almost entirely for our indicator work. Then the cathode-ray oscillograph, with its greater convenience and higher frequency response, became so attractive that we went over to this type. We agree with the author that great care must be exercised in the application of cathode-ray oscillographs to quantitative work.

Fig. L shows a record recently obtained on a 12-element magnetic oscillograph. The top eight traces are simultaneous records obtained with eight condenser indicators. These include records of combustion pressure from four cylinders together with four records from manifolds. We were also able to obtain on this same film a record of movement of one of the engine parts, a record of ignition events, and a timing trace taken from the crankshaft. The advantages of being able to take a large number of simultaneous records, together with the inherent stability of such an oscillograph, may sometimes overbalance its shortcomings from a frequency response standpoint. I wonder whether Mr. McCullough would give us his opinion of the utility of magnetic oscillographs for recording engine indicator cards, particularly in those cases where the study of knock is not the primary objective.

Finally, I want to say that any criticism I have made in no way reflects on the overall excellence of this paper. We hope that it will be possible for us to learn more about the details of the author's work at some future time.

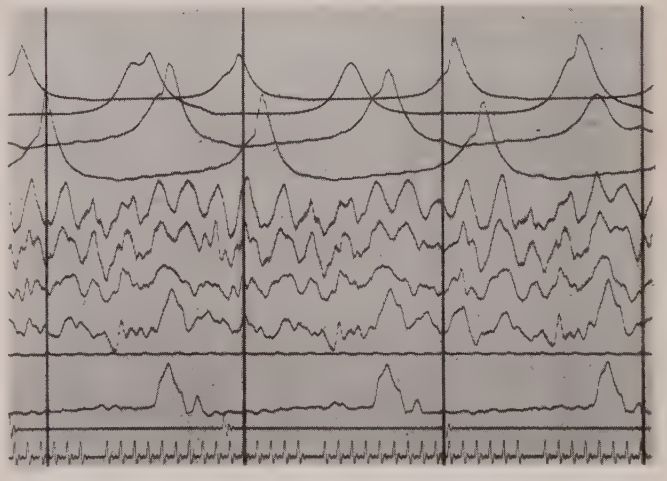


Fig. L – Simultaneous records on 12-element magnetic oscillograph illustrate its applicability to indicator work

Procedures Used in Development of

RECENTLY, we had the unique opportunity of developing an almost completely new type of diesel engine for the United States Air Force. When development was started the basic configurations and specifications were determined in a general way, but little technical know-how was available as to how they could be achieved.

Briefly, the Air Force wanted an extremely lightweight diesel engine which would be aircooled and able to burn fuels of varying cetane numbers, from ordinary diesel fuel through jet fuel, and ordinary gasoline up to 100-octane aviation gasoline. Although the initial intent was to develop a somewhat more general type, usable for all kinds of ground application, the engine, in its first embodiment, was to be used on an en-bloc-design air compressor requiring only one diesel cylinder. This somewhat simplified the development of the diesel engine by omitting problems introduced by multicylinder operation.

In order to have a multipurpose type of lightweight diesel engine available after completion of the development, the Air Force had restricted the possible approaches to this problem to port-controlled 2-cycle engines. They had, furthermore, specified that the scavenging system employed

DEVELOPMENT of a new type of diesel engine for the United States Air Force is described in this paper. Because of the very short time limits, special test stands were built so that development steps usually scheduled consecutively could be carried out simultaneously. This departure required many new approaches to normal testing practices.

The author reviews the unusual technical experience derived from this unique program. The development resulted in a diesel engine that met certain primary specifications – it was extremely lightweight, aircooled, and designed for use in any part of the world where jet fuel, aviation gasoline, or diesel fuel is available.

The Author

E. W. SPANNHAKE is now technical director of Barnes & Reinecke, Inc., after having served with the same organization as consulting engineer for diesel engines. Mr. Spannhake has both a B.S. and an M.S. degree from M. I. T. Prior to joining his present organization he was test engineer for diesel engines and transmissions at American Bosch Corp., American Locomotive Co., and General Machinery Corp.

Barnes & Reinecke Air Force Diesel Engine

E. W. Spannhake, Barnes & Reinecke, Inc.

This paper was presented at the SAE National Diesel-Engine Meeting, St. Louis, Oct. 31, 1952.

should give a symmetrical distribution of the inlet and outlet ports, to obtain full utilization from the lost stroke of the 2-cycle engine. They had also decided that a supercharged version of this diesel-engine principle should be developed, to take advantage of the lightweight features of such a configuration for the first application. Excerpts from the original specification as given by the Air Force read as follows:

"The complete unit including the intercoolers and aftercoolers shall be aircooled with provision to regulate the airflow to obtain proper operation within the temperature range specified.

"The diesel engine shall be of the 1-cyl 2-cycle type, of not less than 20 hp, having a piston displacement of approximately 40 cu in. The diesel exhaust shall be utilized to drive the cooling fan by means of an exhaust turbine.

"The diesel engine shall operate on any fuel from No. 2 distillate to 100-octane gasoline. The piston speed of the diesel engine shall not exceed 1200 fpm at 1500 rpm."

Together with study drawings supplied by the Air Force, the specifications outlined here resulted in a conception of the approximate shape of the diesel engine to be developed, shown in Fig. 1.

The problem then facing the engineers was basically this – a great number of untried technical features had to be combined into a working diesel-engine unit, and very little time was available for development. Therefore, an attempt was made to cut down the necessary development time and come up with a final working engine in the shortest possible interval, by intelligent planning of the course of development and anticipation of difficulties.

An analysis of the expected development difficulties led to the realization that the following basic steps would have to be taken if an engine as proposed by the Air Force could work successfully:

1. The development of a centrally symmetrical port-controlled scavenging principle with sufficient scavenging efficiency to make the unit operative.
2. The development of a suitable combustion chamber to accommodate the residual flows resulting from this scavenging system.
3. The development of an injection pattern which would suit this combustion chamber and make possible the highest mean effective pressures compatible with the trapped air charge inside the cylinder.
4. The development of an aircooling system which would, with reasonable air horsepower, carry away the thermal loads imposed on the different surfaces of the engine without detrimental effects to the critical parts.
5. The development of this engine for high supercharging pressures.
6. The development of a fuel-injection system capable of injecting gasoline.

The essential places at which these enumerated developments would have to occur are indicated by the same numerals shown in Fig. 1.

Basic Plan for Attack on Development Steps

It was immediately recognized that if all these development steps had to be taken chronologically, the chances for successfully fulfilling the development program in the required time would indeed be slim. The problem was therefore analyzed to determine whether they might possibly be attacked simultaneously. For this reason, they were divided into the following classifications:

1. *Independent Component Development Steps* – Those steps which could be developed in their

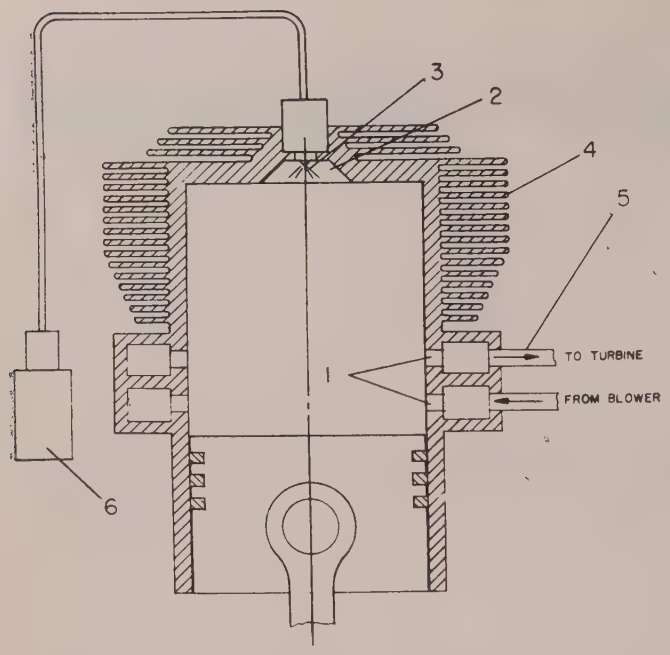


Fig. 1 – Basic concept of engine

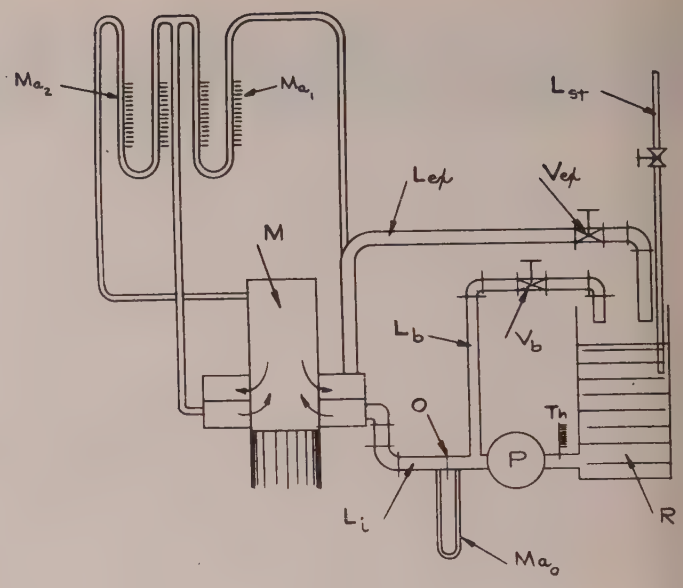


Fig. 2 – Schematic diagram of scavenging test setup

entirety outside of their connection with the engine before incorporation into the engine.

2. *Dependent Component Development Steps* – Those steps which could be developed partially without relation to the engine, although their only valid method of evaluation seemed to be actual tryout on the engine.

3. *Engine Development Steps* – Those steps which could only be tried and evaluated on the engine itself.

Independent Component Development – Looking over the development steps, it seemed to the engineers that enough knowledge of the behavior of the scavenging system could be obtained on a basic flow model so that the first step could be placed in the category of component development. For the same reason it was felt that the investigating technique for injection characteristics was far enough advanced so that a regular development program could be started on this feature without constant reference to engine tests. The sixth development step was therefore put into this category also.

Dependent Component Development – In deciding whether time could be saved by taking at least a part of a development step out of its direct connection with the engine, it seemed that there was an opportunity in the third development step – the injection pattern – to make use of the fact that the combustion in a chamber is to a certain extent a function of the penetration power of the fuel jets. It was therefore thought worth while to try to develop some methods of testing fuel penetration power which could be made outside of the engine.

Engine Development – For the remainder of the development steps it did not seem possible that any attempt at development apart from the engine would speed up the program. It was, therefore, decided to do all this development directly on the test engine.

This meant that the test engine had to be flexible enough to accomplish development on the three remaining steps – two, four, and five – in a reasonable time; also that fairly elaborate test stands would be built for the independent component development. For the dependent component development an auxiliary test facility was deemed sufficient.

Finally the main engine data for the test engine had to be chosen. They were determined by the expectations of performance, together with the compressor, and fixed in agreement with the Air Force at: design speed, 1500 rpm; bore and stroke, 3⅝ by 4¾ in.; supercharging pressure, 10 psi gage.

Description of Test Apparatus

1. *Scavenging Test Apparatus – Test Setup* – Scavenging is a periodic and highly nonstationary fluid phenomenon. Therefore, if more than very general qualitative data are to be obtained, it is necessary to observe in some way the development of the scavenging flow and the sweeping out of the burned charge through its cycle. Since the actual

duration of one scavenging cycle in 2-cycle engines running 1500 rpm is of the order of magnitude of 1/100 sec, any attempt to observe gas flow under these conditions will run into serious difficulties in measuring the extremely short phenomenon, and will thereby be expensive and tedious.

It was therefore decided to lengthen this time by use of a water analogue model which would measure the flow at the same Reynolds number, with the time factor made longer by approximately the ratio of 20/1. This is a mean ratio of the kinematic viscosities of air and water. Since the development will also take place on systems of low scavenging differential pressure, it can be anticipated that after the original blowdown through the exhaust ports, the actual scavenging cycle will occur at essentially low Mach numbers, and correspondingly, compressibility effects should only play a secondary role. A picture showing the test setup is given in Fig. 2.

A pump P takes water out of a reservoir R, and delivers it through the intake line L_i, into the transparent model M. From the model it is conducted through the exit line L_{ex} back into the reservoir R. A bypass line L_b in connection with the bypass valve V_b regulates the flow. This flow is measured in the calibrated orifice O by manometer Ma_o. The pressure level in the system is regulated by exit valve V_{ex} on the exit line. A steam line L_{st} in the reservoir is used to regulate the water temperature, which is measured by a thermometer Th at the exit of the reservoir. Pressure difference between the exit and the entrance of the manifold can be measured by manometer Ma_1. Cylinder pressure is measured against intake pressure by the manometer Ma_2.

Model – Care was taken to build the model M in such a way that changes in the main configuration influencing scavenging performance could be made with the least amount of time. This was done by designing the model according to a stack-up principle, in which it was made up of sections which were piloted to each other and held tightly together by a quickly disconnected clamping mechanism.

The sectioning favored those parts which could reasonably be expected to be subject to the greatest amount of development and change. In this particular case the interchangeable parts were the ported section in the cylinder, the shape of the piston crown, and the shape of the combustion chamber on top of the cylinder. Later, the ported section was subdivided into individual rings containing the exhaust and the intake ports. Furthermore, the exhaust and intake manifold shapes could be changed by inserting strips into a large square box of essentially parallel walls. A further refinement made it possible to move the position of the piston crown so as to simulate various positions of the piston during the scavenging cycle. A section through the model proper, thus developed,

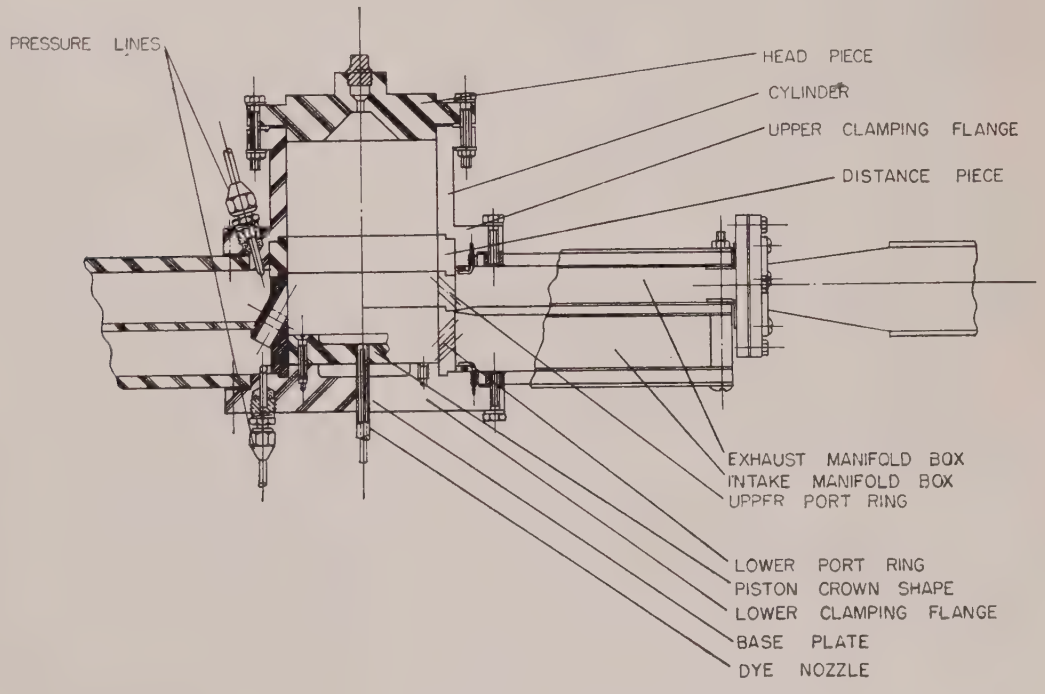


Fig. 3 – Section view of stack-up scavenging model

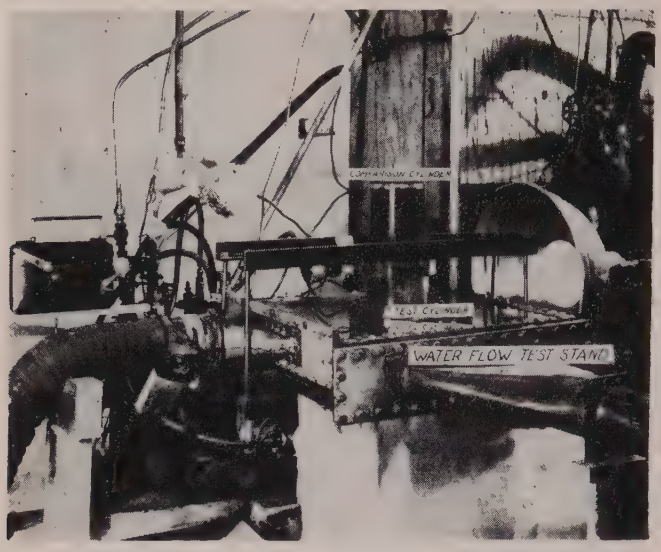


Fig. 4 – Front view of test section, scavenging test setup

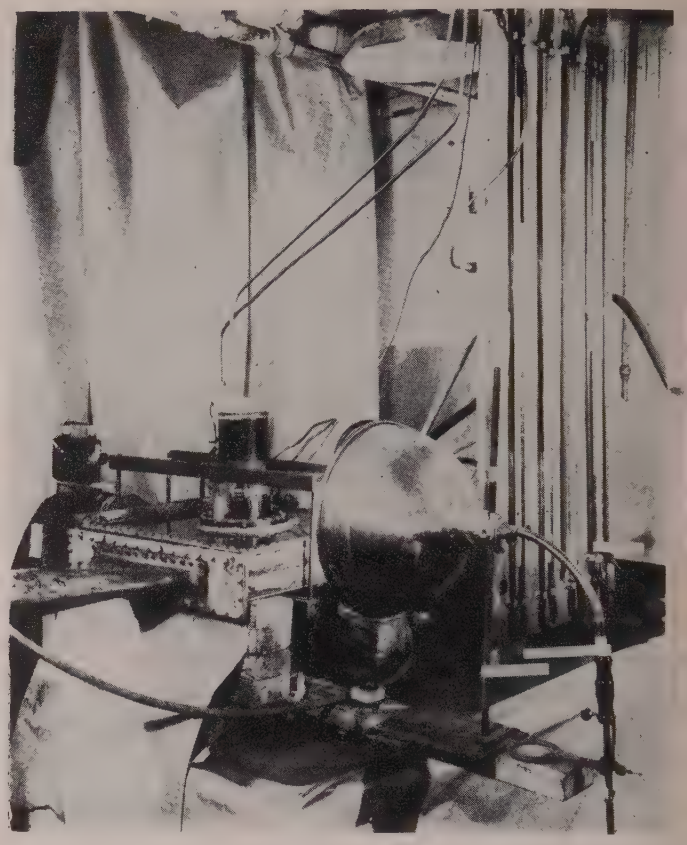


Fig. 5 – Rear view of test section, scavenging test setup

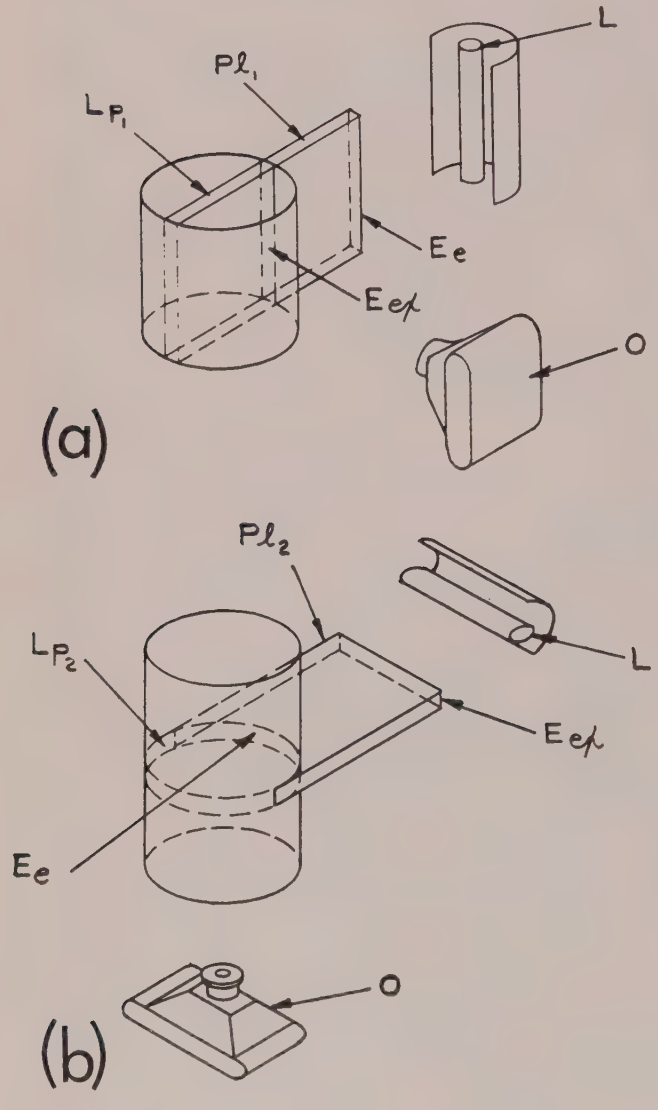


Fig. 6 – Schematic illustration of light plane method

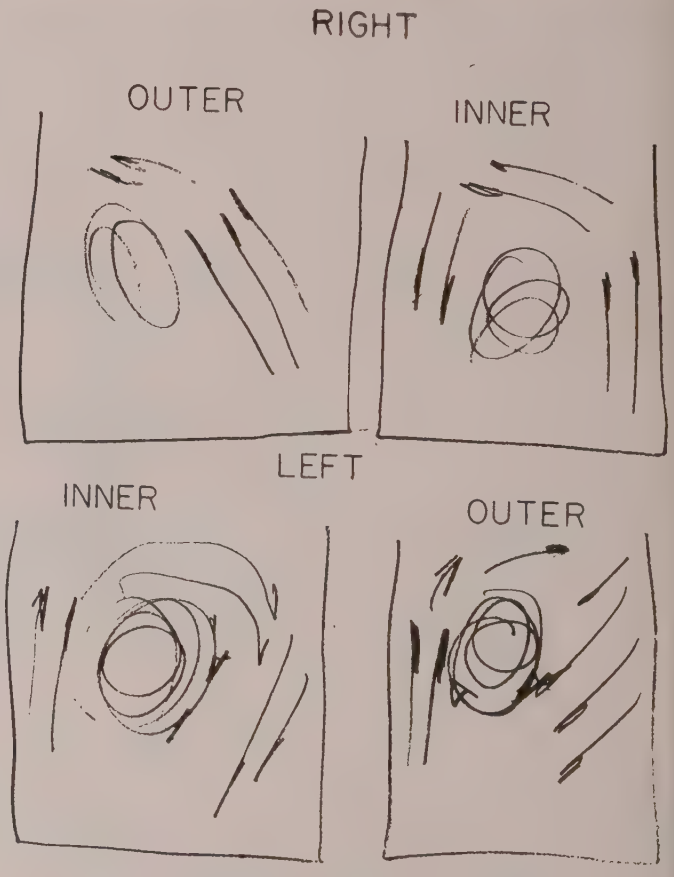


Fig. 7 – Typical sketch from light plane observation

is shown in Fig. 3. Figs. 4 and 5 show views of the model installed in the water-flow test stand.

Test Methods – All of the techniques used on this model were developed for stationary conditions of the main water flow. Although it is quite possible to use this stand for the investigation of nonsteady conditions, the flow was observed while the position of the piston and the scavenging pressure remained constant. It was found, however, that a good approximation of the scavenging behavior in the engine could be obtained by assuming that at each moment during the scavenging process, the flow pattern would be the same as the one which establishes itself as a stationary flow pattern with the piston unmoving. The characteristics of a scavenging cycle could therefore be calculated by a step-by-step integration of each of its momentary configurations. In any respect, a scavenging setup which, according to this method, had been found better than another one, actually would perform better on the engine to about the expected degree.

These stationary techniques can also be divided into two groups – qualitative and quantitative.

Qualitative Methods – Of the qualitative methods, the "light plane" method, which is illustrated in Fig. 6, was used extensively. The light source L throws its light into the entrance edge E_e of a lucite plate Pl_1. It travels through the lucite plate to the exit edge E_{ex}. Due to the length of the travel in the comparatively narrow plate the light comes out in a straight plane as indicated at Lp_1. An observer or a camera O will regard this plane at right angles. If the water is now mixed with particles having a light-refraction coefficient decidedly different from that of water, the flow of these particles can be seen very clearly. It was found that by using air bubbles an excellent observation check could be quickly obtained, showing whether the flow pattern performed as expected, or deviated from previous or expected performance. The direction of deviation could also be observed.

It might be mentioned here that for closer observations, especially if a flow-pattern evaluation should be made, fluids exist having the same specific gravity as water, but a different light-refraction coefficient. These fluids have previously been used with considerable success in other hydrodynamic laboratories for the illustration and evaluation of flow patterns otherwise difficult to measure.

For fairly complete visualization of the flow pattern, three cuts, parallel to the axis after the fashion of Fig. 6 (a), and another three cuts, at right angles to the former ones but still parallel to the axis, were sufficient. Three more cuts were usually made after the fashion of Fig. 6 (b), with light planes at right angles to the axis. Ordinarily, a test engineer observed these cuts visually and sketched on a block of paper his observation of the apparent flow pattern in each cut. This was done when the flow was stationary. Fig. 7 reproduces an actual sketch done by the test engineer. These sketches provided a satisfactory guide for analysis of the flow pattern in a given scavenging configuration.

Quantitative Methods – Dye Injection – Considerable time and thinking were spent in developing a method which could be used in quantitative comparison of different scavenging systems. The main difficulty was to be able to approach, with sufficient accuracy, the nonstationary phenomenon of sweep-out with a stationary main flow. The idea finally selected called for continuous dyeing of the contents of the cylinder until a certain determinable moment, after which clearing the dyed water out of the model could be observed visually. The dyeing was accomplished by continuous admixture of a small known percentage of a powerful dye with the water entering the intake ports. A schematic diagram of the apparatus used to achieve this is shown in Fig. 8. The model is indicated by the test cylinder C_t which is served by the water inlet through the intake manifold M_i and the intake ports P_i. Also shown are the outlet ports P_o which deliver the water through the outlet manifold M_o to the water outlet. Dye is injected at the entrance of the intake ports through the injection of ring R which receives the dye solution, through the injection line L_i. An alternate to this system is an injection nozzle N_i which injects dye at the bottom of the cylinder. Stop cocks C_1 and C_2 are used to connect either injection nozzle or injection ring, or both, to the injection line L_i. Dye is supplied to this system through flask Fl_1 containing the dye solution D. This is kept under a constant pressure through pressure line L_p coming from container Fl_2, inside of which a constant measurable pressure is established by some suitable means. The beginning and end of the injection takes place by electrically opening or shutting the dye valve V_d, which will admit dye into injection line L_i. Connected to the electric circuit effecting the closing of the dye valve V_d is an argon lamp A, which signals valve closure.

Considerable development was required to make lighting of the lamp A coincide with the actual stoppage of dye flow out of either ring R or nozzle N_i. It was found that two effects contributed to a discrepancy between the stoppage of this flow and the first electrical impulse sent into the dye valve V_d. One was inertia of the valve itself, which in the first tests placed the closing time in the order of magnitude up to 0.1 sec. The second source for error was found to be compressibility and inertia in the injection line L_i, which even after the closing of the dye valve caused some of the liquid to issue forth from the injection nozzles.

The first problem was finally solved by running a commercial valve with about five times the voltage for which it was designed and limiting, by a circuit, its exposure to the current, to little more than 2 sec. This reduced the closing time to within the order of magnitude of 1/100 sec.

The second of these errors was overcome by in-

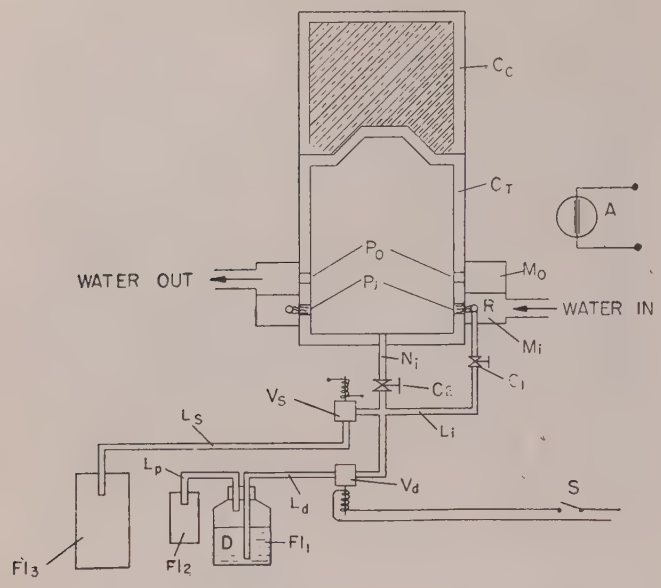


Fig. 8 – Schematic diagram of dye-injection apparatus

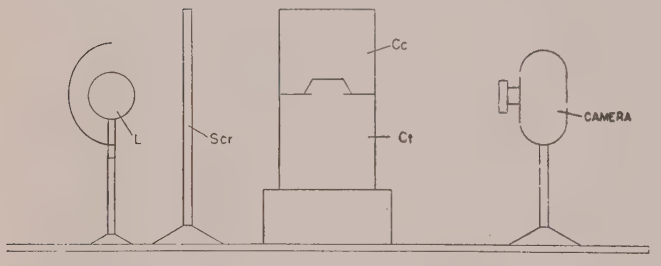


Fig. 9 – Lighting arrangement for dye-injection method

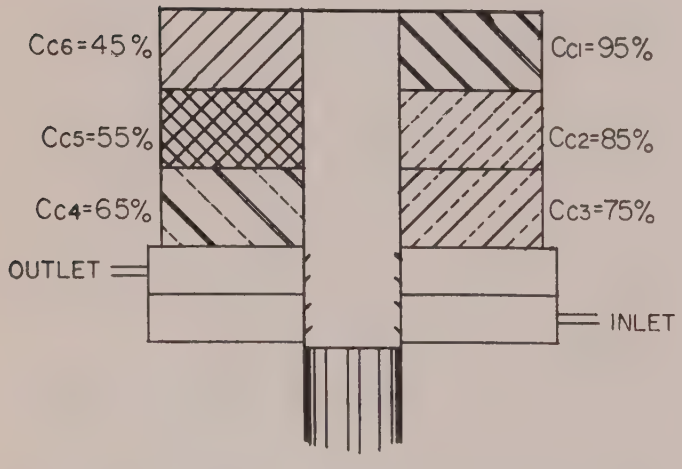


Fig. 10 – Schematic diagram of multicomparative engine. Percentages on comparison cylinder indicate various scavenging efficiencies

creasing the water pressure inside the model to about 20 psi, and making sure that at the moment the dye valve closed, a suction valve V_s connected the injection line L_i with a source of vacuum in flask Fl_3 through the suction line L_s. This, at the moment of closure, created such a powerful back-pressure in the injection line that the dye flow stopped almost instantly. Coordination between the flashing of argon lamp A and cutoff of the dye injection, was tested by putting the whole injection system in a rapidly moving stream of water and photographing the whole process with a high-speed motion picture camera. When it was certain that coordination was achieved, the system was installed in the cylinder and tests were begun.

The tests consisted of photographing the scavenging process by high-speed motion picture camera in transparent light. As the dyed water was swept out by the clear water which was now entering the ports, the dye concentration inside the cylinder was diluted and the light opacity was reduced. In the film the test cylinder appeared brighter as the scavenging process continued. The degree of opacity was a measure of how much of the original dyed water still remained within the cylinder.

In order to make this method quantitative, it is necessary to know what degree of scavenging corresponds to each opacity. To accomplish this, the so-called comparison cylinder C_c was placed on top of the test cylinder C_t in such a manner that both were lighted by the same source of transparent light. It was then a simple matter to pour a fluid of known dye concentration into the comparison cylinder. If the dye concentration of the test cylinder at the beginning of the scavenging cycle is known, the degree to which it is scavenged can be ascertained by comparing it with the concentration in the comparison cylinder. However, to determine the original concentration in the test cylinder, it is necessary to know both the rates of water flow and the rates of dye flow. The water flow can, of course, be calibrated by the manometer Ma_o described earlier in the description of the water test stand. The rate of dye flow was calibrated by weighing the dye flask Fl_1 between two time intervals, during which the conditions pertinent to the dye flow (the rate of water flow, the pressures in the cylinder and the manifolds, and the pressure on top of the dye flask) had been kept constant. Care had to be taken that in the actual scavenging test these conditions were known and remained the same as for the calibration. It was important that the source of transparent light be equal for the test cylinder and the comparison cylinder. This was assured by a method indicated in Fig. 9, where the light source L throws its light on a diffusing screen, Scr, from which it projects with equal intensity through test cylinder C_t and comparison cylinder C_c. Some experimentation and development, both as to the light source and the screen material, were necessary in order to

assure good results. Varying the position of the light source and the material of the screen were the methods employed.

Accuracy and degree of the transparent light were measured by determining the light strength in back of both comparison cylinder and test cylinder when filled with clear water, using a photometer. This was necessary, since some of the lighting methods showed a surprising degree of difference in the light transmitted through the comparison and the test cylinders. After these preparations the scavenging processes were photographed by high-speed motion picture camera. Time was recorded by simultaneously photographing a stop watch. Later, when this method proved inadequate for evaluating the fine points, an argon lamp flashed by a 60-cycle circuit was used, and proved much easier to interpret. The comparison cylinder in our tests was always set for scavenging down to 15% of the original contents. Furthermore, this dye concentration was chosen to give the best contrast on the film for the time exposure employed.

The time from the beginning of the scavenging cycle, indicated by the argon lamp, and the time when the equality of the two opacities showed achievement of 85% scavenging in the test cylinder, was measured. Basically, of course, any degree of scavenging can be measured by this method. We believe it may be possible to have six comparison opacities concurrently, as indicated in Fig. 10. Then an entire time history of degree of scavenging against time can be plotted from each test. This method was not used in the tests described because it was felt that most modern 2-cycle diesel engines operate in the neighborhood of 85% scavenging, and that the scavenging ratios necessary to achieve 85% efficiency would be indicative of the trend for other scavenging efficiencies. The tests were made with a conventional 16-mm movie camera, capable of running with the approximate speed of 60 frames per sec. Reasonably good comparison of scavenging procedures was possible, since the shortest duration achieved for a scavenging cycle was about 12 frames per sec. It is believed, however, that for somewhat closer study into the intricacies of scavenging procedures, a higher speed film should be used in such a way that 20 to 50 frames per sec should correspond approximately to one scavenging cycle.

Fig. 11 shows a film strip of one such test. For reasons of reproduction the consecutive order of pictures starts at the upper right-hand corner and continues to the left of each row, as indicated by the arrow. A 60-cycle timing argon lamp is arranged on the right-hand side of the cylinder. On the left-hand side, the argon lamp indicated by *A* in the previous schematic Fig. 8, showed through its lighting that the dye had been cut off. After the camera is started the operator presses a button and holds it, starting the dye flow. This is indicated by letter A in this picture. After assuring himself that

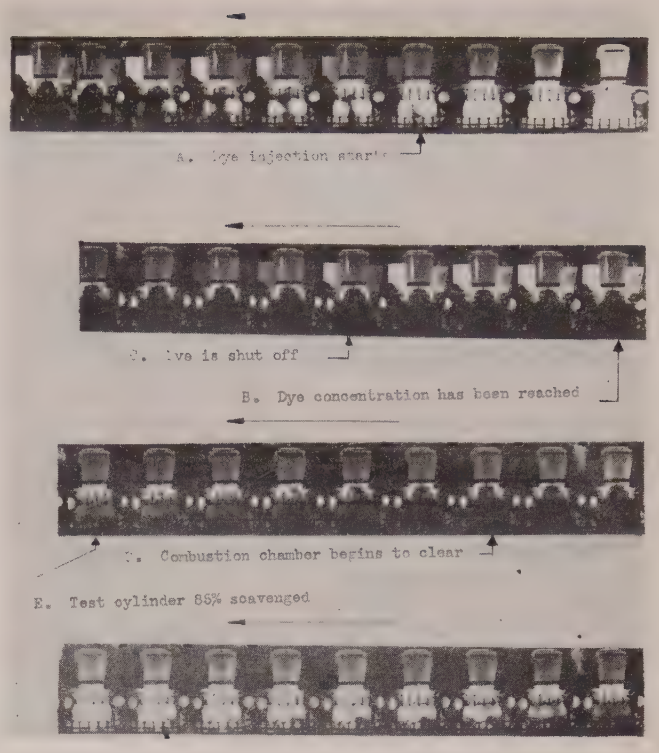


Fig. 11 – Sample motion-picture strip of water flow test showing good scavenging

conditions have become stationary (as shown in B) he releases the button which cuts off the dye as shown at C; this is indicated by the flashing of the lamp, and the scavenging process starts. It is interesting to note that due to the mushroom shape of the scavenging pattern the combustion chamber at the top of the cylinder clears first, as indicated at D. E finally shows the point at which opacity of the test cylinder is equal to the one of the comparison cylinder, and consequently 85% scavenging is reached. A schematic picture of the control circuits employed is given in Fig. 12.

In order to increase the ratio of the time scales, the kinematic viscosity of the water was further increased by heating, thus making possible a further decrease of water velocity for the same Reynolds number. Tests were always run at three Reynolds numbers, the middle one corresponding to the desired engine speed. Testing inaccuracies made it impossible to detect any appreciable influence of the Reynolds number on the scavenging systems. Even with this test procedure it was possible to complete the tests on one scavenging system in one-half hour, with two people running the test stand. One man handled the camera and dye-injection button; the other man took care of the water flow.

Pressure Tests – Since scavenging pressure differential is one of the important considerations in determining the suitability of the scavenging system, pressures in the intake and exhaust manifold

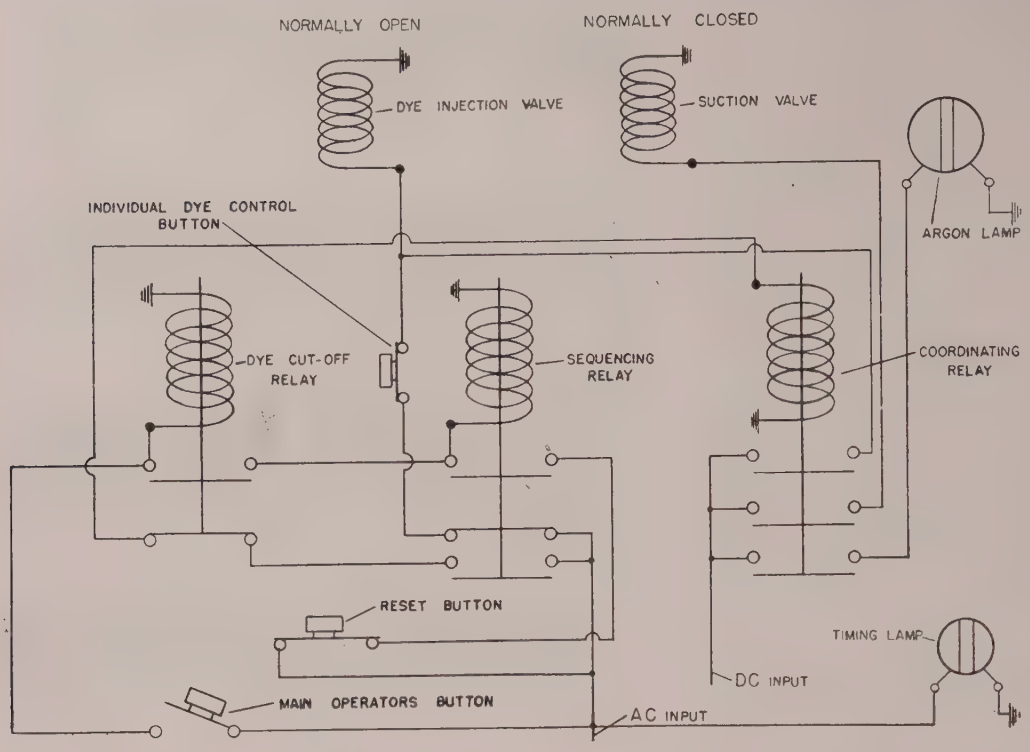


Fig. 12 – Electrical control circuit for dye-injection method

were taken rather carefully for all major scavenging systems investigated. Attempts to establish a mean pressure in the inside of the cylinder, to arrive at a subdivision of the total scavenging loss between intake and exhaust ports, did not prove entirely successful, as the cylinder pressure varied too much within the cylinder itself. This, of course, is due to the violent motion established by the jets within the cylinder, which did not leave any dead spots where a definable cylinder pressure could establish itself.

All scavenging systems showing promise were evaluated by this method for various positions of the piston crown, from its lowest up to almost complete coverage of the intake ports.

Injection Test Apparatus – Fairly elaborate injection tests were necessary because the system for multifuel injection to be developed on this engine was based on the theory, partially borne out by earlier experiments in Germany, that it would be possible to burn fuels of low cetane number successfully if the rate of fuel injection into the cylinder could be controlled in such a way that an initial period of low injection rate would be followed by one in which the remaining charge would be injected into the cylinder at a very much higher rate. Fig. 13 illustrates this point. Fig. 13 (a) shows the basic rate of injection as it appears for ordinary systems. At the beginning of injection, designated by the line *B-B*, a certain rate of injection establishes itself and stays basically constant until the end of the injection, indicated by line *E-E*. For an ordinary diesel fuel the beginning of the ignition as indicated by the line I_D-I_D will happen with a certain lag called the ignition lag. Present knowledge of combustion indicates that the moment a flame phenomenon begins in the cylinder all fuel present will burn, and any new fuel injected into the cylinder will burn almost instantly. For diesel fuels having short ignition lag, this will not lead to undue rise of pressure at the moment of ignition because the total amount of fuel present within the cylinder is small. If, however, the ignition lag, due to a low cetane number, is long – indicated by line I_G-I_G – the total amount of fuel present at the moment of ignition is so large that almost instantaneous combustion will cause bad knocks and pressure rises.

The solution attempted was to develop an injection system giving a rate of injection shown in Fig. 13 (b). Here a pattern which may be called a two-step injection is created. From the beginning of injection, indicated again by line *B-B*, to a crank angle indicated by line *L-L*, the rate of injection is extremely low. Between line *L-L* and the end of injection, then, the rate of injection is again what it would be for a normal diesel injection. If ordinary diesel fuel is injected now, ignition will take place, again at a line indicated by I_D-I_D, and will lead to a satisfactory combustion. However, if gas-

oline or any other fuel of low cetane velocity is injected, ignition will be delayed, as indicated by I_G-I_G, but now the total amount of fuel present in the cylinder at the time of ignition is so low that no bad knocking can occur.

Indications that this theory might lead to success had been confirmed by tests made in Germany, showing that a suction-valve-controlled pump having an injection characteristic similar to the one indicated in Fig. 13 (c) proved advantageous in using propane in a diesel cycle.

For this reason it was necessary to use test stands which could indicate not only the general spray pattern developed by the injection system, but also the rate of injection for each degree of the injection cycle. Fortunately, test stands capable of this were available in the industry, and use was made of them. It was, however, felt necessary – in order to make quick development advances and checks – to supplement these tests with others made on conventional test stands which, by stroboscopic observation of the spray pattern, gave a qualitative picture of rate with quantitative measurement of beginning and end of injection.

Penetration Test Apparatus – The penetration tests were made on a pressure chamber of the simplest kind. It is schematically indicated in Fig. 14. The interior of the pressure chamber *PC* is formed by the cup-shaped cylinder *C* which is closed at its bottom end by a cover plate, *CP*. Tightness is achieved by a gasket *G*. When the pressure chamber is in operation, the cover plate *CP* is forced against the cylinder by a jack *J*, which can be pumped upward by a handle H_j. The pressure of this jack is transferred back to the ground plate *GP* by tie-rods *TR*, which support the braces B_r from which the cylinder is suspended. The cylinder carries a pressure inlet *I*, which is connected to nitrogen flask *Fl* by a pressure line L_p. Pressure in this chamber can be adjusted by the pressure reducer *PR* on the nitrogen flask, and measured on the manometer *M* mounted there. Fuel is injected into the pressure chamber by nozzle *N*, which is mounted on the top of the chamber. It is serviced by the injection pump P_i through the injection line L_i.

Since during the development period emphasis was placed on the initial phase of fuel penetration before ignition, no attempt was made to simulate the speed of the injection stroke, but the injection pump was simply provided with actuating handle H_p, by which a single charge was injected into the combustion chamber. The elasticity of a manometer M_e was used to keep pop-off discharges through the nozzle to the same power and duration.

Measurement of the penetration power of the jet was done by impingement upon a measuring plate P_m. This was inserted into the combustion chamber and supported on the cover plate by variable length distance pieces, *D*, consisting essentially of piloted tube sections.

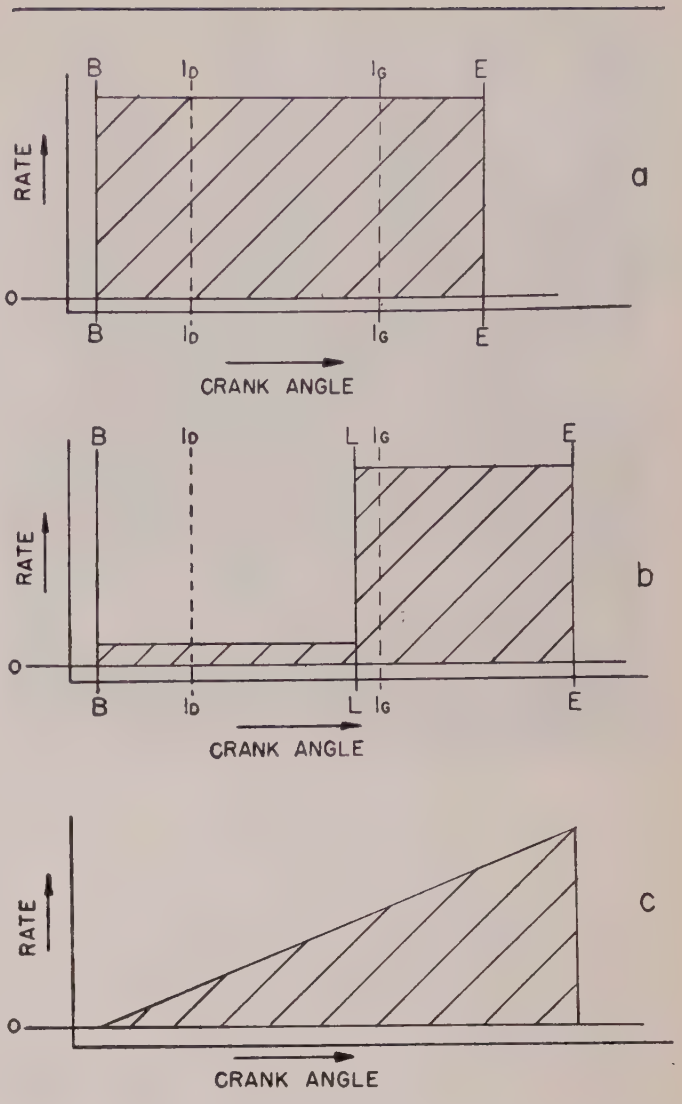


Fig. 13 – Schematic diagram of multifuel-injection system in comparison with standard system

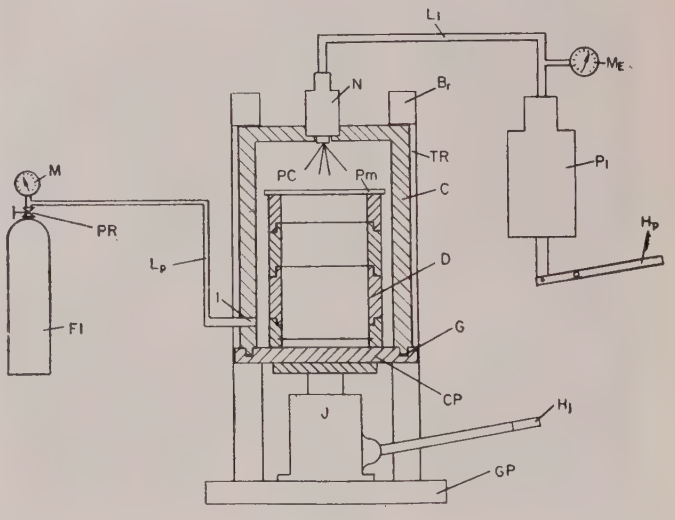


Fig. 14 – Schematic diagram of penetration test setup

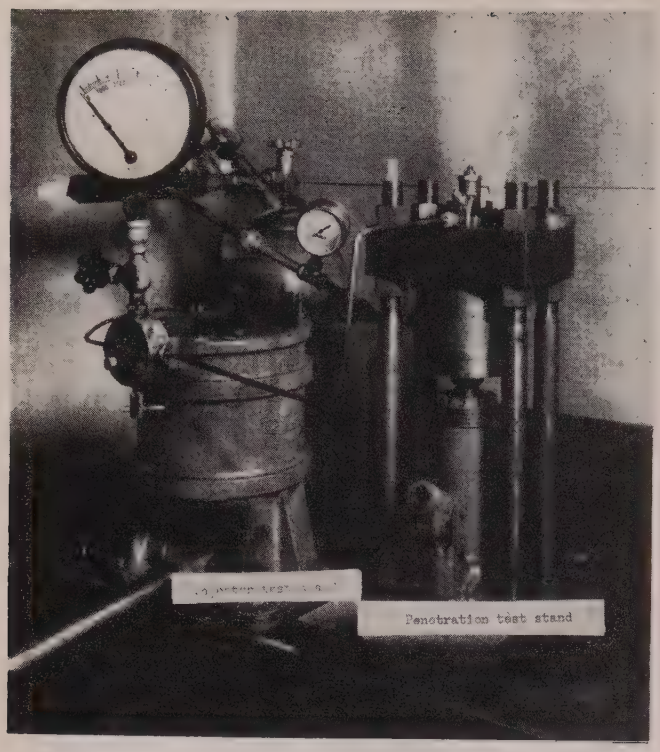


Fig. 15 – Penetration test stand

Various ways of measuring momentum of the jets were tried. Dyeing of the oil and impinging the jets on blotting paper did not prove too successful, inasmuch as the general fog in the combustion chamber tended to dye the paper and prevent any specific impingement places from being discernible. A method was needed, therefore, to measure very slight amounts of impingement momentum. Very loose cotton was tried, but it had a tendency to blow away even when placed upon wire mesh.

The most sensitive system, which was finally adopted, was to make up a fresh mixture of varnish and plaster of Paris to such a consistency that surface tension skins could be stretched between fine wire mesh when brushing on the liquid mixture. A test was immediately run in the pressure chamber after the preparation of one such plate. At those places where the jet still had a traceable momentum, the surface-tension skins would be punctured. If left undisturbed for a short while the remaining plaster would harden, preserving a record of the tests made. The utmost reach and penetration power of the jet was then assumed to be the distance from the nozzle orifice at which the jet was still able to puncture the surface-tension skin of the liquid plaster of Paris. This type of test could be made quickly and with little expense. Fig. 15 shows a picture of the penetration test stand, whereas Fig. 16 shows the measuring plate on top of the distance pieces with some penetration-test discs thus obtained.

Fig. 16 – Discs obtained from penetration test by use of prepared measuring plate

Engine Test Apparatus – It remains to be described how the design of a flexible test engine made possible the tests of a wide variety of components. This extreme flexibility needed was achieved by a stack-up principle, in which all parts of the engine were assembled in a building-block arrangement, and compressed on the assembly base by means of long tie-bolts. Fig. 17 illustrates this principle. A CUE test base was used, carrying a balanceable crankshaft. The assembly was built up on the mounting plate of this test base, starting with an assembly base. Upon this base rested a manifold block. The cylinder liner with its shrunk-on aircooling muff was supported on the block by a belleville spring. On top of the cylinder liner rested the cylinder head. A compression plate, made up of head distance piece and head proper, resting on the cylinder head proper, pressed all the parts tightly together by means of tie-bolts fastened to the assembly base. The compression plate also carried suitable means for mounting the injection nozzle. When compressed, the assembly could resist combustion pressures which sometimes went up to 1800 psi. Only the joint between the cylinder liner and the cylinder-head piece developed leaks, necessitating frequent relapping. This was overcome by the introduction of a copper gasket at the juncture. The combustion chamber proper was pressed into the cylinder head and held between the cylinder head proper and cylinder-head distance piece by the pressure of the tie-bolts. There was never any difficulty with this method of attachment.

The injection-pump drive with a timing device was mounted on the CUE test base in such a way as to establish a 2-piece linkage between the drive of the test base and the pump discharge nipple, thus assuring great flexibility in placement of the pump

nipple in relation to the cylinder head, as indicated in Fig. 18. This helped considerably when the height of the test engine assembly was changed or when, as sometimes happened, the assembly became wider.

Two other ways of increasing the flexibility of the test setup were used. The scavenging air and the cooling air supply were made entirely independent of the engine. Fig. 19 shows a schematic layout of the whole test engine stand. The scavenging air supply was taken out of a reserve tank, from which it ran through a pressure regulator and through a plenum chamber. After that it went through an air heater which could simulate any temperature up to 230 F. The quantity of air as well as all pertinent pressures could be measured.

A considerable range of supercharging or back-pressure values could be established by conducting the exhaust into exhaust plenum chamber 1, hence to exhaust plenum chamber 2, and from there through a manual exhaust throttle into the exhaust system of the laboratory.

In most cases, back pressure was created by closing the exhaust throttle. However, in cases such as investigation of coordination with a gas turbine, a nozzle could be inserted between the first and second exhaust plenum chambers, thus also creating the necessary back pressure.

The cooling air system is also shown in Fig. 19. It consisted of a blower and a large duct. Two headers, attached to the duct, supplied air to the engine. In each header the quantity and pressure of the air could be measured. One header went to the cylinder head, the other, to the liner.

Flexibility of this test stand was such that supercharging pressure, scavenging differential, and scavenging quantity could be changed continuously and quickly with the engine running. The same could be done with the cooling air.

Teardown inspections could be made and the engine put together again in as little as 6 hr. The simple exchange of one combustion-chamber shape for another could be made in about 4 hr. Exchanging one piston or piston crown for another took approximately 2 hr. The fact that the engine was air-cooled and therefore did not need gasketed water passages helped considerably.

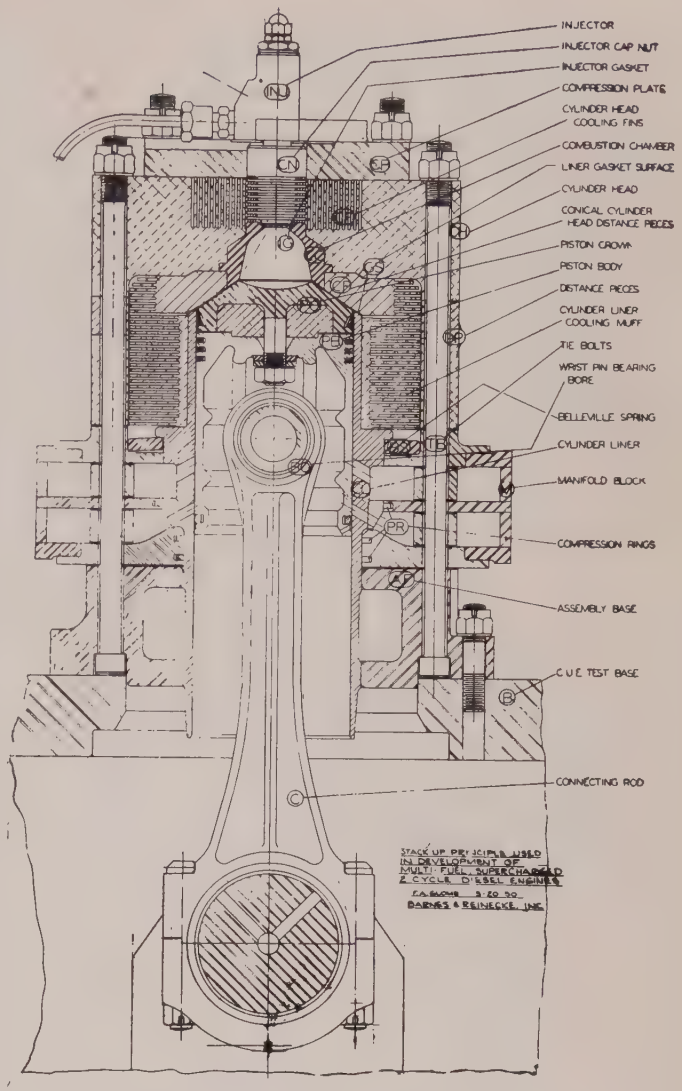


Fig. 17 – Cross-section through test engine

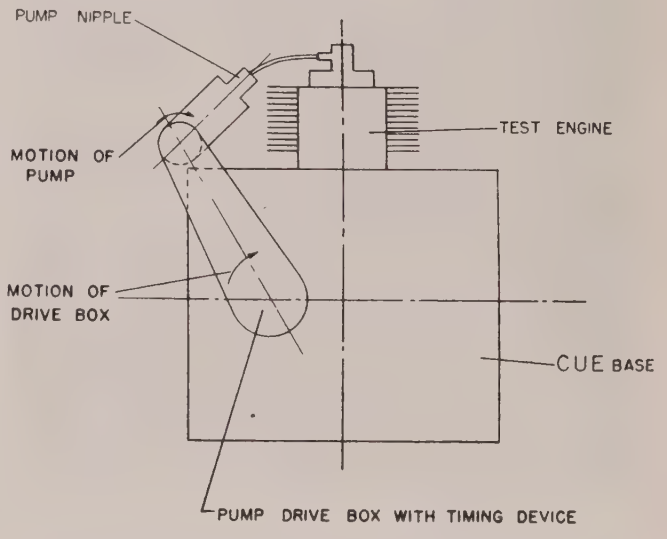


Fig. 18 – Injection-pump drive on test engine

Summary of Experience and Results Obtained

Scavenging Tests – By far the most difficult development turned out to be the attempts to create a centrally symmetrical mushroom-type flow of the kind that originally had been visualized by the Air Force. This may be illustrated by the fact that until a practical scavenging system had been developed, approximately 270 different configurations had to be investigated on the water analogue test stand. Here the flexibility of this stand showed up to best advantage and was the reason that so many configurations could be investigated within the

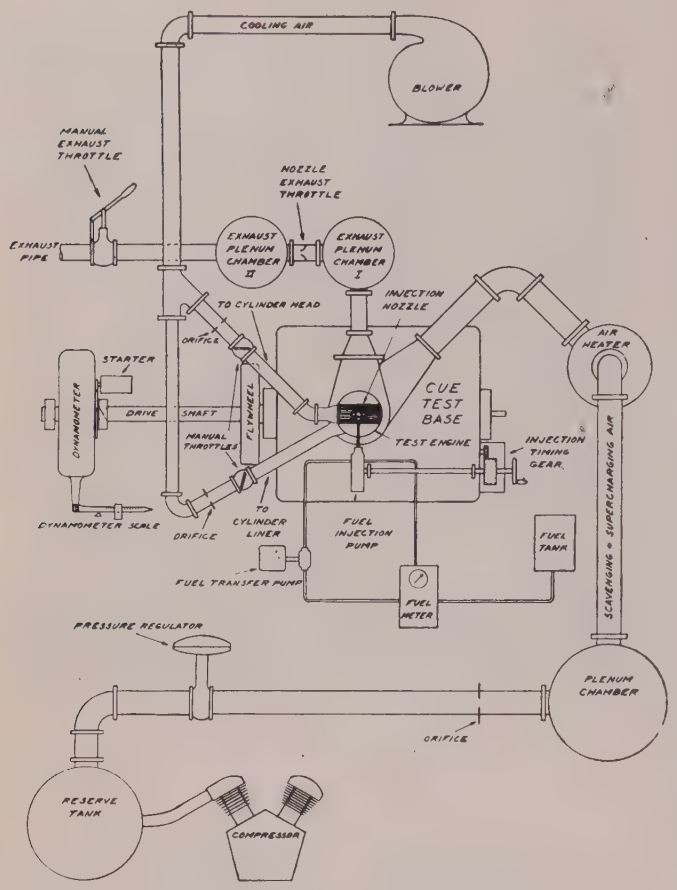


Fig. 19 – Schematic layout of engine test stand

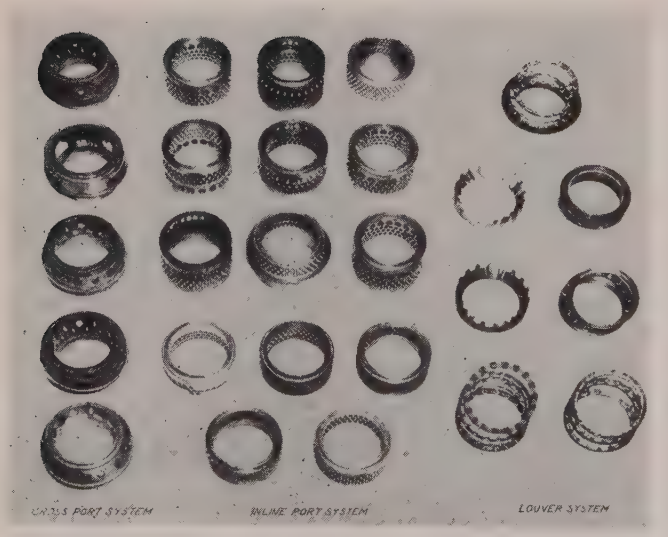


Fig. 20 – Some porting inserts used on scavenging test stand

time limits. New parts could be prepared while other tests were going on, and even major changeovers did not take more than a few hours.

Fig. 20 shows some of the porting inserts that could be stacked to form different porting systems.

Briefly, the investigation started with the simplest possible porting system of round holes drilled at right angles into the cylinder walls, as indicated in Fig. 21. This system showed very poor scavenging results. The flow had an overwhelming tendency to flow from the intake port directly into the most convenient exhaust port, and was thus prevented from penetrating appreciably into the cylinder space above the exhaust ports.

Considerable search for better scavenging performance resulted. It was conducted mainly by changing the angles of inlet and exhaust ports. Finally it was found that success could only be achieved if the intake port system would direct the stream downward against the piston crown, whereas the exhaust ports were directed upward. The system on which this was finally verified looked somewhat like Fig. 22.

This configuration was found by a combination of test and flow analysis. By the time the system mentioned above was tested, accompanying analysis had progressed to the point of establishing the belief that two conditions were necessary for the successful formation of a mushroom-type scavenging system:

1. The possibility of formation of a central upward flow column.

2. The necessity for the central column, once it was established and flowing in a mushroom-type scavenging pattern, to have a smaller hydrodynamic loss than a flow pattern which would short-cut directly from the intake ports into the exhaust ports.

For the first condition it was found to be necessary that the stream coming out of the intake ports would impinge upon the piston crown at a flat angle. The reason for this is believed to be that this will assure clinging of the jet to the piston crown until it has reached the center. It is, of course, well known that a jet which clings to a fixed wall will lose much less kinetic energy due to turbulent exchange than a jet that runs freely through the liquid or gaseous medium.

For the second condition it seemed to be necessary, at least for all configurations tested during this program, that the exhaust ports be slanted upward. Fig. 23 illustrates a plausible reason for this behavior. A jet is shown here, coming from the right intake port and going to the left exhaust port, which follows the desired fluid pattern. Its fluid path is fairly long but nowhere does it have to encounter any abrupt changes in direction or bending around sharp corners, which would lead to eddy losses. Coming from the left intake port, and going into the right exhaust port, is a jet of an alternate short-cut pattern which also might establish itself. The fluid path of this jet is undoubtedly shorter

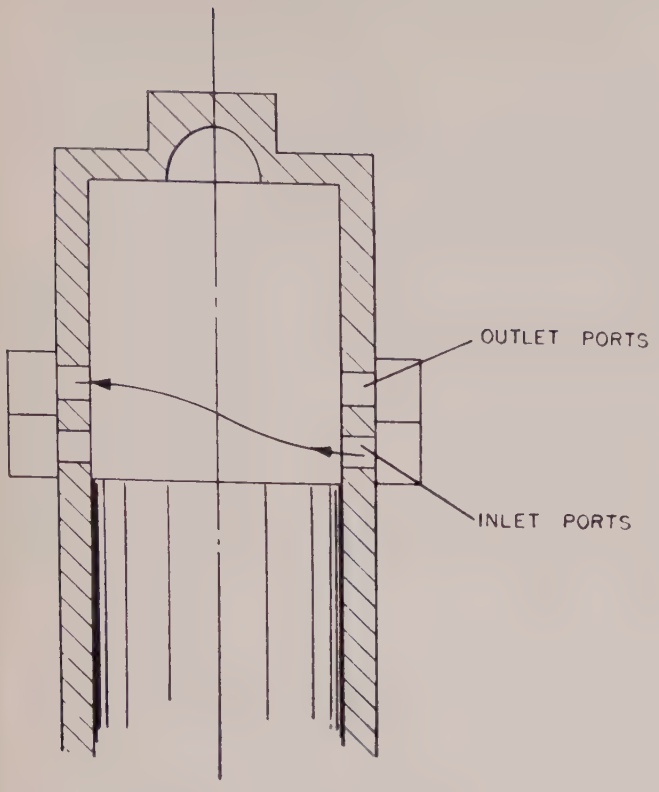


Fig. 21 – Initial configuration of ports

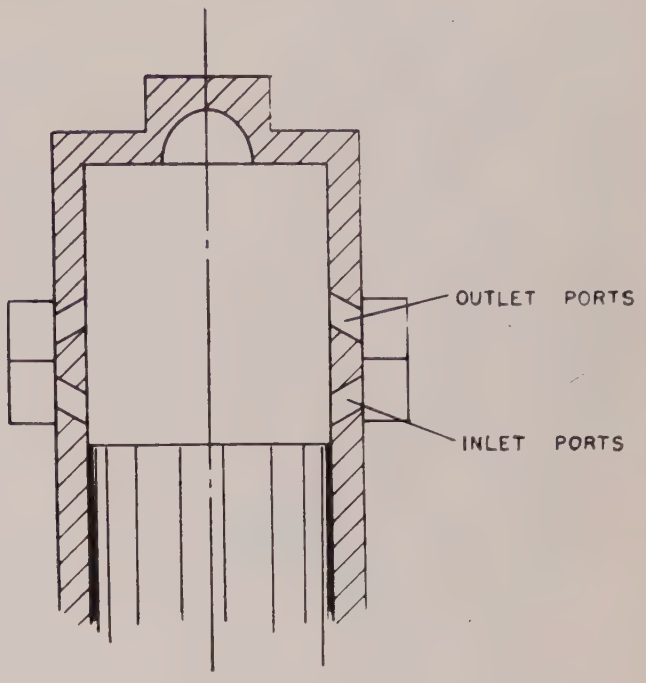


Fig. 22 – Basic principle of mushroom-type scavenging system

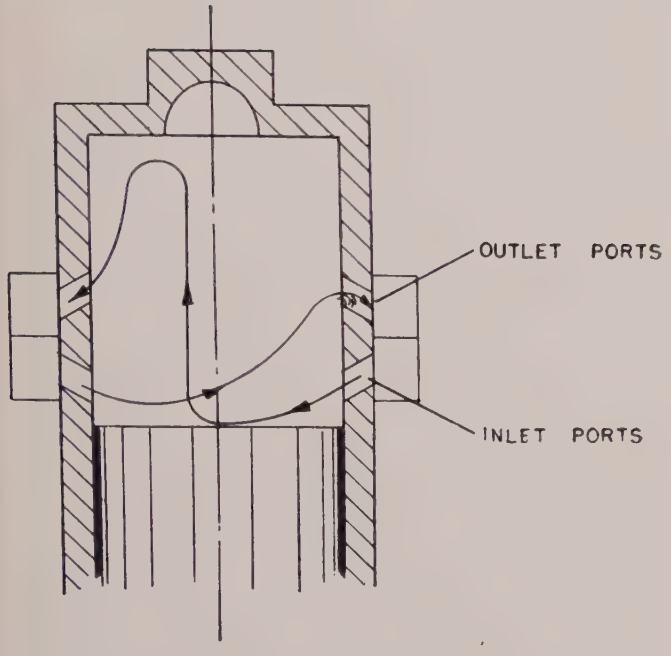


Fig. 23 – Explanation of mushroom-type principle

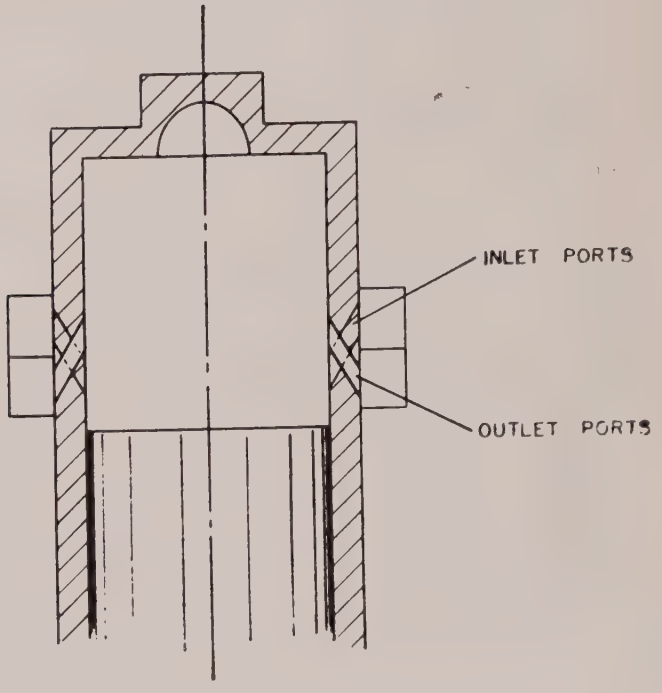


Fig. 24 – Crossed port system

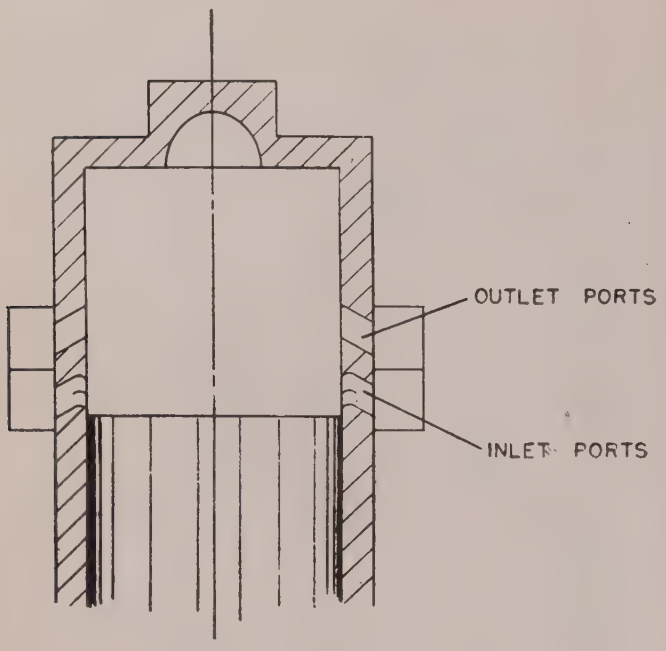


Fig. 25 – Louver-type port system

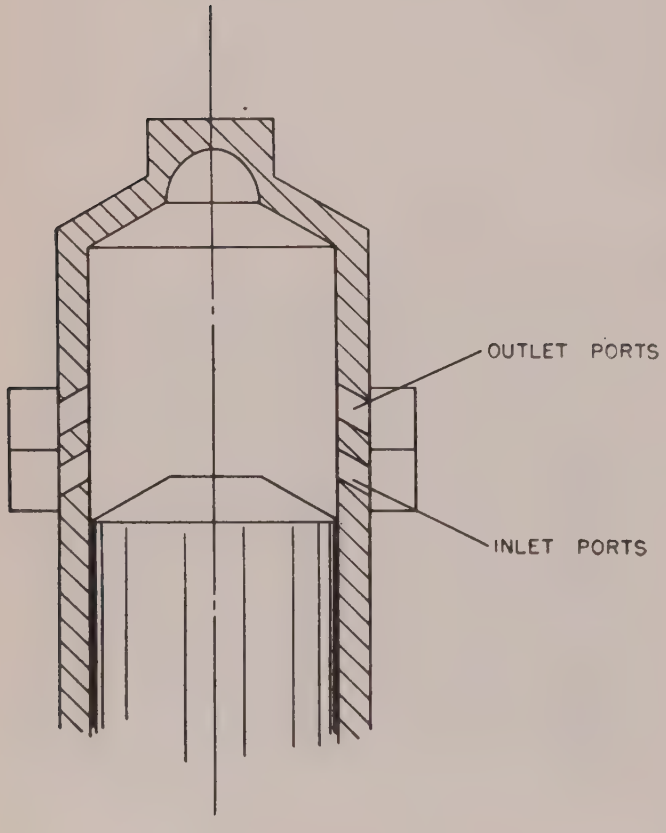


Fig. 26 – In-line port system

than the desired pattern. However, at its entrance into the exhaust port the change of direction is so violent that large eddies would be created at this point, choking the stream and giving rise to fairly high losses.

It is believed that, if the overall loss along such a jet pattern is greater than the overall hydraulic loss of the mushroom-type jet shape, the latter will be stable and the former will not occur.

With this assumption, further refinement of the scavenging system was undertaken. The principle laid down here was found valid throughout the course of our further investigations. The system shown in Figs. 22 and 23 was of course feasible for application in an engine of high bmep, since the lost stroke due to the large unused space between intake and exhaust ports would be excessive. Further research, therefore, centered upon the question of how to make the once-discovered principle usable for a scavenging system having a far less space between the inner openings of the intake and exhaust ports.

Basically, three design configurations were tried in solving this problem. The first one is shown in Fig. 24. Here the exhaust manifold is placed below the intake manifold and the ports cross each other inside the cylinder-wall material.

Significant of the development of the cross-current system was that it was necessary to separate the ports into a fairly large number of small ones. Large ports extending over a considerable cylinder angle did not result in good scavenging flow. Exchanging round ports for square ports did not affect scavenging efficiency in any measurable degree, although it resulted, of course, in lower scavenging pressure due to increased area.

This system worked very well, in that it gave good scavenging efficiencies as a function of the scavenging ratio. However, due to the crossing over, only about half of the circumference of the region set aside for both intake and exhaust can be covered with ports. Therefore, for a given height of the lost stroke the pressure differential to drive the fluid through the porting system is apt to be high, as smaller porting area is available than from a fully covered cylinder circumference.

This disadvantage led to the development of a system called the louver-port scavenging system. This uses the hydrodynamic technique of bending a stream by approximately 90 deg without too much hydraulic loss, by means of specially shaped louvers, as indicated in Fig. 25. Essentially, the direction of intake as well as exhaust ports is parallel and upward. Whereas, however, the exhaust ports are straight upward passages, the openings of the intake ports are covered by the louvers, which end close to the inside wall. Those louvers, at the last portion of the fluid's passage through the ports, will bend the intake flow in the desired direction. Hydrodynamic tests have shown that

such louver systems can reduce the loss of bending the flow to from one-fifth to one-sixth of what it would be for a simple rectangular bend. Significant developments on the louver system established the fact that in no case was it possible even to introduce the slightest amount of vortex into the flow without immediately obtaining short-cut flow of some degree.

When finally developed, this configuration gave the expected reduction of scavenging pressure with an efficiency equally as good as the crossed-port scavenging system. Considerations, however, arose about possible difficulties of manufacturing and using this kind of scavenging system on the actual engine. Application on the engine was therefore postponed until a third possibility of applying the principles described above had been tried.

This system was called the in-line port system and is illustrated by Fig. 26. In this system an attempt was made to modify the principle of impinging the intake jet at a slanting angle upon the piston crown by giving the piston crown a cone-shaped upper part, and thus being able to drill intake and exhaust port at the same angle. It was found possible to stabilize even this configuration, although in this case only such port arrangements were stable where the intake ports were smaller in area than the outlet ports, by a definite ratio. Intake ports larger than the critical ratio would result in short-cut flow. Attempts were made to cut off the tip of the cone as much as possible. Corresponding to the basic pattern (until the formation of the central column the jet has to be guided by the piston crown) this was successful only to a certain quite sharply defined point. If truncation of the cone was carried further, short-cut or unstable flow resulted again.

Generally, the system was characterized by the fact that fins arranged on the piston crown, calculated to assure complete radial flow, helped in only a limited degree to stabilize the flow, and were not considered worth while in view of the difficulty that would be encountered with such an arrangement on an actual engine. From the standpoint of best scavenging efficiency with the least pressure drop and scavenging ratio, the louver system proved the best one. Since, however, it was anticipated that considerable development would be necessary to make such a system feasible for an actual engine, the in-line system, which gave comparable although not quite as good results, was chosen for the final scavenging system on the engine.

In order, however, to obtain immediate experience with the mushroom-type scavenging system as affecting general characteristics of the engine to be developed, tests on the engine were started when the cross-port scavenging system had been developed, and from that time on engine tests and scavenging tests ran simultaneously.

Injection Tests – Two basic systems were used in the injection development, the second one being

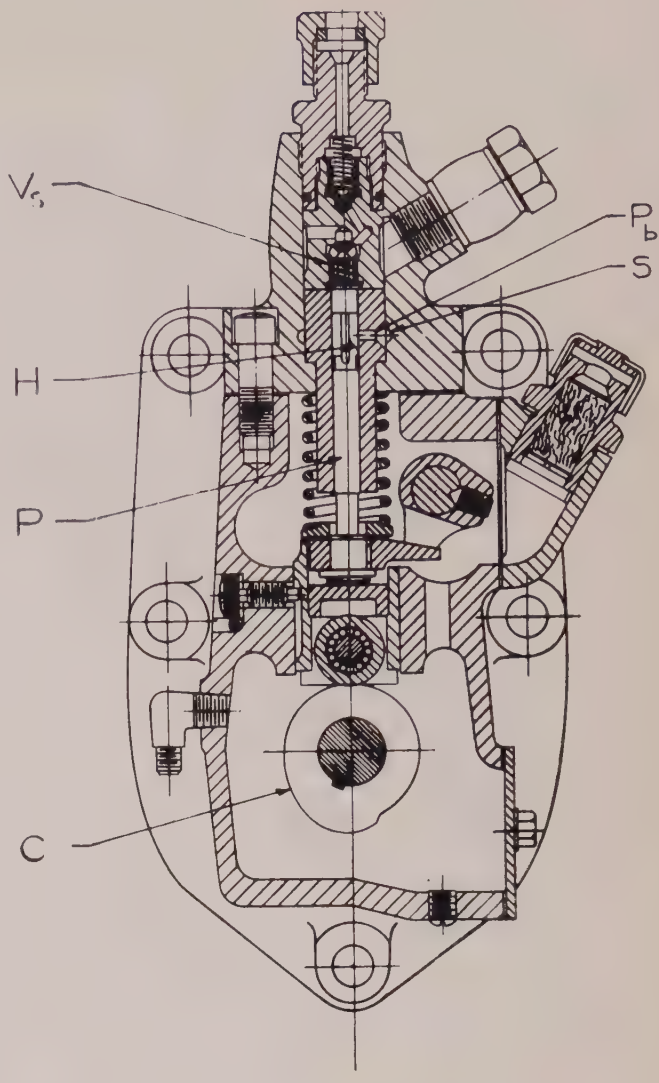


Fig. 27 – Cross-section through initial injection system

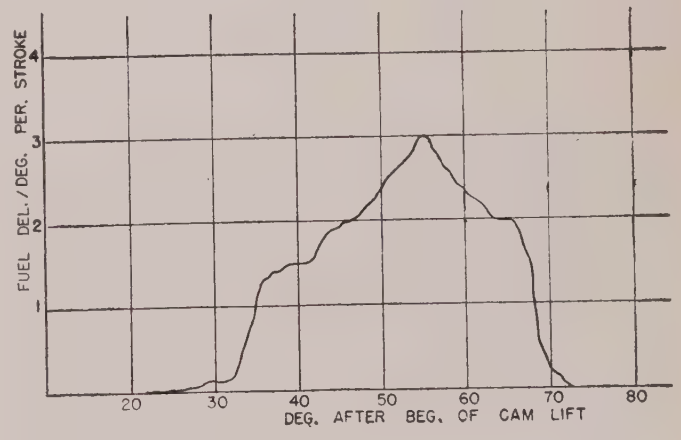


Fig. 28 – Initial rate of injection curve

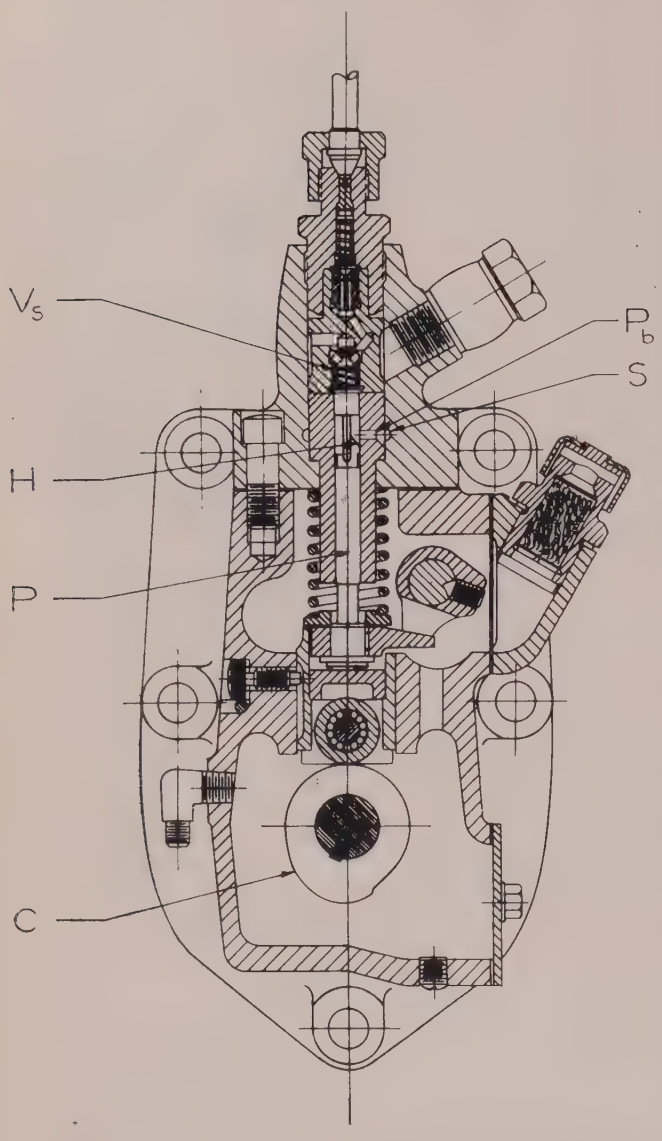


Fig. 29 – Cross-section through refined injection system

simply an improvement on the first. Fig. 27 shows the injection system as first developed. It is characterized by a cam C which imposes upon the plunger P a motion which, at the discharge valve of the pump, creates an injection pattern having the rate of injection shown in Fig. 13 (b). This is made possible by using the upward plunger movement from its beginning. A suction valve V_s, arranged on top of the plunger assembly, permits fuel to be discharged in this manner. The discharge valve is arranged without any special relief piston, so that the residual pressure in the injection lines stays fairly high. These lines must be fairly short in order to avoid after-injections. The amount of fuel injected is controlled in the conventional manner by a control helix H on the plunger. Rotation of the plunger controls the effective plunger lift by bypassing the fuel on top of the plunger through port P_b into sump S. Fig. 28 shows a curve of the rate of injection system measured on the rate of injection test stand.

The stroboscopic test stand was used mainly to measure injection lag, in order to determine correct timing. However, because of the extremely slow initial rate of rise of the cam, the injection lag could be taken as a measure of the degree of correlation between the rate of delivery at the plunger and at the nozzle. A long lag was taken to mean that considerable compression of the volume was necessary before the beginning of injection, and correspondingly less fuel was being injected at a lower rate. Conversely, a short lag meant less compression of the volume was necessary before the beginning of injection and correspondingly more fuel was being injected at a higher rate.

After the diesel test engine was sufficiently developed for safe operation, the first tests with gasoline injection were made. Gasoline was used throughout these tests, mixed with approximately 6% lubricating oil. Tests were run in such a way that peak combustion pressure and peak load were measured on the engine while it was running with diesel fuel. Then a switch was made from diesel fuel to gasoline and a sufficient time was allowed for the remaining diesel fuel to clear out of the injection system. By that time, due to the much greater ignition lag, the engine, now being timed far too late, was pulling a much smaller load. The timing was then readjusted until the load and the peak combustion pressure were the same as for the diesel injection. A run of 1 hr was then made at this load in order to test safe operating behavior.

With the injection system as initially developed on the injection test stands, it was found that the engine could take ordinary 87-octane gasoline quite well. However, with 100-octane aviation gasoline it was apt to be noisy. This was taken as an indication that the period of slow injection rate was not long enough. The stroboscopic test stand was therefore used to correct this condition. The discharge valve was redesigned to cut the residual volume still further. In addition, the return spring on the suction valve was strengthened, permitting the suction valve to settle completely during the dwell portion of the cam. Fig. 29 shows the new configuration. These changes made it possible to increase the portion of low fuel delivery by about 6 crankshaft deg. After this development was completed, the injection system proved capable of handling 100-octane gasoline with about the same noise level, the same load, and the same peak combustion pressure as the diesel fuel.

Penetration Tests – Penetration tests did not become of importance until it was found, during supercharging tests on the test engine, that the

injection nozzles which gave satisfactory performance during the unsupercharged operation did not increase the load obtainable from the engine in proportion to the density of the trapped charge. Since it was logical to suspect that insufficient penetration power at higher air densities might be the cause, a study was made as to how guiding lengths and diameter of the injection holes of the multiple-hole nozzles used would affect the penetration power. One uncertainty in the pressure bomb tests, which of course had to take place at normal temperatures, was the characteristic density by which supercharged and unsupercharged conditions could be imitated. The tests were therefore made first by checking penetration power at the same densities as were expected to exist in the combustion chamber at the moment of fuel injection and before ignition. These did not seem to correlate too well. Another attempt was made to compare them at a different similarity factor, more significant for the development of jets shooting into compressed air.

Theoretical considerations led to the assumption that the expression (absolute viscosity x density)$^{1/2}$, which characterizes conditions of incipient boundary-layer formation, might be a better parameter since the penetration of a jet into relatively slow-moving air should be closely governed by the boundary layer of the air forming around this jet.

This, incidentally, will lead to testing the pressure bomb at higher densities but lower pressures than exist in the actual combustion chamber. Although the tests made were too few to warrant any definite conclusions, it seemed that the tests made on the basis of the similarity parameter correlated better with experience actually obtained on the engine than those made at the same density.

Tests on Test Engine – Development on the test engine was directed primarily toward obtaining the maximum bmep obtainable from that engine. Two effects were found to be preeminent. The first was the necessity of finding a multiple-hole nozzle with sufficient penetration power and distribution for the supercharged conditions in the engine.

A development short-cut was again found to be feasible, due to the flexibility of the test stand. This was done by applying similarity theorems to the engine under both unsupercharged and supercharged conditions. It was assumed that if the engine were operating with the same combustion efficiency, the ratio of the imep to all gas pressures should be the same for all supercharging conditions. Therefore, serious deviation from this constancy should, in the main, be deviations in the ability of the engine to burn its trapped oxygen. In order to simplify this consideration, the imep reduced to atmospheric exhaust pressure for similar operation was plotted as a function of the scavenging ratio. If this unit imep was lower in one case than in another the conclusion could be

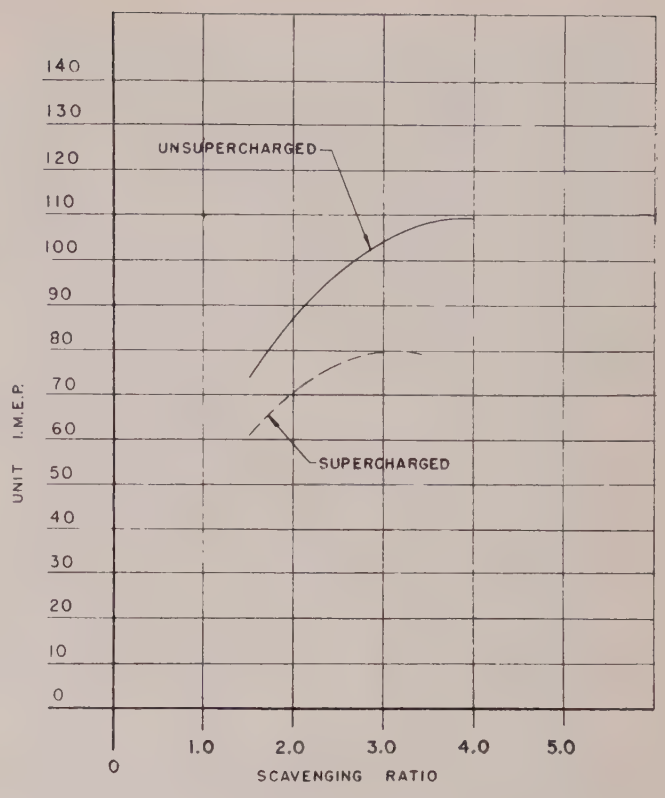


Fig. 30 – Typical test curve of bad supercharging conditions

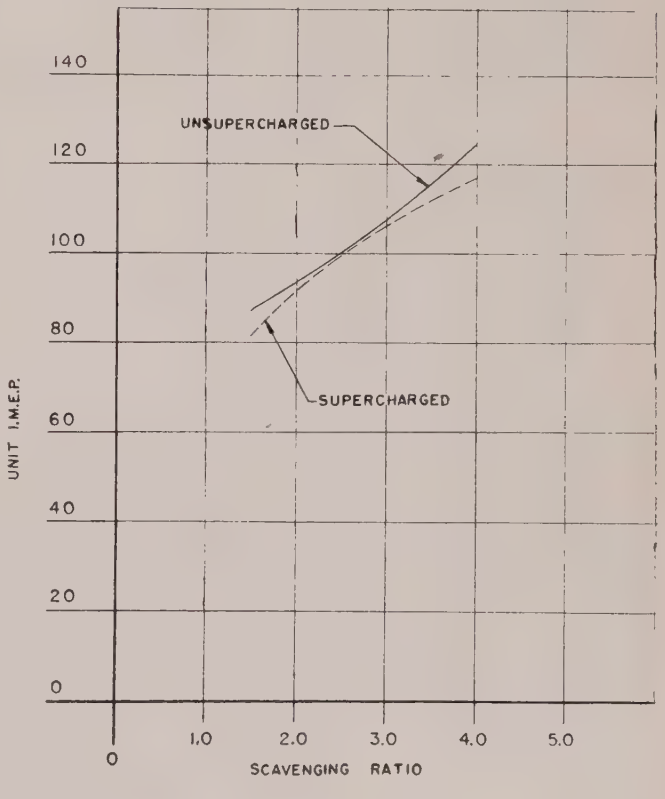


Fig. 31 – Typical test curve of good supercharging conditions

reached that combustion efficiency was not as good.

In the beginning of the development this happened quite frequently for the supercharged condition. Fig. 30 shows an actual test result. It was assumed that lack of penetration power for the supercharged condition was the cause. Accordingly, the attempt was made to develop jets with higher penetration power than described above. The success of this procedure can be seen in Fig. 31, which shows the comparison for supercharged and unsupercharged conditions at a later stage of development. As can be seen, they are approximately the same. From this, it was concluded that the limit to which increased penetration power could be used to increase the imep had been reached.

The next development item was the combustion-chamber shape. As has been said before, this shape could be changed very quickly on the test engine. By trial and error it was found that for each of the two scavenging systems optimum shapes for the combustion chamber could be developed. Fig. 32 shows the best and the worst combustion chamber for the in-line port scavenging system. Comparatively small changes in shape resulted in increased bmep limit obtainable from the engine.

After completion of the supercharging development, which had been run with the air at the same intake temperature as for unsupercharged conditions, the effect of inlet air temperature was tested by means of the heater. The well-known falling off in limit imep as a function of inlet temperature was observed. The temperature coefficient was somewhat higher than usually given in the literature. The exact figure was 0.865.

Very little development was necessary on the fourth step of the development plan – the aircooling system. Most of the ideas incorporated had been worked out in the analytical department through design on paper. The basic principle was to recognize possible hot spots in the engine and give them an extra cooling treatment. This was done for the injection nozzle, which was designed with a finned cooling nut, and for the piston, which was provided with a special heat seal. Except for the shape of the top ring, no further development was found necessary on the engine. Cylinder-wall temperatures as high as 520 deg were run continuously without detrimental effect on the engine, whereas normal operating temperatures of 450 deg could be sustained with cooling hp of approximately 2% of engine hp.

Conclusion

The scope of this paper is too small to permit discussion of all development experience obtained on this test engine. Only those results secured by judicious advance planning of test procedures have been mentioned. It is felt that, because of this planning, the development time of this engine – which incorporated an unusual amount of development – took considerably less time than it would otherwise have taken. To illustrate this point, Fig. 33 has been prepared, showing how the work on the different test stands was coordinated and run simultaneously.

Basically, the program began in November, 1948. Building of the scavenging test stand took approximately one month. The injection stands were already available. The CUE test base, which formed

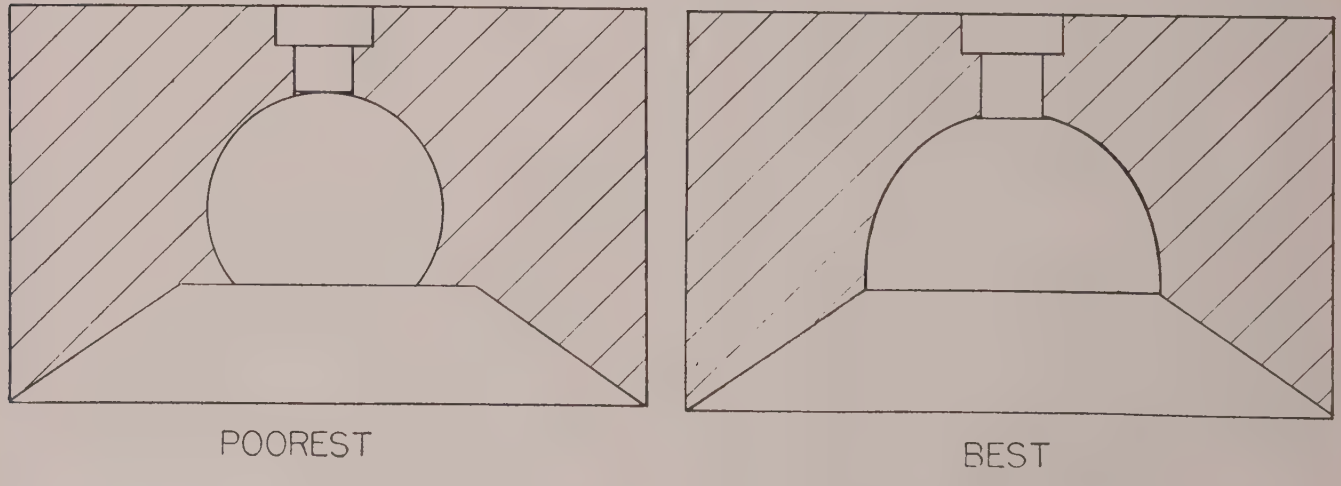


Fig. 32 – Extreme combustion-chamber shapes for in-line port system

SCAVENGING STAND	INJECTION STAND	PENETRATION STAND	ENGINE STAND	
BEGIN. OF TESTS	BEGIN. OF TESTS		BEGIN. OF DESIGN	1948
				1949 1st Half
DISCOVERY OF BASIC PRINCIPLE	END OF PRELIMINARY TESTS			
CROSSED PORTS	CHECK TESTS		BEGIN. OF TESTS	
LOUVER PORTS		BEGIN. OF DESIGN		1949 2nd Half
BEGIN. OF INLINE PORTS		BEGIN. OF TESTS	FIRST GASOLINE TESTS	
	BEGIN. OF REFINEMENT	END OF TESTS		
END OF DEVELOPMENT	END OF DEVELOPMENT		BEGIN. OF FINAL GASOLINE TESTS	1950 1st Half
END OF TESTS	END OF TESTS		INSTALLATION OF FINAL SCAVENGING SYSTEM	
			END OF TESTS	

Fig. 33 – Development time schedule

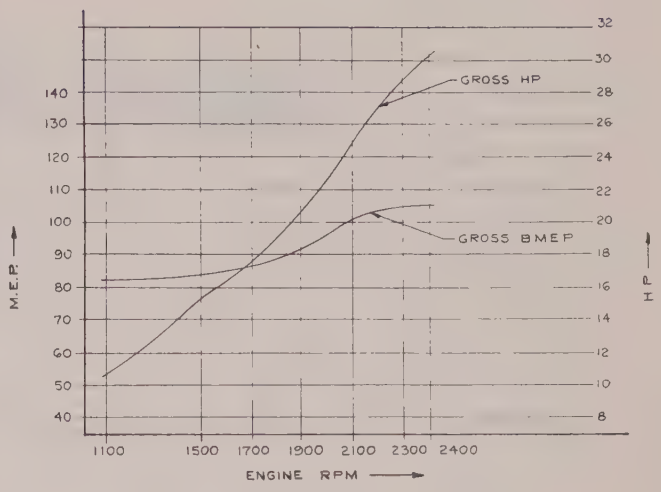


Fig. 34 – Final test engine characteristics

the basis of the engine test stand, was also available through the Armour Research Foundation, which helped considerably in this program; they also provided test space and test personnel. As can be seen from Fig. 33, design on the engine test stand began while the actual tests were being started on the scavenging test stand. It can also be seen that the discovery of the basic scavenging system occurred shortly before the engine test stand was scheduled to go into operation. Therefore, the largest amount of development work on the test engine could be done with the preliminary scavenging system, which gave the same scavenging pattern, but at a higher scavenging differential than the final one. For this reason, the scavenging tests and the engine tests overlapped considerably.

The penetration test stand was used for only a comparatively short time – its use ending approximately two months before completion of tests on the preliminary scavenging system on the engine. The preliminary injection system development was finished in time for the first engine tests. Further tests on the injection test stand tie in rather closely with the gasoline injection tests on the engine test stand, run in October, 1949.

About 1½ months after the final scavenging system was developed on the scavenging test stand it was introduced on the engine. Work on the engine stopped about two months later than the work on the remainder of the test stands. All in all, a satisfactory overlap of the development time occurred, and can be observed on this layout, which shows that by judicious application of flexible test methods, development which usually is done consecutively can be done simultaneously. We believe that the results – development of a new aircooled multifuel lightweight diesel engine – in this comparatively short time, amply justify test planning and the use of special flexible test stands built for the occasion. Fig. 34 shows the characteristics of the engine as it was finally taken off the test stand on June 26, 1950.

ORAL DISCUSSION

Reported by J. G. Ryan
Shell Oil Co.

Harold V. Nutt, U. S. Naval Engineering Experiment Station: In considering performance characteristics of the engine I would like to ask three specific questions.

1. Why was the engine developed for the use of gasoline?
2. Can gasoline be burned properly, and are changes necessary to use diesel fuel?
3. Is it possible to get this engine started and working in below zero temperatures?

Mr. Spannhake: The Air Force wanted an engine which could use any fuel available in any part of the world, whether it be jet fuel, aviation gasoline, or diesel fuel.

In answer to the second question, gasoline can be used. Also, the engine can be started on diesel fuel, and then switched to gasoline. Bleeding of the fuel supply system is required for smooth operation. Peak pressures fell off if the engine was set for best timing. The same peak pressures could be made with diesel fuel. Finally, we checked to be certain that the engine would start on gasoline. We could use 80 octane number, but 100 octane number gave some difficulty, necessitating some change in injection characteristics. Timing was set at 30 deg btc for diesel fuel. For 100 octane gasoline a setting of 42 deg btc was made for starting, and comparably advanced for the end of injection.

In answer to the third question, we did not cold-start the engine. The lowest temperature we tried was 60 F.

Automatic Control Systems
Criteria on

IN the past several years, there has been increasing development and interest in automatic control. In the fields of gun direction, guided-missile control, and control of gas turbine engines, for instance, where very refined and accurate controlled behavior is required, need still exists for further development of the methods of control analysis and design.

Recent developments in this field have been concerned chiefly with the problem of control analysis both in the realm of linear systems[1] and in the realm of nonlinear systems.[2] These analytical works answer questions concerning the behavior of a given system or the effect on its behavior of changing certain constants in the system.

Another problem of equal and, in some cases, greater importance is that of control synthesis. Work on this problem seeks to find the method of design when certain criteria concerning the behavior of a controlled system are given. The method decided upon should give all aspects of the system; for instance, where the system should be linear, what the general configuration should be, and what the precise values of all the constants should be.

This synthesis problem has hardly been broached in literature. The use of analysis as a design pro-

[1] See "Theory of Servomechanisms," edited by Hubert M. James, Nathaniel B. Nichols, and Ralph S. Phillips. Published by McGraw-Hill Book Co., Inc., New York, 1947.
[2] See "Introduction to Nonlinear Mechanics," by N. Minorsky. Published by J. W. Edwards, Ann Arbor, Mich., 1947.

AN analytic method for the design of automatic controls has been developed by these authors. The method starts from certain arbitrary criteria on the behavior of the controlled system. It then gives those physically realizable equations that the control system can follow in order to realize this behavior. The criteria used are developed in the form of certain time integrals.

General results are shown for systems of second order and of any number of degrees of freedom. Detailed examples for several cases in the control of a turbojet engine are presented.

The Authors

AARON S. BOKSENBOM is an aeronautical research scientist at the Lewis Flight Propulsion Laboratory, National Advisory Committee for Aeronautics, in Cleveland. He graduated from Purdue University, receiving a degree in mechanical engineering.

RICHARD HOOD recently joined Chance Vought Aircraft Division of United Aircraft Corp. in Dallas, Texas. During the previous seven years he was an aeronautical research scientist at the NACA Lewis laboratory. He received a bachelor's degree in mechanical engineering from Illinois Institute of Technology, and a master's degree from Case Institute of Technology.

Satisfying Certain General Transient Behavior

Aaron S. Boksenbom and Richard Hood
Lewis Flight Propulsion Laboratory, NACA

This paper was presented at the SAE Annual Meeting, Detroit, Jan. 14, 1953

cedure offers a partial solution to this problem in that the analysis of a large number of cases may reveal, coincidentally, one that satisfies the desired criteria of controlled behavior. Such a method is, at best, long and tedious and almost always would result in compromises because the systems chosen to be analyzed would probably be such that they could never satisfy all the desired criteria.

A method for designing a linear system to satisfy certain special criteria when operating on a random input has been developed by Wiener.[3] This method is applicable as an addition to a control system whenever random external disturbances are involved. A partial solution to the synthesis problem has been developed[4] in satisfying the criteria of noninteraction for systems with many degrees of freedom.

An analysis made at the NACA Lewis laboratory during the latter part of 1950 and presented herein develops a rational method of control synthesis that starts from any arbitrary but physically realizable criteria and results in the equations for the best system that satisfies these criteria. As is shown, the nature of the criteria, in general, requires minimizing certain time integrals by using the calculus of variations, and the methods developed are an application of the calculus of variations to the problem of control synthesis.

A careful scrutiny is first made of the whole problem, followed by a development of general results. These general results are then applied in examples to the design of turbojet-engine control systems. In general, the methods used vary according to the order of the differential equation describing the plant, the process, or the system being controlled and according to the number of degrees of freedom or independent variables. Detailed analyses are presented for application to a system of first order and of one degree of freedom. General results for systems of second order and any number of degrees of freedom are developed in the appendix.

Symbols

The following symbols are used in this report:

a, b, c = Gas turbine engine characteristic constants

C = Constant

E = Function of λ

F, H = Functions of y and $\dot{y}$

$F(t_1)$ = Indicates $F[y(t_1), \dot{y}(t_1)]$

$F_y(y, \dot{y})$ = Indicates partial differentiation with respect to y

f, f_o = Arbitrary function

$f_1, f_2, \cdots$ = Weighting functions used for gas turbine control criteria

G, G_1, G_2 = Functions of y

g = Function used for gas turbine engine surge criterion

h = Function used for gas turbine engine blowout criterion

K = Controller gain

L, L_1, L_2 = Temperature limits for gas turbine engine

N, P, T, w_f = Deviations of gas turbine engine speed, compressor discharge pressure, characteristic temperature, and fuel flow, respectively, from values at some common equilibrium condition

[3] See "Extrapolation, Interpolation, and Smoothing of Stationary Time Series with Engineering Applications," by Norbert Wiener. Published by the M. I. T. Technical Press and John Wiley & Sons, Inc., New York, 1949.

[4] NACA Report No. 980, 1950, "General Algebraic Method Applied to Control Analysis of Complex Engine Types," by Aaron S. Boksenbom and Richard Hood.

N_e = Actual engine acceleration

n = Exponent

t = Time

t_1 = Time at end of transient

δt_1 = Variation in time at end of transient

$w, y, z, \delta w, \delta y, \delta z$ = Independent variables, functions of time

ϵ = Small number

$\lambda, \lambda_1, \lambda_2, \lambda_3$ = Arbitrary constants

σ = Gas turbine engine time constants for response to temperature

τ = Transient time constant of controlled system

Subscripts:

$_0$ = Initial condition of variable

$_f$ = Final value of variable

$_s$ = Setting or desired value of variable

Superscripts:

The asterisk indicates case different from optimum.

The dot indicates differentiation with respect to time.

The prime indicates differentiation with respect to the argument shown.

Survey of Problem

Control Problem – An important aspect of the control-synthesis problem is a clear definition of the criteria of desired controlled behavior. If a variable y is to be controlled, a reasonable criterion is that the time integral of some function of y is to be a minimum or a constant; that is:

$$\int_0^{t_1} f(y)dt = \text{Constant or minimum} \qquad (1)$$

or:

$$\int_0^{t_1} (y-y_s)^2 dt = \text{Constant or minimum} \qquad (2)$$

Equation 2, for instance, weights the error in y as the square and according to the time duration of that error. Another type of criterion may be that which requires a certain time duration to be a minimum or a constant; that is:

$$\int_0^{t_1} dt = \text{Constant or minimum} \qquad (3)$$

The use of a single criterion, such as equation 1, will usually yield $f(y) =$ constant. This result is reasonable because $f(y)$ can usually be made identically a constant if no additional criteria are imposed on other variables in the system. Certain limiting conditions usually exist on other variables in the system and these conditions must be included in the original criteria.

Thus, a possible criterion could be written:

for:

$$\left.\begin{aligned} &\int_0^{t_1} (y - y_s)^2 dt = \text{Minimum} \\ &\int_0^{t_1} f(z)dt = \text{Constant} \end{aligned}\right\} \qquad (4)$$

If, for instance, $y =$ engine speed and $z =$ characteristic temperature of a gas turbine engine, the criterion of equation 4 states that it is desired to design a control system such that, for a particular value of a temperature integral, the integral of the speed-error squared is a minimum. This criterion may be used if, for instance, it is known that an over-temperature condition can be tolerated for a certain period of time and it is desired to keep the average speed error at a minimum during a transient.

The general theory will show that as many criteria as desired of the type shown in equations 1 to 4 can be included together and a control system can be derived that automatically satisfies all these criteria simultaneously.

Another aspect of the control criteria is the end conditions of the integrals of equations 1 to 4. The time interval for which these integrals are to be a minimum or a constant must be chosen. A reasonable time interval is any duration during which essential external disturbances are constant and during which the system to be controlled moves from one essential level of operation to another. The essential external disturbances are those that cannot be immediately corrected by the control system. If an essential external disturbance were allowed within the time interval of the criteria, no physically realizable system could be expected to anticipate this disturbance so as to behave properly before this disturbance occurs. An essential level of operation is any specific condition of only those variables that must be continuous. It will be shown that the essential level of operation appears as a natural boundary condition for the type of criterion used. In the case of a turbojet engine the transient behavior of which can be described by a first-order differential equation, the engine speed determines the level of operation. If a lag exists in the fuel system or between temperature and engine speed, then both engine speed and acceleration are required to describe the essential operating level of the engine.

Analytic Problem – The control system resulting from any design method must be physically realizable. There are two aspects to this problem. First, it is possible to set down criteria that are not realizable with any system or are incompatible with each other. If such criteria are used, the unrealizability will appear either as a requirement on the control to look ahead into the future or as an inability to satisfy the boundary conditions of some differential equation. In most cases, a clear understanding of the criteria used and of the system to be controlled will indicate incompatibilities of this sort.

The second aspect of physical realizability is purely mathematical. It is desired to derive a description (a differential equation) of the control or the controlled system that satisfies the criteria of control and all the necessary boundary condi-

tions that arise in the derivation of this equation. Although the mathematical solution of the problem may be any derivative or integral of this differential equation, the physical solution of the problem requires the differential equation that itself satisfies the boundary conditions and for which no undetermined constants of integration exist. Thus, such forms as:

$$\left.\begin{aligned} \dot{y} &= C\dot{x} \\ y &= Cx \end{aligned}\right\} \qquad (5)$$

are not necessarily interchangeable as descriptions of some part of a controlled system because the forms differ by an undetermined constant of integration. For stable linear systems, the effect of this constant becomes vanishingly small; for the general nonlinear systems presented herein, however, this constant must be considered.

Stability Problem – The requirement of stability is a special criterion that does not enter into the main body of the methods of this report. It may enter in the final steps of the method where the final differential equation describing the controlled system may be the integral of a higher-order differential equation that satisfies the necessary boundary condition for stability. In addition, it is always necessary to add to the controlled system a stability device that does not affect the behavior of the system as far as satisfying the other criteria is concerned. This device can be described as follows:

$$\dot{y} = 0 \quad \text{when} \quad y = y_s \qquad (6)$$

or, for a second-order system:

$$\ddot{y} = 0 \quad \text{when} \quad y = y_s \quad \text{and} \quad \dot{y} = 0 \qquad (7)$$

General Theory and Results

It has been shown that the criteria for control can be developed in the following forms:

$$\left.\begin{aligned} &\int_0^{t_1} f(y)dt \\ &\int_0^{t_1} (y - y_s)^2 dt \\ &\int_0^{t_1} f_o(z)dt \\ &\int_0^{t_1} dt \end{aligned}\right\} \qquad (8)$$

and so forth. If, for such a list of criteria, one of the integrals is to be a minimum under the conditions that the other integrals are to be constant, it is sufficient[5] to make:

$$\int_0^{t_1} f(y)dt + \lambda_1 \int_0^{t_1} (y - y_s)^2 dt + \lambda_2 \int_0^{t_1} f_o(z)dt + \lambda_3 \int_0^{t_1} dt = \text{Minimum} \qquad (9)$$

or:

$$\int_0^{t_1} [f(y) + \lambda_1 (y - y_s)^2 + \lambda_2 f_o(z) + \lambda_3]dt = \text{Minimum} \qquad (10)$$

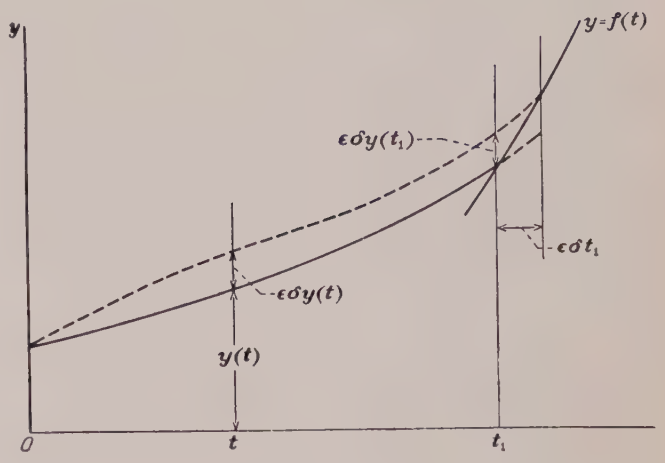


Fig. 1 – Illustration of curves for minimization of $\int_0^{t_1} F(y, \dot{y})dt$

The λ's are arbitrary constants that enter into the control system as the adjustable parameters and are precisely determined by the choice of values that the constant integrals are to have.

The technique of the λ multipliers is widely used for problems of this type where one condition is to be a minimum under other restrictive conditions. Indeed, the conditions need not be in integral form and any functional or differential relation among variables can be handled in a similar manner.[6]

Equation 10 can be made very general when all possible restrictive conditions are included. In the final equations, which are derived later, if any one criterion is not to be used, then the corresponding $\lambda \to 0$. If any of the criteria is to be zero (as for the case of a variable having an absolute limit), then the corresponding $\lambda \to \infty$.

If, for this development, the system to be controlled is of first order and of one degree of freedom (has one independent variable), then the variables y and z are such that $z = z(y, \dot{y})$. Equation 10 can then be written, in general, as:

$$\int_0^{t_1} F(y, \dot{y})dt = \text{Minimum} \qquad (11)$$

In equation 11, F is a continuous function of y and $\dot{y}$, and y is a continuous function of time.

The calculus of variations[5] is used to determine y as a function of time such that the integral of equation 11 is a minimum; that is, if the solid curve of Fig. 1 makes the integral a minimum,

[5] See "Lectures on the Calculus of Variations," by Oskar Bolza. Published by Hafner Publishing Co. (G. E. Stechert & Co.), New York, 1946 (reprint of 1904 edition).

[6] See "The Variational Principles of Mechanics," by Cornelius Lanczos. Published by University of Toronto Press, Toronto, 1949.

any other curve (such as the dashed one) will make the integral equal to or greater than the first integral. If the two curves are very close, the condition that the two integrals are equal will make the integral of equation 11 stationary (maximum, minimum, or inflection point).

The problem is similar to that of finding the maximum or minimum point on a curve by setting the first derivative equal to zero. Whether a maximum or minimum point exists is decided by the second derivative at that point. In the variational problem, proving a true minimum involves taking the "second variation." As in the problem of finding a minimum on a curve, the second variation proves, at most, a local minimum and not an absolute minimum. For the specific examples discussed herein, an absolute minimum is proved by another method. In many cases, the physical meaning of the equations will indicate the existence of a unique minimum that obviates going further than the "first variation."

On Fig. 1, the curve for minimizing the integral of equation 11 is the solid curve having the value $y(t)$ at any time. Any other curve differing from it by a small amount is shown as the dashed curve having the value $y(t) + \varepsilon\,\delta y(t)$ at any time, where ε is a small number and $\delta y(t)$ is an arbitrary continuous function of time. The condition for the integral of equation 11 to be stationary is:

$$\frac{d}{d\epsilon}\int_0^{t_1+\epsilon\delta t_1} F(y + \epsilon\delta y, \dot{y} + \epsilon\delta\dot{y})dt = 0 \qquad (12)$$

at: $\varepsilon = 0$

where $\frac{d}{dt}[\delta y(t)] = \delta\dot{y}(t)$ (see footnote 5). The time duration for the integral of equation 12 is such as to start at some definite time ($t = 0$) but not to end at a definite time but rather along some curve $y = f(t)$ (Fig. 1), in order to allow the proper boundary conditions of moving from one essential level of operation to another, as previously discussed.

Performing the operation indicated by equation 12 leads to:

$$\int_0^{t_1} F_y\delta y dt + \int_0^{t_1} F_{\dot{y}}\delta\dot{y}dt + F(t_1)\delta t_1 = 0 \qquad (13)$$

Integrating the second term of equation 13 by parts gives:

$$\int_0^{t_1}\left[F_y - \frac{d}{dt}(F_{\dot{y}})\right]\delta y dt + F_{\dot{y}}\delta y\Big]_0^{t_1} + F(t_1)\delta t_1 = 0 \qquad (14)$$

Because δy is an arbitrary function, the integral and the boundary-condition terms must vanish separately. The geometry at the end condition ($t = t_1$, Fig. 1) gives:

$$\delta y(t_1) = [f'(t_1) - \dot{y}(t_1)]\delta t_1 \qquad (15)$$

as $\varepsilon \to 0$. The two conditions that follow from equation 14 then become:

$$\int_0^{t_1}\left[F_y - \frac{d}{dt}(F_{\dot{y}})\right]\delta y dt = 0 \qquad (16)$$

and:

$$\delta t_1\left\{F(t_1) + F_{\dot{y}}(t_1)[f'(t_1) - \dot{y}(t_1)]\right\} - F_{\dot{y}}(0)\delta y(0) = 0 \qquad (17)$$

The time interval during which the criterion of equation 11 is to hold is considered as that during which the system moves from one essential operating level to another; in this case, from one definite value of y to another definite value of y. Thus:

and:
$$\left.\begin{aligned}\delta y(0) &= 0\\ f'(t_1) &= 0\end{aligned}\right\} \qquad (18)$$

Equation 16 is satisfied only if the integrand is zero, and, because $\delta t_1 \neq 0$, the two conditions of equations 16 and 17 become:

$$F_y = \frac{d}{dt}(F_{\dot{y}}) \qquad (19a)$$

and:

$$F(t_1) = F_{\dot{y}}(t_1)\dot{y}(t_1) \qquad (19b)$$

Equation 19a need not hold at $t = 0$ because $\delta y(0) = 0$. The only condition that need hold at $t = 0$ is that $F_{\dot{y}}(0)$ is finite, which will be true if y is continuous at $t = 0$. At the start of a new transient, $\dot{y}$, F, F_y, and $F_{\dot{y}}$ may be discontinuous, whereas at other points ($0 < t \leq t_1$) $F_{\dot{y}}$ will be continuous because of equation 19a.

Equation 19a is the differential equation for the $y(t)$ that satisfies the original criterion of equation 11. The physical answer to the problem is the first integral of equation 19a, which satisfies the boundary-condition equation 19b. This solution is:

$$F(y, \dot{y}) = \dot{y}F_{\dot{y}}(y, \dot{y}) \qquad (20)$$

and whenever y, $F_{\dot{y}}$, and so forth are continuous:

$$\dot{y}\left[F_y - \frac{d}{dt}(F_{\dot{y}})\right] = 0 \qquad (21)$$

by differentiating equation 20 with respect to time. Thus, either $\dot{y} = 0$ (which is true only during stability) or equation 19a is satisfied.

Thus, equation 20 is the description of that physically realizable system the behavior of which will automatically and simultaneously satisfy those criteria included in the function F during that time interval for which the external disturbances are constant and during which the system goes from one operating level to any other operating level. A stability device must be added to the system; the description of such an ideal device is:

$$\dot{y} = 0 \quad \text{when} \quad y = y_s \qquad (22)$$

An additional condition must be met if $\dot{y}$ is discontinuous in the interval $0 < t < t_1$. In this case, $F_{\dot{y}}$ must be continuous during the discontinuity in $\dot{y}$; $F_{\dot{y}}$, however, it will usually be discontinuous when $\dot{y}$ is discontinuous. This discontinuity in $F_{\dot{y}}$ usually means that some essential external disturbance has entered the system and the point of discontinuity must be the start of a new time interval.

Application to Turbojet-Engine Controls

In the usual case of designing turbojet-engine controls, the engine speed that sets the essential operating level of the engine and, in the main, other pertinent characteristics such as thrust, is to be set or controlled. Limiting conditions of the engine are those of overspeed, overtemperature, compressor surge, and rich burner blow-out. The following criteria on the behavior of this engine are typical:

$$\left.\begin{array}{ll} \int_0^{t_1} f_1(N - N_s)dt & \text{for speed control} \\ \int_0^{t_1} f_2(N)dt & \text{for speed overshoot} \\ \int_0^{t_1} f_3(T)dt & \text{for temperature overshoot and undershoot} \\ \int_0^{t_1} f_4[P - g(N)]dt & \text{for compressor surge} \\ \int_0^{t_1} f_5[P - h(N)]dt & \text{for blow-out} \\ \int_0^{t_1} dt & \text{for rise time} \end{array}\right\} \quad (23)$$

Figs. 2a to 2e show the nature of the functions appearing in expressions 23. The variable involved is essentially weighted by the function shown and integrated with respect to time. The quantity $P - g(N)$ is the amount the compressor-discharge pressure exceeds the safe pressure for surge and $g(N)$ (shown in Fig. 2f) is the compressor-discharge pressure for each engine speed at a safe value below surge. Rich burner blow-out can be handled in a similar manner ($h(N)$ as shown in Fig. 2g). The rise time is the total time for the system to move from one essential operating level to the other.

The linearized engine characteristics can be expressed, by assuming first-order behavior, as follows:

$$T = aN + a\sigma\dot{N} \quad (24a)$$

$$P = bN + cT \quad (24b)$$

Thus, the integrals of expressions 23 are of the form:

$$\int_0^{t_1} f(N, \dot{N})dt \quad (25)$$

where f is a continuous function of N and $\dot{N}$, and N (barring impulsive temperatures and the like) is a continuous function of time.

Speed Control – Case A – If only the error in speed control is considered important, the criterion becomes:

$$\int_0^{t_1} f_1(N - N_s)dt = \text{Minimum} \quad (26)$$

where $F = f_1(N - N_s)$.

Equation 20 becomes:

$$\left.\begin{array}{r} f_1(N - N_s) = 0 \\ \text{and (from the nature of } f_1\text{, Fig. 2a),} \\ N = N_s \end{array}\right\} \quad (27)$$

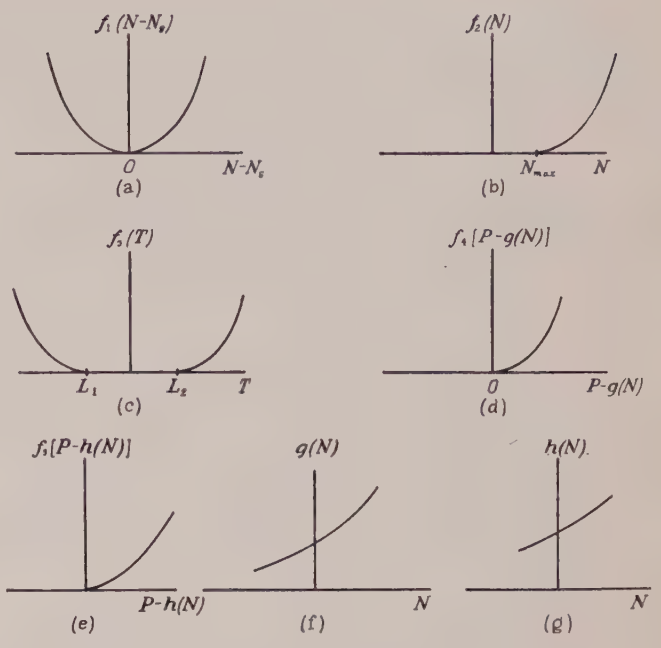


Fig. 2 – Arbitrary weighting functions for various control criteria and pertinent engine characteristics involved; (a) speed control; (b) speed overshoot; (c) temperature overshoot and undershoot; (d) compressor surge; (e) blow-out; (f) compressor line below surge; (g) compressor line below blow-out

This result means that, in the absence of other criteria on the engine behavior, this speed control should keep speed error identically zero, which is physically realizable only in the sense of allowing infinite temperatures and the like to keep the speed error identically zero. This result, however, is inconsistent with the previous development of equation 20 in that N is now a discontinuous function of time, and the time interval of the integral of equation 26 is zero. This instance is actually a trivial case of the general problem. The result (equations 27) does indicate that a criterion like that of equation 26 must be accompanied by an additional criterion (temperature overshoot, for instance) to give a physically realizable system.

Speed Control with Temperature-Limiting Criteria – Case B – If the error in speed control and the overshoots and undershoots in temperature are to be considered the primary criteria of control, then from equations 10 and 11:

$$\int_0^{t_1} [f_1(N - N_s) + \lambda f_3(T)]dt = \text{Minimum} \quad (28)$$

where $F = f_1(N - N_s) + \lambda f_3(T)$.

Equation 20 then becomes:

$$f_1(N - N_s) + \lambda f_3(T) = \lambda a\sigma\dot{N}f_3'(T) \quad (29)$$

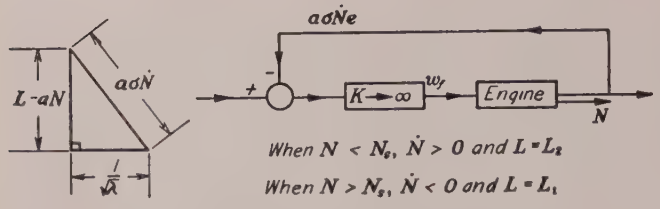


Fig. 3 – Schematic diagram of control for turbojet engine for case of speed control with temperature-limiting criteria (case B). Stability device must be added to this system

and the ideal stability device is such that:

$$\dot{N} = 0 \quad \text{when} \quad N = N_s \tag{30}$$

Equations 29 and 30 describe the complete control system.

In Fig. 2c, it is convenient to let $f_3(T) = (T - L_2)^n$ for $T > L_2$ and $f_3(T) = (T - L_1)^n$ for $T < L_1$. In general, the power n should be >1, because, when $n < 1$, T may be infinite and of such nature as to make N discontinuous and physically unreal even though $\int_0^{t_1} f_3(T)dt$ is finite. In the examples of this report, $n = 2$ and $f_1(N - N_s) = (N - N_s)^2$.

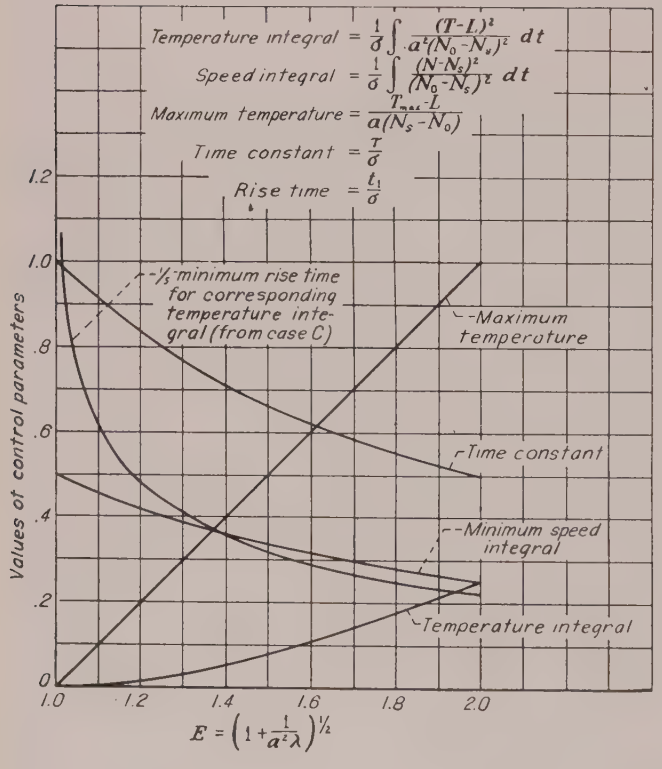


Fig. 4 – Various control parameters as functions of E for speed control time with temperature-limiting criteria (case C) when accelerating to limiting temperature

A variety of methods of setting up the control system to realize equation 29 exists. When the preceding expressions for f_1 and f_3 are used, equation 29 can be put in the convenient form:

$$\left.\begin{aligned} &\frac{(N - N_s)^2}{\lambda} + (L - aN)^2 = a^2\sigma^2\dot{N}^2 \\ &\text{where, for acceleration, when } N < N_s, \\ &\qquad \dot{N} > 0 \quad \text{and} \quad L = L_2 \\ &\text{and, for deceleration, when } N > N_s, \\ &\qquad \dot{N} < 0 \quad \text{and} \quad L = L_1 \end{aligned}\right\} \tag{31}$$

A schematic diagram of the system is shown in Fig. 3, where equation 31 is considered to give a desired $\dot{N}$ for any value of N, N_s and L. Consistent with equation 31, a right-triangle construction is used to give a desired N. The actual $\dot{N}_e$ can be made very close to $\dot{N}$ by using a high-gain proportional controller, as shown. Provision must be made to change the sign of $\dot{N}$ and the value of L when $N - N_s$ changes sign. In addition, the stability condition requires a provision for making $\dot{N} = 0$ whenever speed error is very small or zero.

The use of a high-gain proportional controller, which follows from the requirement that $\dot{N}$ may be discontinuous, means that the fuel-flow rate required may be infinite if lags exist in the fuel system or in the feedbacks. But, as no criterion has been set on fuel-flow rate, this requirement does not violate the original criteria. If necessary, however, a criterion on fuel-flow rate may then be added to equation 28. Even though a criterion on temperature is being satisfied, no direct measurement of temperature is used in Fig. 3. Actually, the equation for temperature (equation 24a) is used as an indirect indication of temperature.

The control system of Fig. 3 has one adjustable parameter λ. For any value of λ, this system will, for the value of integral temperature-overshoot squared obtained, give the minimum value of integral speed-error squared. The value of λ determines the actual value of the integral temperature-overshoot squared.

The integral temperature-overshoot squared as a function of λ is shown in Fig. 4 for the special case where $aN_s = L$; that is, acceleration or deceleration to the speed that corresponds to limiting temperature. In this case, the system of Fig. 3 becomes a simple first-order lag system and equation 31 becomes:

where:

$$E(L - aN) = a\sigma\dot{N} \tag{32}$$

$$E = \left(1 + \frac{1}{a^2\lambda}\right)^{1/2}$$

In Fig. 4, the integral speed-error squared, the maximum temperature, and the time constant for this transient are also shown as functions of λ. The

curve showing the minimum rise time for the corresponding temperature integral is included for a comparison that will be discussed later. The equations for the curves of Fig. 4 are:

$$\left.\begin{aligned} \frac{1}{\sigma}\int\left(\frac{T-L}{L-aN_0}\right)^2 dt &= \frac{(E-1)^2}{2E} \\ \frac{a^2}{\sigma}\int\left(\frac{N-N_s}{L-aN_0}\right)^2 dt &= \frac{1}{2E} \\ \frac{T_{max}-L}{L-aN_0} &= E-1 \\ \text{(Time constant)}\ \frac{\tau}{\sigma} &= \frac{1}{E} \end{aligned}\right\} \quad (33)$$

The left sides of these equations have been put in dimensionless form. The maximum temperature T_{max} occurs at the beginning of the transient. The time constant τ is that for the controlled system and is shown compared with the engine-time constant σ.

From Fig. 4, the value of λ is chosen as a compromise between the various quantities of equations 33. For $E = 1$ ($\lambda = \infty$), the temperature does not overshoot, the speed integral is 0.5, and $\tau = \sigma$. As E increases (λ decreasing), the temperature integral and the maximum temperature increase, whereas the speed integral and the time constant decrease. A value of $E = \sqrt{2}$ (or $a^2\lambda = 1$) appears to be a good compromise and is used for the subsequent discussion.

The behavior of the system of Fig. 3 can be best seen by drawing the trajectories in the phase plane shown in Fig. 5 where $a\sigma\dot{N}$ is plotted against aN for lines of constant aN_s according to equation 31. Lines of constant temperature are 45 deg parallel lines in this plot and the lines $T = L$ are shown. Each trajectory intersects, and is tangent to, the line $T = L$ at $N = N_s$. Fig. 5 completely describes the transient behavior of the system. For any starting point anywhere on the plane (for instance, point A), the system will automatically move the operating point to that trajectory corresponding to the N_s that exists (point B) and then along this trajectory to the point $T = L$ ($N = N_s$, point C), and finally the stability condition will enter to move the operating point along the solid vertical line to $\dot{N} = 0$ (point D).

The time sense for these transients is obtained by solving the differential equation (equation 31) for the speed and the temperature-time responses. The equations for these solutions are as follows:

$$\left.\begin{aligned} E^2\left(\frac{aN-aN_s}{L-aN_s}\right) &= 1+\sqrt{E^2-1}\ \text{Sinh}\ E\left(\frac{t}{\sigma}+C\right) \\ E^2\left(\frac{T-L}{L-aN_s}\right) &= \sqrt{E^2-1}\ \text{Sinh}\ E\left(\frac{t}{\sigma}+C\right)+ \\ & E\sqrt{E^2-1}\ \text{Cosh}\ E\left(\frac{t}{\sigma}+C\right)+(1-E^2) \end{aligned}\right\} \quad (34)$$

These transients are shown in the general case in Fig. 6 for a step increase in N_s. Maximum tem-

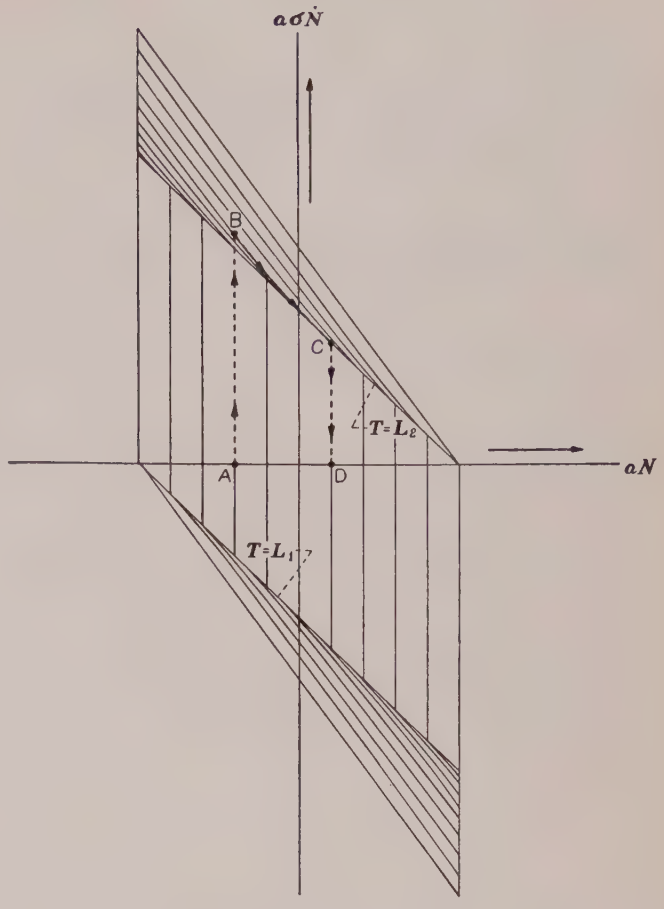


Fig. 5 – Phase plane showing dynamics of controlled system for speed control with temperature-limiting criteria (case B) where $a^2\lambda = 1$

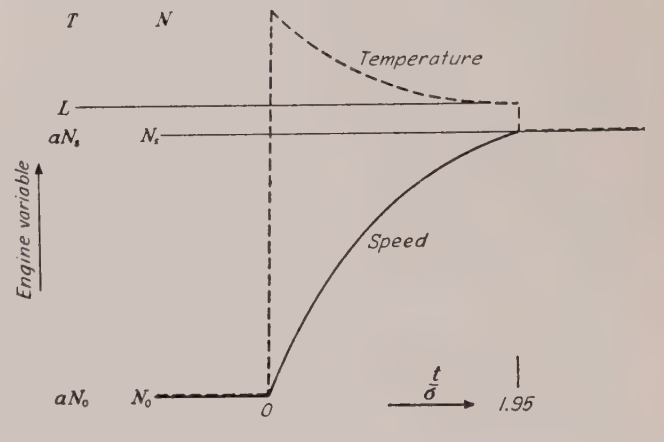


Fig. 6 – Typical transient of controlled system for speed control with temperature-limiting criteria (case B) where $E = \sqrt{2}$

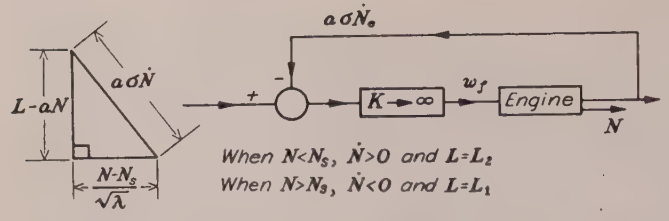


Fig. 7 – Schematic diagram of control for turbojet engine for minimum rise time with temperature-limiting criteria (case C). Stability device must be added to this system

perature occurs at $t = 0$ and the temperature overshoot decreases as N increases, and when $N = N_s$, $T = L$ and $\dot{T} = 0$. The stability condition then causes T to drop to its equilibrium value at this point. The time scale shown in Fig. 6 corresponds to the specific relative values of the ordinates shown.

Minimum Rise Time and Temperature-Limiting Criteria – Case C – In order to obtain a minimum rise time for a constant temperature integral, the requirement is that:

$$\int_0^{t_1} [1 + \lambda f_3(T)]\, dt = \text{Minimum} \tag{35}$$

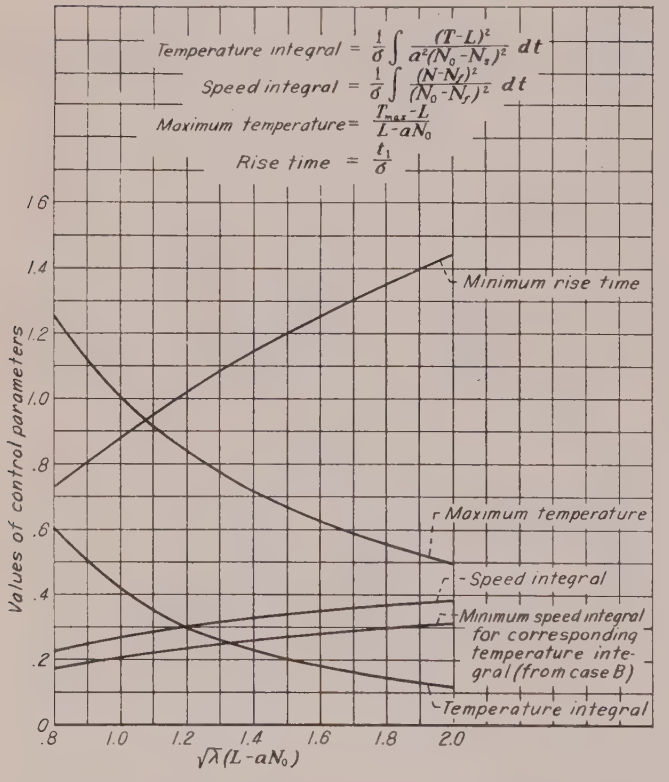


Fig. 8 – Various control parameters as function of λ for minimum rise time with temperature-limiting criteria (case C) when accelerating to limiting temperature

where $F = 1 + \lambda f_3(T)$. Equation 20 becomes:

$$1 + \lambda f_3(T) = \lambda a\sigma\dot{N} f'_3(T) \tag{36}$$

and the ideal stability device is such that:

$$\dot{N} = 0 \text{ when } N = N_s \tag{37}$$

where N_s is the desired stable N. Equations 36 and 37 describe the complete control system. Equation 36, and therefore the control system proper, does not include N_s and, in this case, the stability device is independent of the control system proper.

When the same $f_3(T)$ is used as was used in the previous case, equation 36 can be put in the convenient form:

$$\left.\begin{aligned} &\frac{1}{\lambda} + (L - aN)^2 = a^2\sigma^2\dot{N}^2 \\ &\text{where, for acceleration, when } N<N_s, \\ &\qquad \dot{N}>0 \text{ and } L = L_2 \\ &\text{and, for deceleration, when } N>N_s, \\ &\qquad \dot{N}<0 \text{ and } L = L_1 \end{aligned}\right\} \tag{38}$$

A schematic diagram of this system satisfying equation 38 is shown in Fig. 7. This control system is the same as the previous system (Fig. 3) except that in Fig. 7, $\frac{1}{\sqrt{\lambda}}$ replaces $\frac{N - N_s}{\sqrt{\lambda}}$ on one leg of the right-triangle construction and the stability condition must be imposed outside the control system, as N_s does not enter into the criterion of control (equation 35).

The system of Fig. 7 has one adjustable parameter λ, which, as before, sets the precise values of the integrals entering into the criterion (equation 35), as well as all other behavior characteristics.

The temperature integral is shown in Fig. 8 as a function of λ for the special case where $aN_s = L$; that is, for acceleration or deceleration to the speed that corresponds to limiting temperature. In addition, the rise time, the maximum temperature, and the integral speed-error squared are shown for this transient. A curve showing the minimum speed integral for the corresponding temperature integral (from case B) is also shown and will be discussed later. The equations for the curves of Fig. 8 are as follows:

$$\left.\begin{aligned} &\frac{1}{\sigma}\int_0^{t_1}\left(\frac{T-L}{L-aN_0}\right)^2 dt = \frac{\sqrt{1+\lambda(L-aN_0)^2}}{(L-aN_0)\sqrt{\lambda}} - 1 \\ &\frac{1}{\sigma}\int_0^{t_1} dt = \text{Sinh}^{-1}\left[(L-aN_0)\sqrt{\lambda}\right] \\ &\frac{T_{max}-L}{L-aN_0} = \frac{1}{\sqrt{\lambda}\,(L-aN_0)} \\ &\frac{a^2}{\sigma}\int_0^{t_1}\left(\frac{N-N_f}{L-aN_0}\right)^2 dt = \frac{\sqrt{1+\lambda(L-aN_0)^2}}{2\,(L-aN_0)\sqrt{\lambda}} - \\ &\qquad \frac{1}{2\lambda(L-aN_0)^2}\,\text{Sinh}^{-1}\sqrt{\lambda}\,(L-aN_0) \end{aligned}\right\} \tag{39}$$

Again λ is to be chosen as a compromise between these various quantities. In this case, $\frac{1}{\sqrt{\lambda}}$ has the units of temperature and $\sqrt{\lambda}\,(L - aN_0)$ is the pertinent dimensionless parameter. For a fixed λ, the initial speed N_0 will therefore change the time integrals and an increased $L - aN_0$ will decrease the temperature integral and the maximum temperature and increase the rise time and the speed integral.

The dynamic behavior of the system of Fig. 7 is shown in the phase plane in Fig. 9. This figure is a plot of $a\sigma\dot{N}$ against aN, according to equation 38, for various values of λ. Lines of constant temperatures are 45 deg parallel lines and the lines $T = L$ are shown. For any fixed λ, only two trajectories are followed, one for accelerating and one for decelerating. From any starting point on the plane, the system will automatically move to that trajectory corresponding to acceleration or deceleration and will move along this trajectory until the stability condition enters (at $N = N_s$) to make $\dot{N} = 0$.

The dependence of the time integrals on N_0 may require λ to vary with N_0. In Fig. 9, N_0's corresponding to each λ are shown such as to keep $\lambda(L - aN_0)^2 = 8$. The value of this parameter was so taken as to have the temperature integral in this case equal to that of the previous case for purposes of comparison.

The time sense for these transients is obtained by solving the differential equation 38 for the $N - t$ and $T - t$ transients. The equations for these solutions are:

$$\left.\begin{aligned} \sqrt{\lambda}\,(aN - L) &= \operatorname{Sinh}\left(\frac{t}{\sigma} + C\right) \\ \sqrt{\lambda}\,(T - L) &= e\left(\frac{t}{\sigma} + C\right) \end{aligned}\right\} \qquad (40)$$

These transients are shown for the general case in Fig. 10 for a step increase in N_s. The temperature will jump to some value above L, but in this case, unlike the previous one, the temperature continues to increase as the speed increases. Maximum temperature occurs at the end of the transient. Whenever $N = N_s$ (two such conditions are shown), the stability condition takes over and T is reduced to its equilibrium value. The time scale shown on Fig. 10 corresponds to the specific relative values of the ordinates as shown.

Comparison of Cases B and C – In case B, a system was devised that, for a constant value of the integral temperature-overshoot squared, gives a minimum integral speed-error squared. In case C, a system was devised that, for a constant temperature integral, gives a minimum rise time. For case

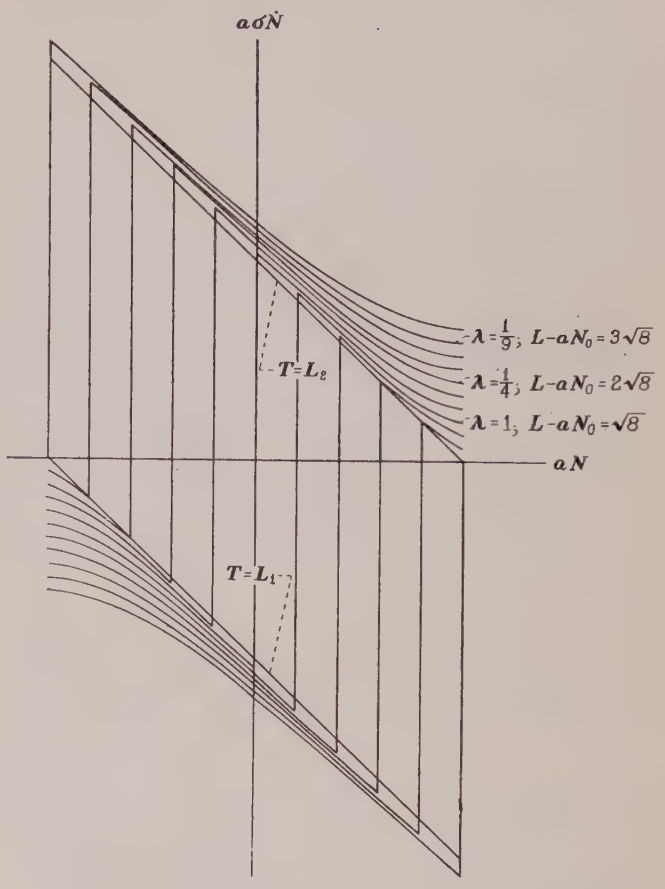


Fig. 9 – Phase plane showing dynamics of controlled system for minimum rise time with temperature-limiting criteria (case C) where $\sqrt{\lambda}(L - aN_0) = \sqrt{8}$

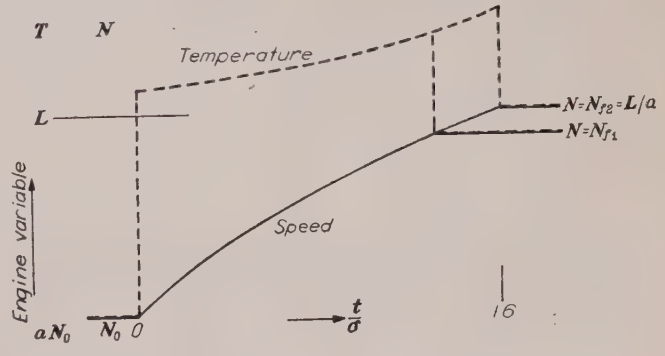


Fig. 10 – Typical transients for controlled system for minimum rise time with temperature-limiting criteria (case C)

B, the rise time is not a minimum but can be compared with the minimum rise time of case C. For the special transient of accelerating to $T = L$, the system of case B reduced to a first-order system, the time constant of which is shown in Fig. 4. Because five time constants are considered as the rise time of an exponential, the time constant of case B should be compared with one-fifth the minimum rise time of case C. In Fig. 4, for corresponding values of the temperature integral, one-fifth the minimum rise time from case C is plotted. Fig. 4 now shows that the rise time for case B is about twice the minimum possible rise time.

At corresponding values of the temperature integral, $\int_0^{t_1} (N - N_f)^2\, dt$ for case C can be compared with the minimum possible value of this quantity from case B. The minimum value of $\int_0^{t_1} (\dot{N} - \dot{N}_f)^2\, dt$ is plotted in Fig. 8. It is seen that, for the same temperature integral, $\int_0^{t_1} (N - N_f)^2\, dt$ when minimum rise time is obtained is about 120% of the minimum possible value of $\int_0^{t_1} (N - N_f)^2\, dt$.

Proof of Absolute Minimum – As previously noted, the minimization of equation 11 involves not only the condition that the first variation be stationary, which leads to equation 20, but also the condition that the second variation be positive. This second condition, however, would prove at most a local minimum for the condition of equation 11. A special proof of an absolute minimum is shown as follows:

If equation 11 is written in the form:

$$\int_{y_0}^{y_f} F(y, \dot{y})\, \frac{dy}{\dot{y}} = \text{Minimum} \tag{41}$$

and $\dot{y}$ is considered as a function of y for the minimum condition and $\dot{y}^*$ is considered as a function of y for any other possible case, then the condition for an absolute minimum is that:

$$\int_{y_0}^{y_f} F(y, \dot{y}^*)\, \frac{dy}{\dot{y}^*} - \int_{y_0}^{y_f} F(y, \dot{y})\, \frac{dy}{\dot{y}} \geq 0 \tag{42}$$

or:

$$\int_{y_0}^{y_f} \left[F(y, \dot{y}^*) - \frac{\dot{y}^*}{\dot{y}} F(y, \dot{y}) \right] \frac{dy}{\dot{y}^*} \geq 0 \tag{43}$$

where $dy/\dot{y}^*$ is a positive differential. If the function F is considered in a general form, quadratic in $\dot{y}$, as:

$$F(y, \dot{y}) \equiv G(y) + \dot{y} G_1(y) + \dot{y}^2 G_2(y) \tag{44}$$

then equation 43 becomes:

$$\int_{y_0}^{y_f} (\dot{y}^* - \dot{y}) \left[\dot{y}^* G_2(y) - \frac{G(y)}{\dot{y}} \right] \frac{dy}{\dot{y}^*} \geq 0 \tag{45}$$

The equation of the control system (equation 20) becomes, for the form of F assumed in equation 44:

$$\dot{y}^2 G_2(y) = G(y) \tag{46}$$

Using this expression for $G(y)$ changes equation 45 to the form:

$$\int_{y_0}^{y_f} G_2(y)\, (\dot{y}^* - \dot{y})^2\, \frac{dy}{\dot{y}^*} \geq 0 \tag{47}$$

For cases B and C, $G_2 = \lambda a^2 \sigma^2$ where $\lambda \geq 0$. Thus an absolute minimum is proved for these two cases.

Degree of Minimum – Case B – The left side of equation 47 can be used as a measure of the deviation from optimum conditions when the methods of this report are not used. If the two cases ($\dot{N}$ for the optimum case and $\dot{N}^*$ for any other case) are compared at the same value of the temperature integral, the left side of equation 47 if the difference between the integral speed-error squared for any case and the minimum possible value of this quantity. The ratio of this deviation to the minimum value becomes:

$$\text{Fractional increase in speed integral} = \frac{2E}{E^2 - 1} \int_{y_0}^{y_f} \left(\frac{T^* - T}{T_f - T_0} \right)^2 \frac{dy}{\sigma \dot{y}^*} \tag{48}$$

for the transient of acceleration or deceleration to limiting temperature. The coefficient of the integral of equation 48 for the value of λ previously chosen ($a^2\lambda = 1$) is $2\sqrt{2}$.

Case B can also be considered as giving a minimum value of the temperature integral for any definite value of the speed integral. If two cases ($\dot{N}$ for the optimum and $\dot{N}^*$ for any other case) are compared at the same value of the speed integral, the left side of equation 45 is proportional to the difference between the integral temperature-overshoot squared for any case and the minimum possible value of this quantity. This deviation can be written:

$$\text{Fractional increase in temperature integral} = \frac{2E}{(E - 1)^2} \int_{y_0}^{y_f} \left(\frac{T^* - T}{T_f - T_0} \right)^2 \frac{dy}{\sigma \dot{y}^*} \tag{49}$$

for the transient of acceleration or deceleration to limiting temperature. The coefficient of the integral of equation 49 for the value of λ previously chosen ($a^2\lambda = 1$) is 16.4.

Degree of Minimum – Case C – If the two cases $\dot{N}$ and $\dot{N}^*$ are compared at the same value of the temperature integral, the left side of equation 47 is the difference between the rise time for any case and its minimum possible value. This deviation can be written:

$$\text{Fractional increase in rise time} = \frac{\lambda (1 - aN_0)^2}{\text{Sinh}^{-1} \sqrt{\lambda}\ (L - aN_0)} \int_{y_0}^{y_f} \left(\frac{T^* - T}{T_f - T_0} \right)^2 \frac{dy}{\sigma \dot{y}^*} \tag{50}$$

for the transient of acceleration or deceleration to limiting temperature. The coefficient of the integral of equation 50 for the value of previously chosen is 4.54. $\sqrt{\lambda}\,(L - aN_0) = 2\sqrt{2}$

Case C can also be considered as giving a minimum value of the temperature integral for any definite value of rise time. If the two cases $\dot{N}$ and $\dot{N}^*$ are now compared at the same value of rise time, the left side of equation 47 is proportional to the difference between the integral temperature-overshoot squared for any case and the minimum possible value of this quantity. This deviation can be written:

$$\text{Fractional increase in temperature integral} = \frac{\sqrt{\lambda}\,(L - aN_0)\int_{y_0}^{y_f}\left(\frac{T^* - T}{T_0 - T_f}\right)^2 \frac{dy}{\sigma\dot{y}^*}}{\sqrt{\lambda(L - aN_0)^2 + 1} - \sqrt{\lambda}\,(L - aN_0)} \qquad (51)$$

for the transient of acceleration or deceleration to limiting temperature. The coefficient of the integral of equation 51 is 16.4 for the value of $\sqrt{\lambda}\,(L - aN_0)$ previously chosen.

General Summary

When the criteria on the behavior of a controlled system can be expressed in certain general forms, as:

$$\left.\begin{array}{l} \int_0^{t_1} f(y)dt \\ \int_0^{t_1} (y - y_s)^2 dt \\ \int_0^{t_1} f_o(z)dt \\ \int_0^{t_1} dt \end{array}\right\} \qquad (8)$$

where the time interval is taken as any duration during which essential external disturbances are constant and during which the system moves from one essential operating point to another, the optimum control can be considered as one that minimizes one of the integrals of equations 8 while maintaining the other integrals at prescribed values. The analytical problem, according to the calculus of variations, reduces to the following equation:

$$\int_0^{t_1} f(y)dt + \lambda_1\int_0^{t_1} (y - y_s)^2 dt + \lambda_2\int_0^{t_1} f_o(z)dt + \lambda_3\int_0^{t_1} dt = \text{Minimum} \qquad (9)$$

For general first-order systems, where $z = z(y,\dot{y})$, equation 9 reduces to:

$$\int_0^{t_1} F(y,\dot{y})dt = \text{Minimum} \qquad (11)$$

where $F = f(y) + \lambda_1(y - y_s)^2 + \lambda_2 f_o(z) + \lambda_3$.

The equation necessary for the control system to satisfy equation 11 and all the boundary conditions becomes:

$$F(y, \dot{y}) = \dot{y}F_{\dot{y}}(y, \dot{y}) \qquad (20)$$

This equation should be followed by the control system proper. In addition, a stability device must be added to the system, the idealized characteristics of which would make $\dot{y} = 0$ when $y = y_s$.

The arbitrary multipliers λ are then found by evaluating the integral criteria involved in F. The transient behavior of the system derived is found by solving the differential equation 20. The degree of the minimum or the amount suffered when any other control system is used was evaluated for the special cases considered. A summary of these developments follows for a special form of the F function, quadratic in $\dot{y}$, where:

$$F(y, \dot{y}) \equiv G(y) + \dot{y}G_1(y) + \dot{y}^2G_2(y) \qquad (44)$$

Control-System Equation – For F in the form of equation 44, the control-system equation 20 becomes:

$$\dot{y}^2 = \frac{G(y)}{G_2(y)} \qquad (52)$$

The function $G_1(y)$ does not affect the control system and the control-system equation gives an explicot expression for $\dot{y}$ as a function of y.

Evaluation of Integrals – If an integral of a function $H(y,\dot{y})$ is any one of the criteria to be considered, it can be evaluated as follows:

$$\int_0^{t_1} H(y, \dot{y})dt = \int_{y_0}^{y_f} H\left[y, \sqrt{\frac{G(y)}{G_2(y)}}\right]\sqrt{\frac{G_2(y)}{G(y)}}\,dy \qquad (53)$$

Thus the integrals can be evaluated without solving the differential equation 52.

Transient Behavior – The differential equation 52 can be easily solved as follows:

$$\int_{x=y_0}^{x=y} \sqrt{\frac{G_2(x)}{G(x)}}\,dx = t - t_0 \qquad (54)$$

Degree of Minimum – Equation 47 was derived to prove the absolute minimum for the special cases considered and can be used to evaluate the degree of the minimum found. Thus:

$$\int_0^{t_1^*} H(y, \dot{y}^*)dt^* - \int_0^{t_1} H(y, \dot{y})dt = \int_{y_0}^{y_f} G_2(y)[\dot{y}^*(y) - \dot{y}(y)]^2 \frac{dy}{\dot{y}^*(y)} \qquad (55)$$

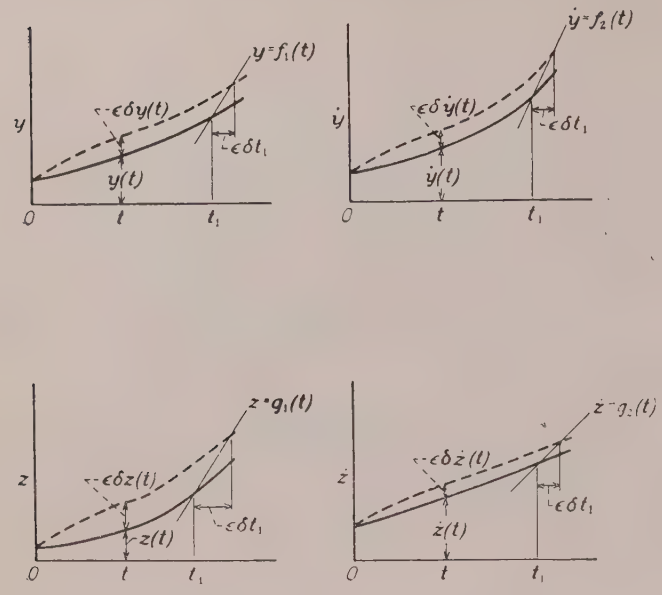


Fig. 11 – Illustration of curves for minimization of

$$\int_0^{t_1} F(y, \dot{y}, \ddot{y}, z, \dot{z}, \ddot{z})dt$$

where:

$$\int_0^{t_1} H(y, \dot{y})dt$$

is to be a minimum or maximum.

If $G_2(y) > 0$ for $y_0 < y < y_f$, then an absolute minimum is obtained; if $G_2(y) < 0$ for $y_0 < y < y_f$, an absolute maximum is obtained. Equation 55 also indicates that the degree of minimum (or maximum) varies with the magnitude of $G_2(y)$. The $\dot{y}^*$ in the denominator indicates that for any deviation $|\dot{y}^* - y|$ in $\dot{y}$, it is better to err on the side of a larger $|\dot{y}^*|$.

Summary of Results

A rational analytic method for the design of automatic control systems was derived. Criteria on the behavior of the controlled system were developed in the form of certain time integrals. When any of these arbitrary but physically realizable criteria were used as a starting point, those equations that the control system must follow were derived. The criteria developed required the minimization of certain time integrals using the calculus of variations. The method gave not only a description of the behavior of the controlled system but also those physically realizable equations that the control system can follow in order to realize this behavior. General results were shown for systems of second order and of any number of degrees of freedom.

APPENDIX

Second-Order Systems of Several Degrees of Freedom

It is beyond the scope of this report to detail the cases of higher-order systems and those having several independent variables. The general equations for these cases will be developed and it is expected that subsequent reports will cover their applications in detail.

For the case of a second-order system with two degrees of freedom, equation 11 becomes:

$$\int_0^{t_1} F(y, \dot{y}, \ddot{y}, z, \dot{z}, \ddot{z})dt = \text{Minimum} \qquad (A1)$$

where y and z are independent functions of time. The condition to satisfy equation A1 is:

$$\frac{d}{d\epsilon}\int_0^{t_1+\epsilon\delta t_1} F(y + \epsilon\delta y, \dot{y} + \epsilon\delta\dot{y}, \ddot{y} + \epsilon\delta\ddot{y}, z + \epsilon\delta z, \dot{z} + \epsilon\delta\dot{z}, \ddot{z} + \epsilon\delta\ddot{z})dt = 0 \qquad (A2)$$

at:

$$\epsilon = 0$$

The time duration of the integral of equation A1 is that beginning at a definite time ($t=0$) but not ending at any definite time but rather along some curves: $y=f_1(t)$, $\dot{y}=f_2(t)$, $z=g_1(t)$, and $\dot{z}=g_2(t)$ (Fig. 11). The functions of δy and δz are arbitrary and independent functions of time.

Performing the operation indicated by equation A2 gives:

$$\int_0^{t_1} (F_y\delta y + F_{\dot{y}}\delta\dot{y} + F_{\ddot{y}}\delta\ddot{y} + F_z\delta z + F_{\dot{z}}\delta\dot{z} + F_{\ddot{z}}\delta\ddot{z})dt + F(t_1)\delta t_1 = 0 \qquad (A3)$$

After integration by parts:

$$\int_0^{t_1}\left[F_y - \frac{d}{dt}(F_{\dot{y}}) + \frac{d^2}{dt^2}(F_{\ddot{y}})\right]\delta y dt + \int_0^{t_1}\left[F_z - \frac{d}{dt}(F_{\dot{z}}) + \frac{d^2}{dt^2}(F_{\ddot{z}})\right]\delta z dt + \left(F_{\dot{y}}\delta y + F_{\ddot{y}}\delta\dot{y} - \frac{d}{dt}F_{\ddot{y}}\delta y\right)\Big]_0^{t_1} + F(t_1)\delta t_1 + \left(F_{\dot{z}}\delta z + F_{\ddot{z}}\delta\dot{z} - \frac{d}{dt}(F_{\ddot{z}})\delta z\right)\Big]_0^{t_1} = 0 \qquad (A4)$$

As before, the integrands of the integrals and the boundary-condition terms must vanish separately. From the geometry at the end condition ($t = t_1$, Fig. 11):

$$\left.\begin{aligned}\delta y(t_1) &= [f_1'(t_1) - \dot{y}(t_1)]\,\delta t_1\\ \delta\dot{y}(t_1) &= [f_2'(t_1) - \ddot{y}(t_1)]\,\delta t_1\\ \delta z(t_1) &= [g_1'(t_1) - \dot{z}(t_1)]\,\delta t_1\\ \delta\dot{z}(t_1) &= [g_2'(t_1) - \ddot{z}(t_1)]\,\delta t_1\end{aligned}\right\} \qquad (A5)$$

The three conditions from equation A4 then become:

$$\left.\begin{aligned}
&F_y - \frac{d}{dt}(F_{\dot y}) + \frac{d^2}{dt^2}(F_{\ddot y}) = 0 \\
&F_z - \frac{d}{dt}(F_{\dot z}) + \frac{d^2}{dt^2}(F_{\ddot z}) = 0 \\
&\left[F - \dot y F_{\dot y} + \dot y \frac{d}{dt}(F_{\ddot y}) - \ddot y F_{\ddot y} - \dot z F_{\dot z} + \dot z \frac{d}{dt}(F_{\ddot z}) - \ddot z F_{\ddot z}\right]_{t=t_1} + \\
&f_1'(t_1)\left[F_{\dot y} - \frac{d}{dt}(F_{\ddot y})\right]_{t=t_1} + f_2'(t_1)\,F_{\ddot y}(t_1) + \\
&g_1'(t_1)\left[F_{\dot z} - \frac{d}{dt}(F_{\ddot z})\right]_{t=t_1} + g_2'(t_1)\,F_{\ddot z}(t_1) + \\
&\delta y(0)\left[F_{\dot y} - \frac{d}{dt}(F_{\ddot y})\right]_{t=0} + \delta\dot y(0)\,F_{\ddot y}(0) + \\
&\delta z(0)\left[F_{\dot z} - \frac{d}{dt}(F_{\ddot z})\right]_{t=0} + \delta\dot z(0)\,F_{\dot z}(0) = 0
\end{aligned}\right\} \quad \text{(A6)}$$

The first two of equations A6 are the differential equations that satisfy the original criteria of equation A1. The physical solution to the problem is the pair of solutions of these equations that satisfies the boundary-condition equation of equations A6. If the first of equations A6 is multiplied by $\dot y$ and the second, by $\dot z$ and the equations are added, an exact derivative is formed, the integral of which is:

$$F - \dot y F_{\dot y} + \dot y \frac{d}{dt}(F_{\ddot y}) - \ddot y F_{\ddot y} - \dot z F_{\dot z} + \dot z \frac{d}{dt}(F_{\ddot z}) - \ddot z F_{\ddot z} = C \quad \text{(A7)}$$

A study of the boundary-condition equation of equations A6 then shows that the physically reasonable boundary conditions should be as follows:

$$\left.\begin{aligned}
&\text{If } F_{\ddot y} \neq 0 \text{ or } F_{\ddot y} \neq K, \text{ then } \delta y(0) = 0 \text{ and } f_1'(t_1) = 0 \\
&\text{If } F_{\ddot y} \neq 0, \text{ then } \delta\dot y(0) = 0 \text{ and } f_2'(t_1) = 0 \\
&\text{If } F_{\dot z} \neq 0 \text{ and } F_{\ddot z} \neq K, \text{ then } \delta z(0) = 0 \text{ and } g_1'(t_1) = 0 \\
&\text{If } F_{\ddot z} \neq 0, \text{ then } \delta\dot z(0) = 0 \text{ and } g_2'(t_1) = 0
\end{aligned}\right\} \quad \text{(A8)}$$

Then in equation A7, $C = 0$ and the final solution to the problem of equation A1 for the boundary conditions of equations A8 is:

$$\left.\begin{aligned}
&F - \dot y F_{\dot y} + \dot y \frac{d}{dt}(F_{\ddot y}) - \ddot y F_{\ddot y} - \dot z F_{\dot z} + \dot z \frac{d}{dt}(F_{\ddot z}) - \ddot z F_{\ddot z} = 0 \\
&\text{and either:} \\
&F_y - \frac{d}{dt}(F_{\dot y}) + \frac{d^2}{dt^2}(F_{\ddot y}) = 0 \\
&\text{or:} \\
&F_z - \frac{d}{dt}(F_{\dot z}) + \frac{d^2}{dt^2}(F_{\ddot z}) = 0
\end{aligned}\right\} \quad \text{(A9)}$$

The meaning of the boundary-condition equations A8 is to define the original criteria for that duration during which the system moves from one essential operating level to another. Thus, if all conditions of equations A8 must hold, the system goes from one definite y, $\dot y$, z, and $\dot z$ to any other definite y, $\dot y$, z, and $\dot z$. The first differential equation of equations A9 would be of third order and the second or third equations of A9 would be of fourth order. If equations A9 are integrated to obtain a pair of second-order differential equations having three constants of integration, the choice of these three constants can then determine a desired end point, that is, the values of $\dot y$, z, and $\dot z$ at some final y.

The physical solution to the problem, then, is the pair of second-order differential equations that are solutions of equations A9 with the constants of integration evaluated so that the system goes through some desired end point. This end point must be such as to allow the possibility of stability. Such an end point may be written as follows:

$$\dot y = 0,\ z = z_s, \text{ and } \dot z = 0 \text{ when } y = y_s \quad \text{(A10)}$$

which gives three conditions for the evaluation of the three constants of integration. A stability device must still be added to the system so that, at the point when equations A10 hold:

$$\left.\begin{aligned}
&\text{and:} \quad \ddot y = 0 \\
&\qquad\quad \ddot z = 0
\end{aligned}\right\} \quad \text{(A11)}$$

Equations A9 are symmetric in the variables y and z, which indicates the nature of the extension for more independent variables. Thus, if a third independent function w exists, the original criterion would be written:

$$\int F(y, \dot y, \ddot y, z, \dot z, \ddot z, w, \dot w, \ddot w)dt = \text{Minimum} \quad \text{(A12)}$$

This condition is satisfied under boundary conditions similar to equations A8, where two additional conditions are added on the variable w, and the following equations describe the controlled system:

$$\left.\begin{aligned}
&F - \dot y F_{\dot y} + \dot y \frac{d}{dt}(F_{\ddot y}) - \ddot y F_{\ddot y} - \dot z F_{\dot z} + \dot z \frac{d}{dt}(F_{\ddot z}) - \\
&\qquad \ddot z F_{\ddot z} - \dot w F_{\dot w} + \dot w \frac{d}{dt}(F_{\ddot w}) - \ddot w F_{\ddot w} = 0 \\
&\text{and any two of the following three equations:} \\
&F_y - \frac{d}{dt}(F_{\dot y}) + \frac{d^2}{dt^2}(F_{\ddot y}) = 0 \\
&F_z - \frac{d}{dt}(F_{\dot z}) + \frac{d^2}{dt^2}(F_{\ddot z}) = 0 \\
&F_w - \frac{d}{dt}(F_{\dot w}) + \frac{d^2}{dt^2}(F_{\ddot w}) =
\end{aligned}\right\} \quad \text{(A13)}$$

The three equations of equations A13 can then be integrated to give three second-order differential equations where the five constants of integration are evaluated so that the system goes through the desired value of $\dot y$, z, $\dot z$, w, $\dot w$ for some final value of y.

WIND EFFECTS ON

SIDEWINDS and gusts affect automobile stability to an important degree. The determination of air resistance of automobiles by road tests and by wind-tunnel tests has not been infrequent. However, comparatively little attention has been given the problem of the effects of sidewinds and gusts on the stability of cars. This quantitative analysis of the behavior of a single automobile traveling on the highway in extremely gusty weather was made with the purpose of defining more clearly some of the aspects of the wind-stability problem.

All of the work was done at the University of Michigan under the sponsorship of the Engineering Research Institute. A unique test procedure was proved workable. The body of a test car was made a separate unit, and forces transmitted from body to chassis were measured during highway tests. Reaction of the test car to sidewinds is depicted by: (a) the variation in location on the car of the center of pressure of the wind, (b) the variation of the total sidewise force on the car, and (c) the variation of the yawing moment of the car. The transient effects of a gust striking a moving automobile are analyzed also.

The resultant wind striking a moving automobile will be considered the vector sum of the natural wind and the wind over the car due to its motion. The point of application along the car's longitudinal axis of a single force equal to the thrust on the car exerted by the resultant wind will be referred to as the center of pressure. The magnitude and direction of the resultant wind is the *cause* dealt with in this investigation; the reaction of the automobile, the *effect*.

Winds striking a moving car at some angle inclined to the direction of travel produce resisting

BEHAVIOR of a single automobile traveling on the road in windy weather provided the data analyzed in this paper. This quantitative analysis was intended to define the effects of sidewinds and gusts on the stability of cars. Highway tests were made with a car in which the body functioned as a separate unit, so that forces transmitted from body to chassis could be measured.

One study was made of the transient effects of a gust striking a moving automobile. The other study was concerned solely with the reaction of the test car to sidewinds. The authors conclude that body design is the key to aerodynamic stability.

The Authors

W. E. LAY (M '19) is professor of mechanical engineering in charge of the automotive laboratory at the University of Michigan. He continued on the faculty there after his graduation, but during the years has collaborated on experimental test and research work for Packard, Chevrolet, General Motors, NACA, and other similar groups. During World War II he conducted special training courses in testing aircraft and tank engines.

P. W. LETT, JR., (M '51) is project engineer for ordnance development at Chrysler Corp. At present he is occupied with the design and development of certain components for a new medium tank. He has a B. S. degree in mechanical engineering from Alabama Polytechnic Institute, an M. S. from the University of Alabama, and a Ph.D. in mechanical engineering from the University of Michigan.

CAR STABILITY

W. E. Lay and P. W. Lett, Jr.
University of Michigan Chrysler Corp.

Paper presented at SAE Nat'l Passenger-Car, Body, and Materials Mtg., Detroit, March 3, 1953.

forces at the front and rear wheels. The resisting force between tire and road surface at each wheel may be resolved into two component forces, one acting parallel and the other perpendicular to the car's longitudinal axis. The distribution of lateral forces between front and rear wheels determines the tendency of the vehicle to yaw. The location of the center of pressure with respect to the center of gravity of the car determines the distribution of lateral forces between front and rear wheels. The magnitude of the yawing moment about the center of gravity depends not only on the distance between the center of pressure and the center of gravity but also on the magnitude of the lateral force exerted on the vehicle. The body shape of the car determines: (a) its lateral as well as its longitudinal drag, and (b) the location of the center of pressure for any particular resultant wind.

Perhaps the problem of sidewinds and automobile directional stability has been given most attention by the Research Institute for Vehicle Behavior, Stuttgart, Germany. One series of wind-tunnel tests was conducted at the institute using a free-running model car, on a moving belt simulating a road surface in the wind tunnel. A wire was attached to the model at its center of gravity. The other end of the wire was fixed to the wind tunnel upstream of the model, thus locating the model on the moving belt. The center of gravity of the model could be changed as desired, and various plastic bodies were fitted on the model chassis in the course of the tests. The behavior of the model when subjected to sidewinds was recorded by a camera suspended above it. Results indicated that most current body designs were affected adversely by sidewinds due primarily to the center of pressure location. It was shown, however, that the car body could be designed to distribute sidewind forces equally between front and rear wheels, reducing to a minimum yawing moments caused by sidewinds. An experimental automobile was constructed and tested with satisfactory success on the basis of these wind tunnel test results. The instability of teardrop body shapes in crosswinds was made evident in other tests. Also, a thorough study was made of the stabilizing effects of fins on various body shapes.

The investigation which was conducted at the University of Michigan sacrificed controlled conditions available in a wind tunnel for the opportunity of measuring reactions experienced in road testing. Confinement of the tests to a single car was a limitation of the investigation. However, the magnitude of the instrumentation job in this case made it impractical to test a number of cars. A rather complete quantitative analysis of the behavior of a single full-size automobile under actual road conditions was deemed preferable to wind-tunnel testing.

Aerodynamic Stability

Aerodynamic stability is an important part of the total stability of a car. It is known that moments act on free bodies suspended in an air stream as soon as their longitudinal axis forms an angle with the stream direction. Consider an airplane without rudder – the moments would tend to turn the plane crosswise to the wind direction. The behavior of the plane would be considered unstable.

If stabilizing surfaces were added to the tail of the plane there would be created a turning moment tending to align the longitudinal axis of the plane with the stream flow. This behavior of the plane we term stable.

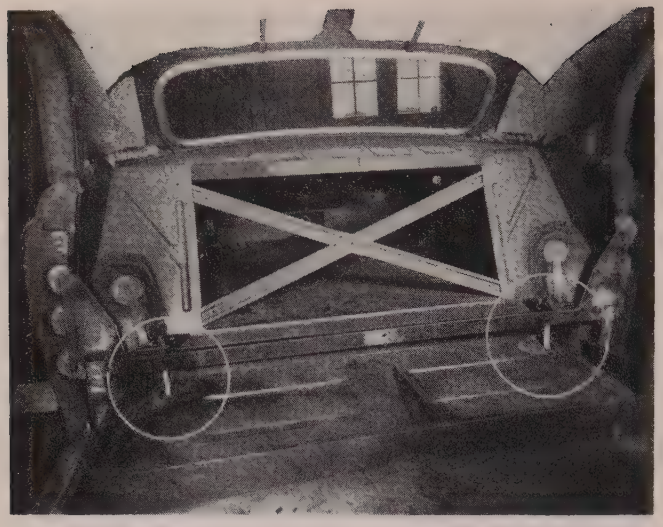


Fig. 1 – Rear body supports

Aerodynamic stability as it applies to automobiles is different in a number of respects from the aerodynamic stability of aircraft. The forward speed of an aircraft is normally such that resultant winds moving over its surfaces are practically coincident with the longitudinal axis. Due to the much lower average speed of automobiles the sidewinds that strike a car on a windy day produce resultant winds that vary in angle considerably more than those striking aircraft. Regardless of the magnitude or direction of the sidewind the tendency of the automobile to yaw should be minimized for stable performance. For aircraft, stability infers the tendency to align with the stream flow; for cars, a negligible tendency to yaw when subjected to sidewinds.

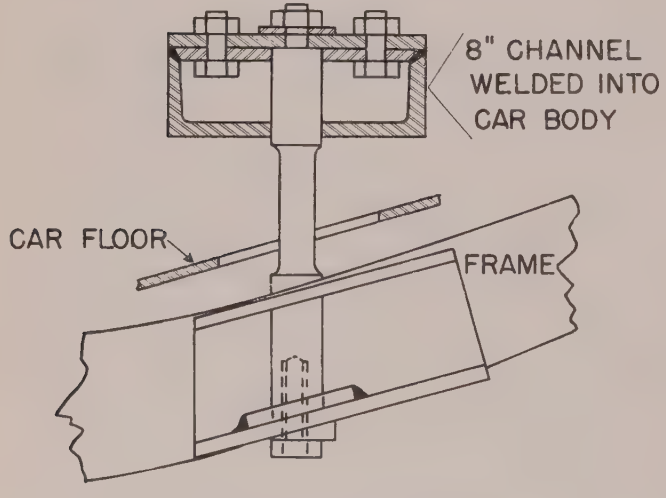


Fig. 2 – Left rear support column

Although aerodynamic stability is the concern of this investigation other factors affecting total car stability should be mentioned. Some of the more important factors are:

1. Tire inflation. Low-pressure front tires tend to cause understeering.
2. Weight distribution between front and rear axles.
3. Front-wheel linkage. Independently sprung front wheels with rigid articulating linkages cause a change in toe-in under roll conditions that affects directional control.
4. Rear-end suspension. Where Hotchkiss suspension is employed rear-end steering may result.
5. Steering mechanism.

Separation of Body and Chassis

Analyzing the reaction of a moving car to winds required the measurement of forces involved. The thrust on an automobile body due to winds is transmitted through the body mounts and other connections to the frame, and hence to wheels and tires. The method employed to measure these forces in these tests involved the separation of body and chassis, with the installation of instrumentation between the two. Forces in a horizontal plane were measured by these devices. Forces were measured at four points along the frame, the four points forming the corners of a rectangle approximately 60 in. by 80 in. A new 1947 automobile, mechanically complete but lacking trim, upholstery, and seats, was obtained from an automotive manufacturer. An 8-in. steel channel was welded crosswise into the body just forward of the rear wheels as shown in Fig. 1.

At either end of the channel a square hole was cut and a ¾-in. square steel column fitted in a vertical position. Each column, approximately 12 in. long, was attached to the channel at the upper end and extended through a hole in the floor to the frame underneath. The column on either side was rigidly attached to the frame by fitting into a short channel-section which was welded to the frame with its web vertical. Figs. 1 and 2 show the columns in position.

The method of supporting the front of the body, once separated from the frame, was a more difficult problem. Here it was necessary to construct a subframe for either side of the body. The subframe was attached to the body at the three forward body support points. The upper end of a ¾-in. square metal column was fitted into the subframe on either side. The lower end of these columns was fitted into a box-section transverse member rigidly attached to the underside of the frame. Fig. 3 shows this arrangement.

With a support system provided for the body as described, separation of body and chassis was possible. This required removal of body bolts used to attach body to frame, removal of plates in the

floorboard around controls, replacement of braces from radiator to bulkhead by braces from radiator to frame, and pendulum-like support of the front-end sheet metal (grill, fenders, and hood) from the top of the radiator. The latter step was necessary because of the lack of rigidity of the front end, without benefit of the support of the front crossmember of the frame. The body was elevated approximately 1/8 in. above its normal position on the chassis when carried by the support columns. In addition safety bolts were installed to prevent separation of body and chassis in case of failure of the support columns. The safety bolts permitted body movement only to the limit required for the tests.

Instrumentation and Calibration

The method employed to support the body of the test car allowed determination of the car's reaction to winds by measurement of forces transmitted from body to frame through the four support columns. Wind forces acting on the body subjected the columns to bending strains. These strains, directly proportional (within the elastic limit) to applied forces, were measured by electric resistance strain gages. It was thus possible to calibrate the system to indicate applied force directly. Strain sensitivity of support columns was increased by a reduction of cross-sectional area. Fig. 4 shows strain gages attached to a support column.

Gages on each support column were wired to form two Wheatstone bridge circuits. Gages A, B, C, and D, sensitive to lateral bending forces, were wired as shown in Fig. 5.

Gages F and G were used to measure bending strains induced by forces applied parallel to the longitudinal axis of the car. Gages F and G were wired in a bridge circuit similar to that shown in Fig. 5, except that 500-ohm resistors were used in the two arms of the bridge adjacent to the voltage divider. Forty-six volts direct current was applied to each bridge. Bridge unbalance was indicated by microammeters having a sensitivity great enough to require no amplification of the signal from the gages. The instrumentation diagram for the bridge circuits is shown in Fig. 6, and the wiring diagram in Fig. 7.

Two instruments attached to the front of the test car were required for the measurement of the speed and direction of winds striking the car (see Fig. 8). A bridled anemometer was used to measure wind speed. This instrument had to provide an accurate indication of fluctuating wind speeds regardless of wind direction. The anemometer was constructed as a wind torque meter, where the angular displacement of the anemometer rotor was a function of wind speed. The angular movement of the rotor was transmitted electrically (by self-synchronous motors) to a pointer moving over a dial calibrated in mph.

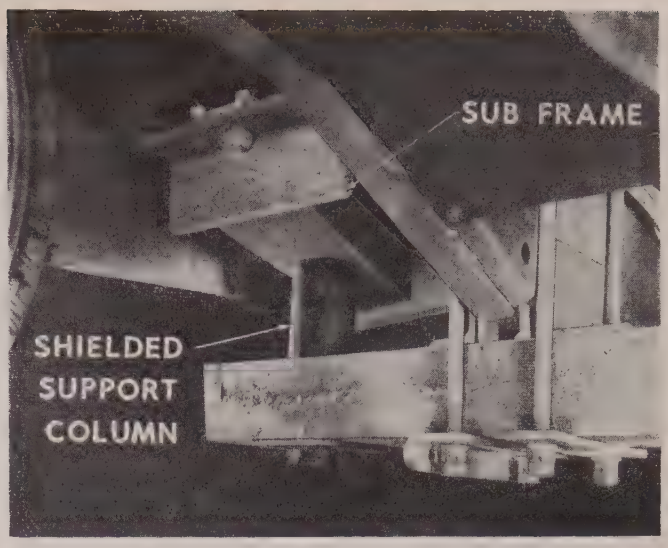


Fig. 3 – Right front body support

A vane was required to indicate the direction of winds striking the test car. Sensitivity to gusts was a necessary specification for this instrument. The wind indicator was constructed by mounting two vertical curved blades of sheet aluminum equidistant from a pivot shaft as shown in Fig. 8.

This construction insured a small moment of inertia. At each blade a wind torque was produced which opposed the torque acting at the other blade, providing an underdamped system with an inherent high sensitivity. The angular position of the vane

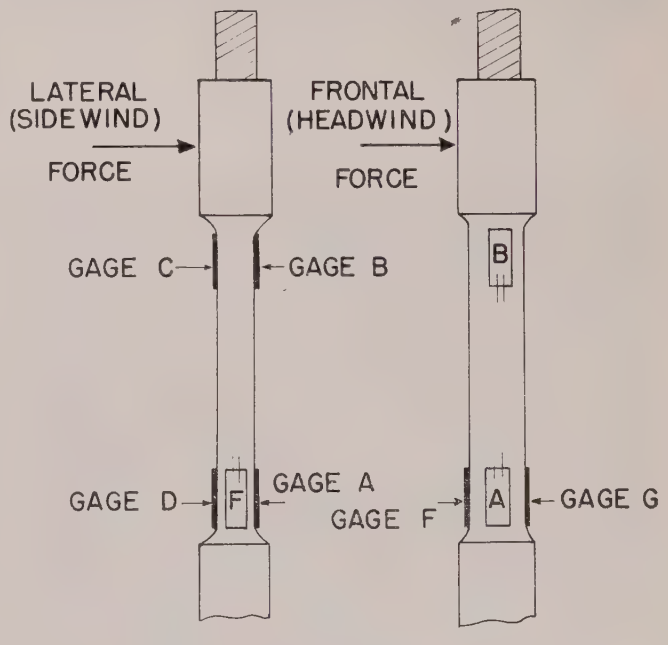


Fig. 4 – Gages on support column

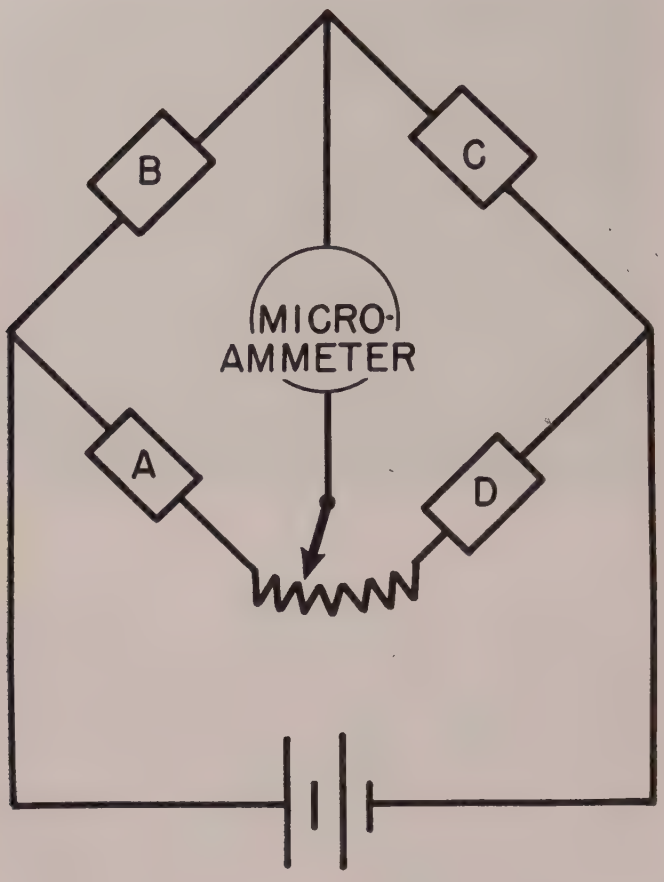


Fig. 5 – Lateral force bridge circuit

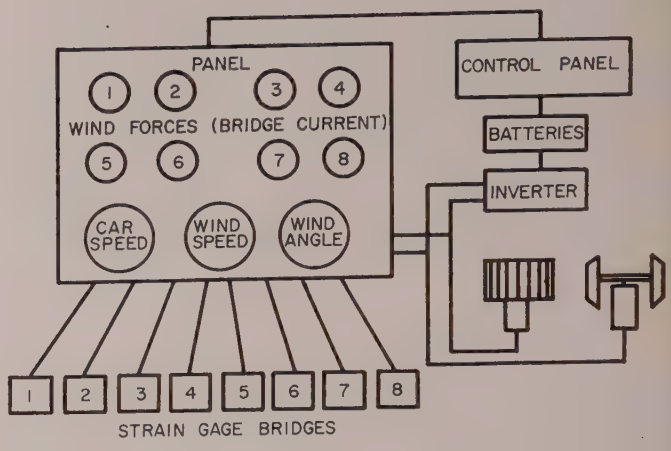


Fig. 6 – Instrumentation diagram

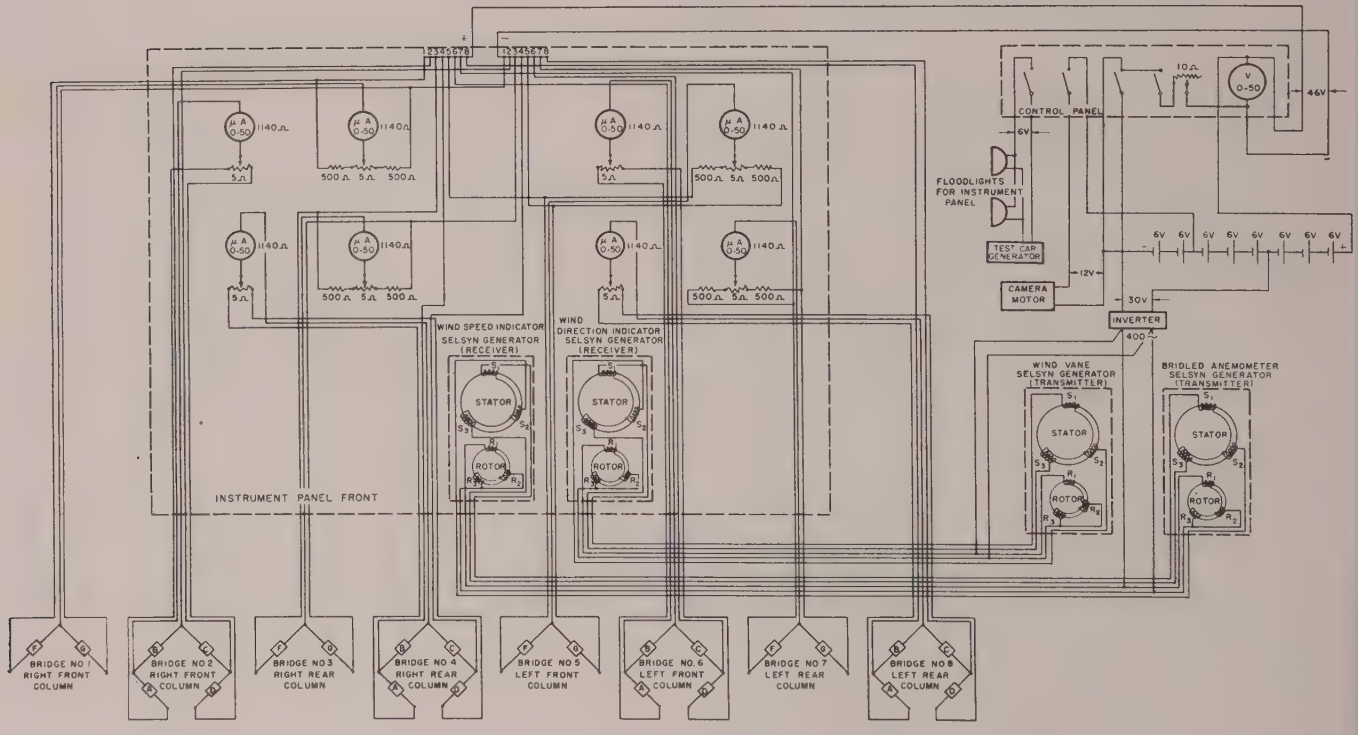


Fig. 7 – Instrumentation wiring diagram

was transmitted electrically to a dial and pointer within the car. The wiring diagram of the wind-instrument circuit is included in Fig. 7.

Microammeters, wind-speed and direction indicators, a precision speedometer, and clock were installed on a shock-mounted panel within the test car as shown in Fig. 9.

A small control panel was mounted to the right of the driver's seat in the test car. Switches to control the wind instrument circuit, the strain-gage bridge circuits, the floodlights for illuminating the instrument panel, and the camera motor were located here. The wiring diagram of the control panel is shown in Fig. 7. Fig. 10 shows a pictorial view of the panel.

For test purposes it was desired to have the weight of the test car with the driver equal to that of a standard production car with five passengers. Ballast was added and batteries positioned in the vehicle to obtain the proper weight distribution at each wheel (see Fig. 11).

Two tests were required for the static calibration of strain-gage bridge circuits. The first test consisted of applying incremental lateral forces to the car body and recording the resulting bridge unbalance. Forces were applied from left to right, then right to left as shown in Fig. 12.

The second calibration test for the strain-gage bridges made use of the weight of the body to exert lateral bending forces on the support columns. The procedure followed was to elevate one side of the car in increments by placing blocks under the front and rear wheels on that side. Knowing the weight of the body and the angle of tilt the component of the body's weight acting to bend the support columns was determined. Bridge unbalance was recorded for all forces applied.

From the lateral force and the body-tilt calibration tests, factors were determined for converting meter readings of bridge current into force. For the bridges on the front columns sensitive to lateral forces this factor was designated K_f. Similarly, K_r applied to bridges on the two rear columns sensitive to lateral forces. These conversion factors were determined to be:

$$K_f = 4.32 \text{ lb per microamp}$$
$$K_r = 3.43 \text{ lb per microamp}$$

Installation of the body support columns, so that the sides of the columns were exactly parallel to the longitudinal axis of the car, was impossible. Thus forces parallel to the longitudinal axis could cause some lateral bending strains when support columns were not positioned perfectly; strain gages sensitive to lateral forces would indicate some side force where none existed. A dynamic calibration of the bridge circuits was carried out to determine how much effect the support-column positioning had on bridge current with no side winds striking the car. With negligible natural wind the unbalance of bridges sensitive to lateral forces is shown as a function of car speed in Fig. 13.

Fig. 8 – Bridled anemometer and wind vane

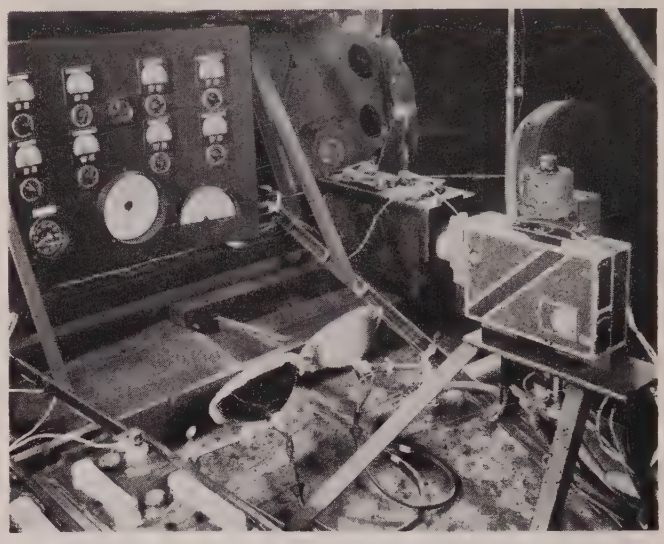


Fig. 9 – Instrument panel and camera

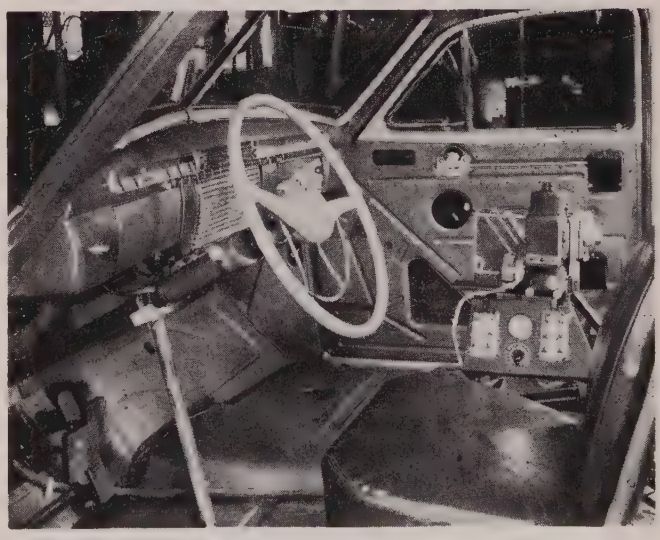


Fig. 10 – Control panel and camera

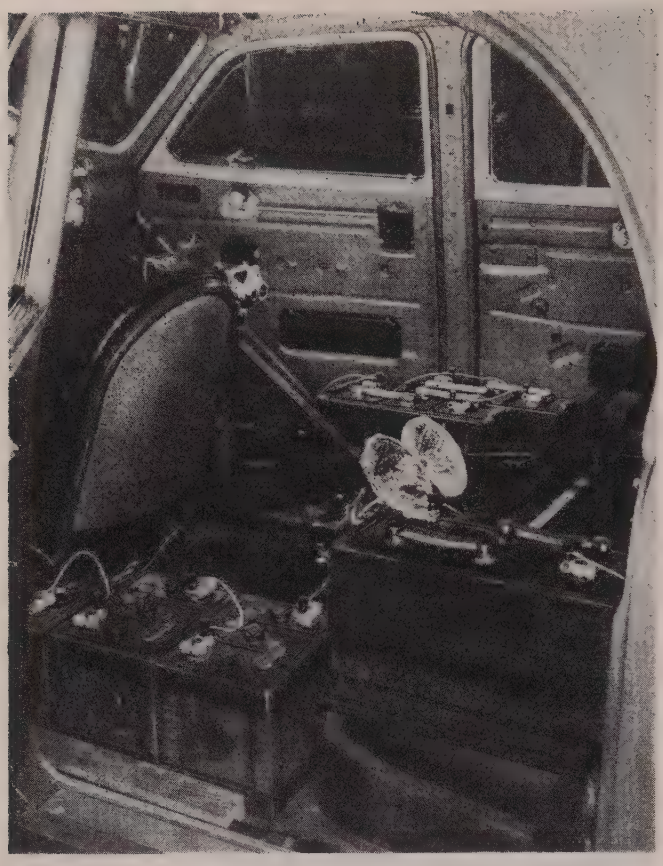


Fig. 11 – Batteries in test car

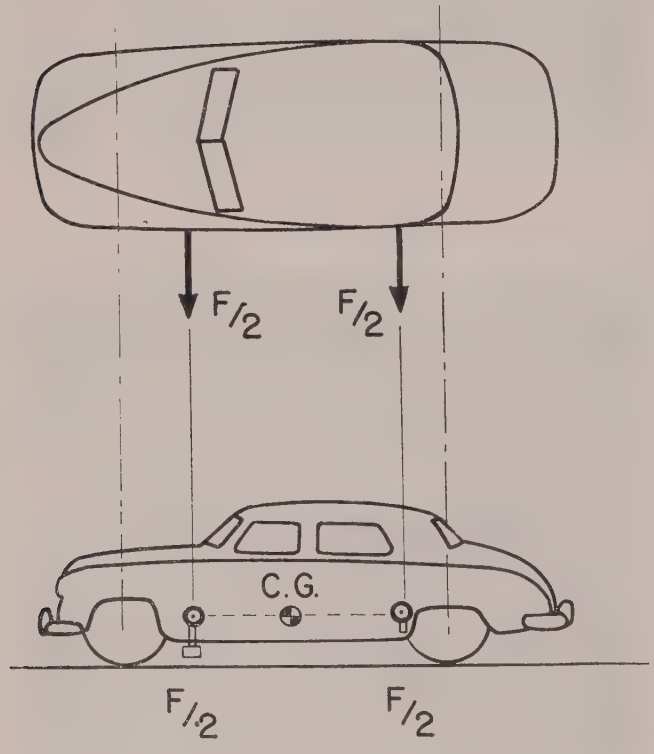


Fig. 12 – Calibration of strain gages by lateral forces

The curves of Fig. 13 served as datum lines for determining bridge unbalance due to externally applied forces.

Calibration of the instruments for measuring wind speed and direction was accomplished by road tests and by wind-tunnel tests.

The determination for the test car of the cross-sectional area normal to winds from various directions was considered essential to an analysis of the car's behavior. This was done photographically. Fig. 14 shows the variation of cross-sectional area with change in horizontal angle.

Over 1000 miles of road testing was carried out. Procedure was established with the objective of isolating forces due to winds as much as possible. An attempt was made to avoid centrifugal forces by running tests only while the vehicle was traveling in a straight direction. The car was driven in the center of the road for the interval when data was being recorded to prevent the crown of the road from affecting readings. Areas for testing were sought where a variety of conditions were available – roads running through alternate wooded areas and open fields, areas where buildings were located near the road, and where there were long stretches with no obstructions on either side of the road. Once a section of straight level road had been located the test car was driven over it in both directions at various speeds held constant during each run. Data was recorded photographically during each run.

Sidewinds

It was necessary to distinguish between *sidewinds* and *gusts* before an analysis of wind effects on car behavior could be made. Wishing to indicate fairly steady-state conditions by the term sidewinds, the problem became one of defining limits of variability. When does the speed and direction of a wind vary sufficiently in a given time interval to become a gust instead of a sidewind? The answer to this question was sought in the effects of winds on the moving vehicle. As described previously four road tests for calibrating purposes were made with natural winds practically nil (3 mph or less). With no sidewinds acting, very near zero side force was indicated by the meters in the bridge circuits sensitive to lateral forces. There was a slight dispersion in the readings of these meters due to minor extraneous causes such as irregularities in the road surface, steering effort, and so forth. The range of dispersion of meter readings without sidewinds or gusts was established from these four calibration road tests.

From statistical analysis a measure of the dispersion of a set of numbers can be obtained by finding the standard deviation from the average value. The standard deviation (σ) of a set of n numbers, X_1, X_2, . . . X_n, is defined as the square root of the average of the squares of the deviations

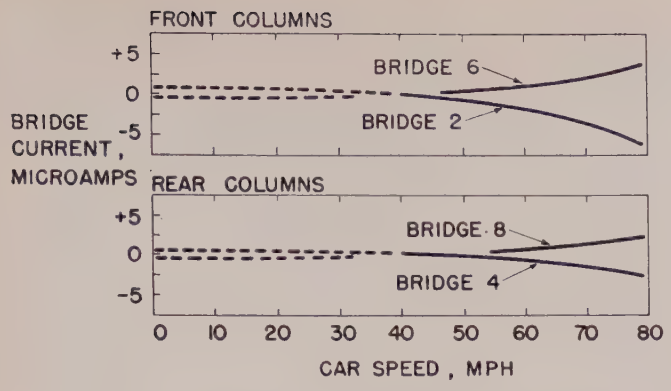


Fig. 13 – Bridge current with natural wind negligible

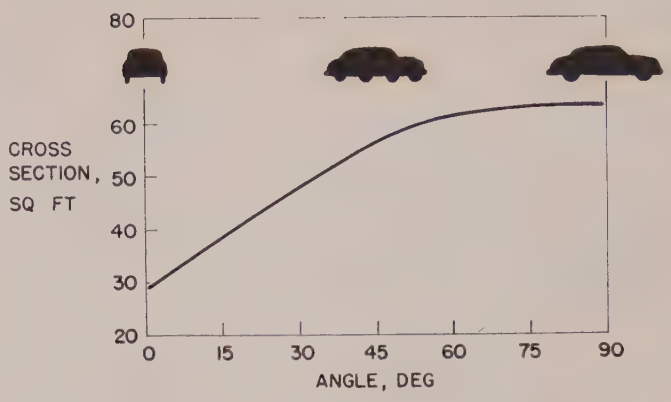


Fig. 14 – Car cross-section versus horizontal angle

of the numbers from their arithmetic mean, $\overline{X}$.

$$\sigma = \left[\frac{(X_1 - \overline{X})^2 + (X_2 - \overline{X})^2 + \cdots + (X_n - \overline{X})^2}{n} \right]^{\frac{1}{2}}$$

$$\sigma = \left[\frac{\sum_{i=1}^{n} X_i^2}{n} - \overline{X}^2 \right]^{\frac{1}{2}}$$

The standard deviation of the meter readings taken during the calibration tests was then calculated. From the law of probability 99.73% of a group of readings could be expected to fall within a band of $\pm 3\sigma$. Thus when more than 0.27% of the readings taken during road tests fell outside the 3σ limits causes other than chance were responsible, such as changing winds. By applying the $\pm 3\sigma$ limits determined from the calibration tests to all the data taken during road tests, it was possible to separate those portions of the data where fairly steady winds (sidewinds) were acting. The data from all road tests were divided into groups representing approximately 8 sec of testing. After discarding those readings outside the 3σ limits, average values of bridge current for each bridge, car speed, resultant wind speed, and direction were obtained. Sidewind-induced lateral forces transmitted through support columns were then calculated, using the conversion factors K_f and K_r previously described. The total sidewise force was found by taking the algebraic sum of lateral forces at front and rear columns. The yawing moment was determined by taking the sum of the moments of the lateral forces about the center of gravity of the vehicle. The yawing moment was equivalent to the product of the total sidewise force and the distance between the center of pressure and the center of gravity. From this relationship the location of the center of pressure was determined for each test interval.

Car reaction to sidewinds was defined quantitatively by the sidewise force, the location of the center of pressure with respect to the center of gravity, and the yawing moment created by a sidewind. The *cause* of these reactions was considered to be the resultant wind striking the car. The resultant wind was used here since it represented the vector sum of the sidewind and the wind over the car due to its motion.

For a specific resultant wind angle the sidewise force exerted on the test car was shown to grow

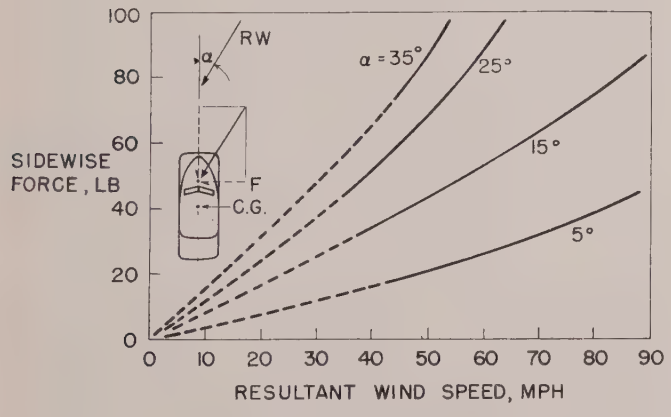


Fig. 15 – Sidewise force versus resultant wind speed

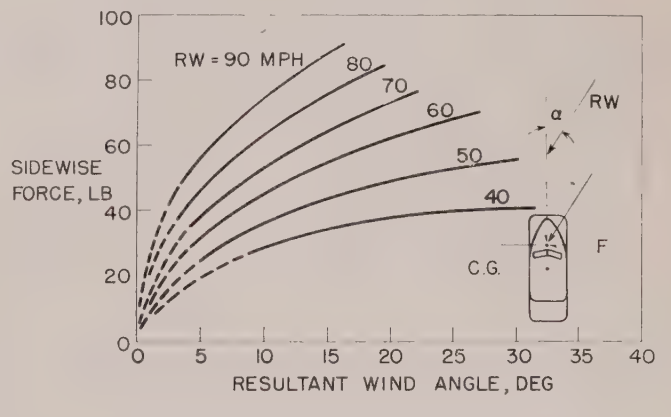


Fig. 16 – Sidewise force versus resultant wind angle

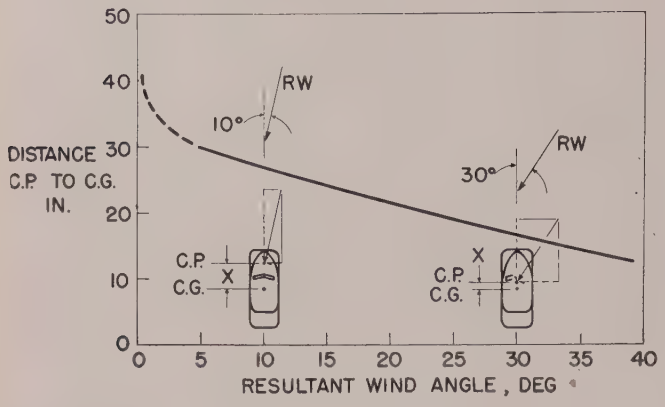


Fig. 17 – Center of pressure location versus resultant wind angle

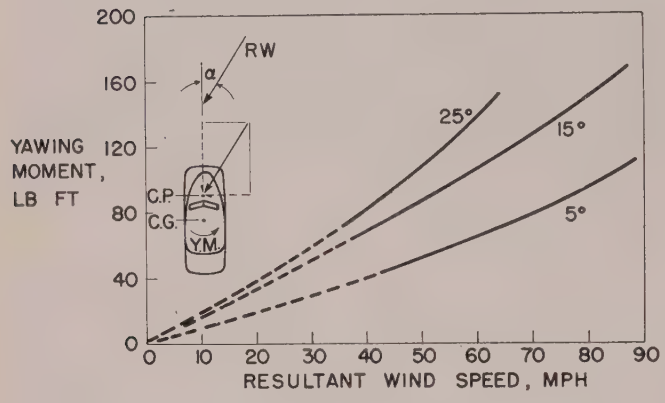


Fig. 18 – Yawing moment versus resultant wind speed

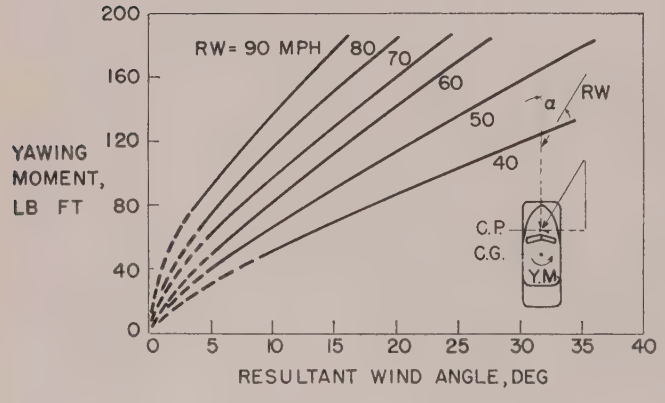


Fig. 19 – Yawing moment versus resultant wind angle

larger at an increasing rate as the resultant wind speed became greater (see Fig. 15). This could be expected since wind resistance varies as the square of the wind velocity. Also, it is to be noted that sidewise forces recorded in these tests reached rather significant values. Forces of 90 lb were recorded, which is equivalent to the frontal air resistance of a car with a coefficient of air resistance of 0.0015 lb per ft²mph² traveling at 45 mph in still air. (The coefficient of air resistance of a 1947 Chevrolet is approximately 0.0015 lb per ft²mph².) It is to be remembered that this is a steady-state condition – momentary sidewise forces due to gusts reached much higher values. With a sidewind striking the car the direction of travel was maintained by the lateral tractive forces between tires and road surface, created by the wheels running at some slip-angle.

The variation of sidewise force with a change in angle of resultant wind is shown in Fig. 16. With a change in direction of the resultant wind (wind speed constant) the sidewise force might be expected to increase in direct proportion to the cross-sectional area of the car. However, test results indicated that the total sidewise force increased at a more rapid rate than the cross-sectional area normal to the resultant wind. This was most noticeable at small resultant wind angles. The fact that resistance is a function of shape as well as cross-section normal to flow was responsible for this happening. For example, the increase in cross-sectional area normal to the wind was relatively small as resultant wind direction changed from 0 deg to 10 deg, indicating a correspondingly small increase in sidewise force. (Zero degrees indicates a head-on resultant wind.) However, resistance due to turbulence increased rapidly with this change in direction of 10 deg. The increase in resistance due to turbulence was found to be most significant where the direction of resultant wind was less than 10 deg.

The location of the center of pressure of a resultant wind striking the car varied with the angle of the wind, but was unaffected by changes in wind speed. As shown in Fig. 17, the location of the center of pressure for a resultant wind angle 45 deg from the direction of travel was approximately 10 in. forward of the center of gravity. With a resultant wind angle 5 deg from the direction of travel the center of pressure was found to be 30 in. forward of the center of gravity.

The greater the distance between center of pressure and center of gravity, the greater the tendency to yaw, and the poorer the aerodynamic stability of the car. The center of pressure location for winds from any angle is dependent upon the body shape of the car. In a side view, the larger the percentage of cross-sectional area forward of the center of gravity, the farther forward the center of pressure will be. For example, a teardrop body shape would have a center of pressure located

quite far forward of the center of gravity. Wind-tunnel tests bear out the fact that aerodynamic stability would be difficult to obtain with this body shape without the addition of stabilizing surfaces to the rear of the car. In general, body design that results in center of pressure location near the center of gravity results in improved aerodynamic stability.

For a particular resultant wind angle the yawing moment varied with changes in wind speed exactly as did the sidewise force (see Fig. 18). This was true since the location of the center of pressure changed only with a shift in wind direction. Non-intermittent yawing moments in excess of 160 lb-ft were recorded during road tests.

As resultant wind direction changed from head-on to some angle from the side, the tendency of the car to yaw was increased greatly. This is illustrated in Fig. 19. The rate of increase of the yawing moment was greatest at small resultant wind angles because of the forward location of the center of pressure. As the wind direction became more nearly broadside the yawing moment continued to increase because of increasing sidewise force. However, because the center of pressure moved closer to the center of gravity the rate of increase in yawing moment diminished. Improvement of the aerodynamic stability of present-day cars requires a reduction of the tendency to yaw when subjected to sidewinds.

Gusts

Important differences exist in the reaction of a car to a sidewind and the reaction to a gust. In the former case a direction of travel is maintained by causing the wheels to run at some slip-angle which for all practical purposes is constant so long as wind and direction of travel do not change. In contrast, the sudden lateral thrust imposed on a car by a gust causes abruptly a sidewise displacement of the vehicle and a tendency to yaw. When this happens the driver immediately exerts corrective steering, turning the front wheels so that the tractive forces between wheels and road will counteract the wind-induced yawing moment. Perhaps he overcorrects – the car momentarily yaws in the opposite direction, and so forth, until a condition of relative stability is secured.

In road tests there is the opportunity to measure the total of what the car feels. Resolution of forces into their respective components was impossible in this case, however, due to the inability to measure the percentage of total sidewise force attributable to the steering efforts of the driver. Thus the total reaction of the test car to gusts necessarily included driver-induced forces and moments. The reaction to gusts is illustrated by the following example taken from the test data.

The car was traveling at 64 mph at the beginning of the time interval in which the gust acted. The resultant wind was between 80 and 85 mph,

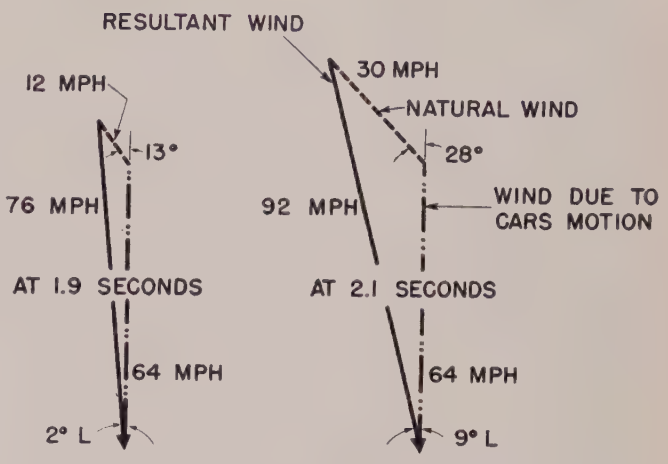


Fig. 20 – Wind vector diagram

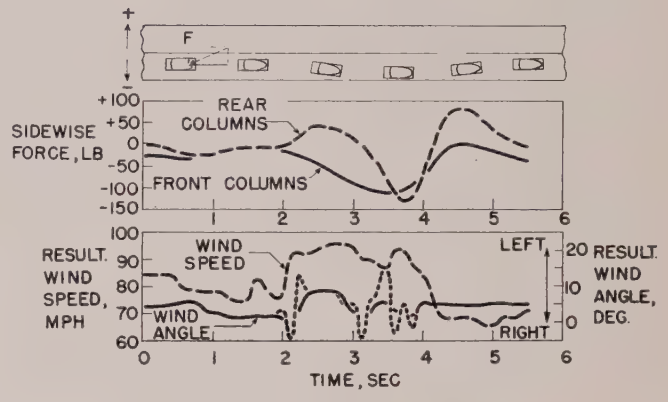


Fig. 21 – Car reaction to gusts – distribution of sidewise forces

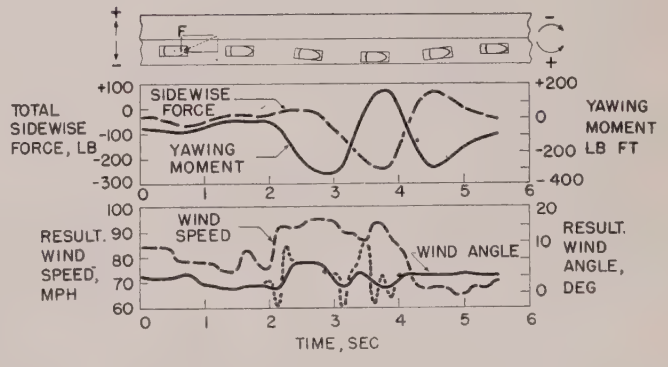


Fig. 22 – Car reaction to gusts – total sidewise force and yawing moment

from approximately 4 deg left of the direction of travel. This indicated a natural wind of about 15 mph from 25 deg left. At approximately 2.0 sec the gust struck the car. The resultant wind speed jumped to a speed between 90 and 95 mph due to the sudden 20 mph increase in natural wind speed. The vector diagram of wind velocities at 1.9 sec and at 2.1 sec is shown in Fig. 20. Striking the vehicle at an angle, the gust produced sidewise forces. Applying a lateral force to the body at a center of pressure forward of the center of gravity logically produced a tendency for the body to rotate in a horizontal plane about a vertical axis through its center of gravity. This is clearly indicated in Fig. 21.

From 2.0 to 2.3 sec the curves show a sudden increase in sidewise thrust to the right at the front columns, and a sharp increase in the opposite direction at the rear columns. Next, the driver enters the picture. The car has been pushed sidewise and turned at an angle to the direction of travel. By turning the front wheels into the gust he causes the vehicle to move back toward the original direction. The traction of the front wheels must first halt the lateral acceleration of the mass of the vehicle to the right, then accelerate it to the left again, until the original course is established. Due to the mass of the vehicle the force produced by this action is considerably greater than that caused by the wind initially. As the steered wheels turn the front of the car back toward the original direction, the tendency is for the rear of the car to skid to the right just as though a left turn were being made. This results in a large increase in lateral force at the rear body-support columns. This sidewise force momentarily exceeds the sidewise force in the same direction at the front columns. This accounts for the counterclockwise (+) moment at 3.8 sec (see Fig. 22). Total sidewise force (to the right) reaches a maximum of 240 lb at this instant also. As the car approaches its original direction of travel the angle of the front wheels is changed again by the driver, and the car sways a bit in the opposite direction before its course is stabilized. The whipping action of the rear of the car causes lateral forces to be noticeably accentuated at the rear support columns throughout the gust interval.

As illustrated by this example, lateral forces varied considerably throughout the gust interval. Corrective steering efforts on the part of the driver were responsible for the largest forces in any particular interval where a gust acted.

Conclusions

1. As the resultant wind direction changed from head-on, the total sidewise force exerted on the test car by the wind was shown to increase at a significantly greater rate than the cross-sectional area of the car normal to the wind. This effect was most pronounced where resistance due to turbulence was most significant, that is, for resultant wind angles within 10 deg of head-on.

2. The center of pressure of a resultant wind striking the car was shown to move toward the center of gravity from a location forward of the center of gravity, as the direction of the wind shifted from head-on to the side.

3. The tendency of the car to yaw was shown to be most affected by small changes in resultant wind angle from a head-on direction. This was due to the forward location of the center of pressure, and the rapid increase in sidewise force with small changes in wind angle from a frontal direction.

4. A gust striking the car at an angle to the direction of travel was shown to cause a sudden lateral displacement and to turn the car from its direction of travel. Corrective action of the driver stopping the car in its sidewise movement, and turning it into the gust so as to regain the direction of travel, was responsible for the greatest forces recorded during a gust interval.

5. Whether gusts or sidewinds strike a moving car its aerodynamic stability will be determined primarily by: (a) the degree to which lateral thrust forces due to winds are equalized between front and rear wheels, and (b) the extent to which the air resistance of the car to winds from the side is reduced. Additional cross-sectional area rearward of the center of gravity and normal to cross-winds would reduce the distance between the center of pressure and center of gravity for present-day cars and definitely improve stability in winds. Body design is the key to aerodynamic stability.

References

1. "Steering Geometry Characteristics and Their Influence on Car Handling," by H. E. Churchill and P. G. Hykes. Paper presented at SAE Summer Meeting, French Lick, Ind., June 10, 1948.

2. "Seitenwind und Fahrtrichtungshaltung des Kraftfahrzeugs" (Sidewind and Directional Behavior of Vehicles), by L. Huber. Research Report No. 44, Institut für Kraftfahrwesen der Technischen Hochshule, Stuttgart, Germany. (No date.)

3. "Einfluss der Luftkräfte auf die Fahrtrichtungshaltung des Kraftfahrzeugs" (Influence of Wind Forces on the Directional Behavior of Vehicles), by E. Sawatzki. Research Report No. 50, Institut für Kraftfahrwesen der Technischen Hochshule, Stuttgart, Germany. (No date.)

4. "Car Control Factors and Their Measurement," by Kenneth A. Stonex. Published in *SAE Journal* (Transactions), Vol. 48, March, 1941, pp. 81-93.

5. "A Quantitative Analysis of the Effects of Sidewinds and Gusts on the Stability of an Automobile Operating at Medium and High Speeds as Determined from Road Tests," by Philip W. Lett, Jr. Doctoral dissertation, University of Michigan, 1951.

Improved Body Shapes Increase Car Stability

– Victor Boghossian
Kaiser-Frazer Corp.

THE present trend toward increased car speeds – over 100 mph – puts an imperative obligation upon automotive engineers to concentrate on problems dealing with greater safety, among which stability is certainly the most important. By definition, stability of a road vehicle traveling along a given path at a given speed, is the behavior of same vehicle under the action of external forces determined mainly by road surface imperfections and lateral wind actions. The results of the work of the authors, illustrating this latter aspect of the general problem of road stability, show clearly the quantitative effects of aerodynamic stability on present-day automobiles.

The importance of this problem is evidenced by the fact that aerodynamic stability decreases with high car speeds. Improvements for greater aerodynamic stability are offered by:

1. Improved body shapes.
2. Bringing the center of pressure closer to the center of gravity.

In Fig. A we show the relationship of the center of pressure and center of gravity as they effect the aerodynamic stability of the car. We have used the term "metacentric curve" and its location, as a measure of the moments that are tending to deflect the car from its given path. The metacentric curve is the envelope of the tangent points of the resultant winds acting through the corresponding centers of pressure. In this figure, we have replotted the data in this paper to show the relative position of the metacentric curve for the car tested.

Older body types, as shown in Fig. B, had very good aerodynamic stability because of the large lateral area toward the rear of the car, thus moving the metacentric curve very close to the center of gravity. On the other hand, teardrop body shapes with low drag coefficient have poor aerodynamic stability because of the forward position of the center of pressure. It is our opinion that everybody is clear about the merits and dangers of a teardrop body shape. Evidently the best answer to the problem would be in a compromise, that is, low longitudinal drag coefficient, with the moving of the center of pressure toward the rear. This was achieved successfully in Mr. Kamm's experimental car some 14 years ago. The front end was low, with the bulk of the lateral area toward the rear, diminishing the wind actions at the front and increasing them at the rear, thus producing a stabilizing effect.

At the Stuttgart Research Institute, Mr. Kamm worked with directional fins on his experimental car. The results of this work were outstanding. He was able to produce a car which steered into the wind and thereby automatically corrected itself for the effects of sidewinds, giving the car a positive aerodynamic stability.

In the light of past experience, what can be done to improve the aerodynamic stability of our present-day automobiles?

Despite the fact that improved streamlining exaggerates sensitivity to sidewinds, it is generally well agreed today that present and future automobiles will follow the path of lower aerodynamic drag within the limits of functional design and sound esthetic conceptions. If air resistance of automobiles is gradually reduced by the normal evolution in body design, and little or no attention given to the problem of aerodynamic stability, we will be approaching a dangerous condition of instability.

Modern automobiles present the general shape of Fig. C. Obviously, moving the windshield forward shifts the center of pressure ahead, while lengthening the hood shifts it to the rear. Present-day styling trends can bring about an improvement in aerodynamic stability, and by a proper design of rear-fender tails, with a simultaneous lowering

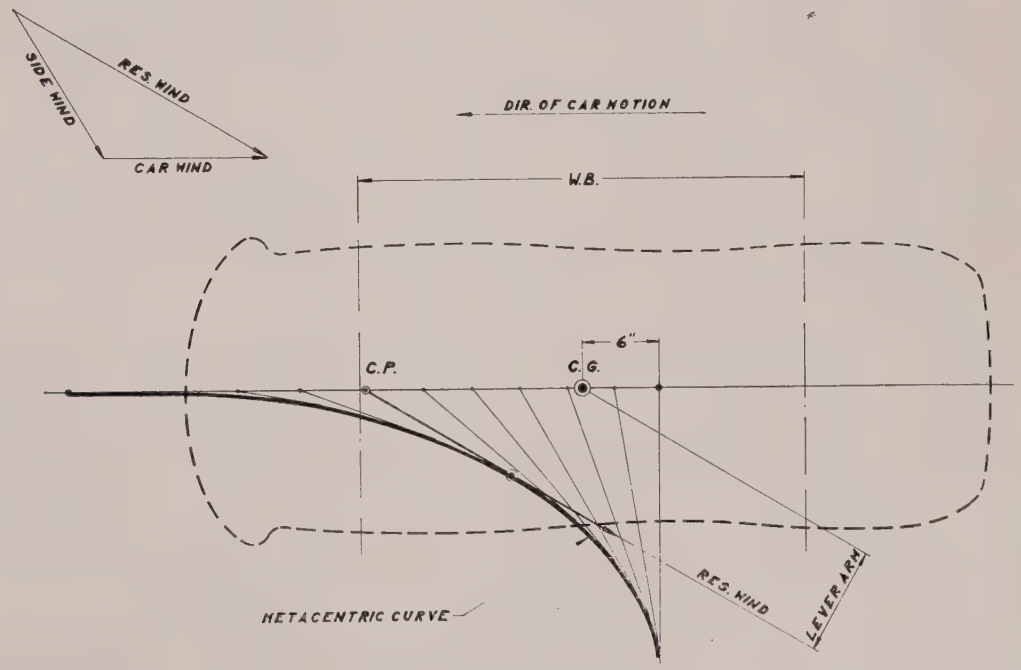


Fig. A – Relationship of center of pressure and center of gravity and their effect upon aerodynamic stability of car

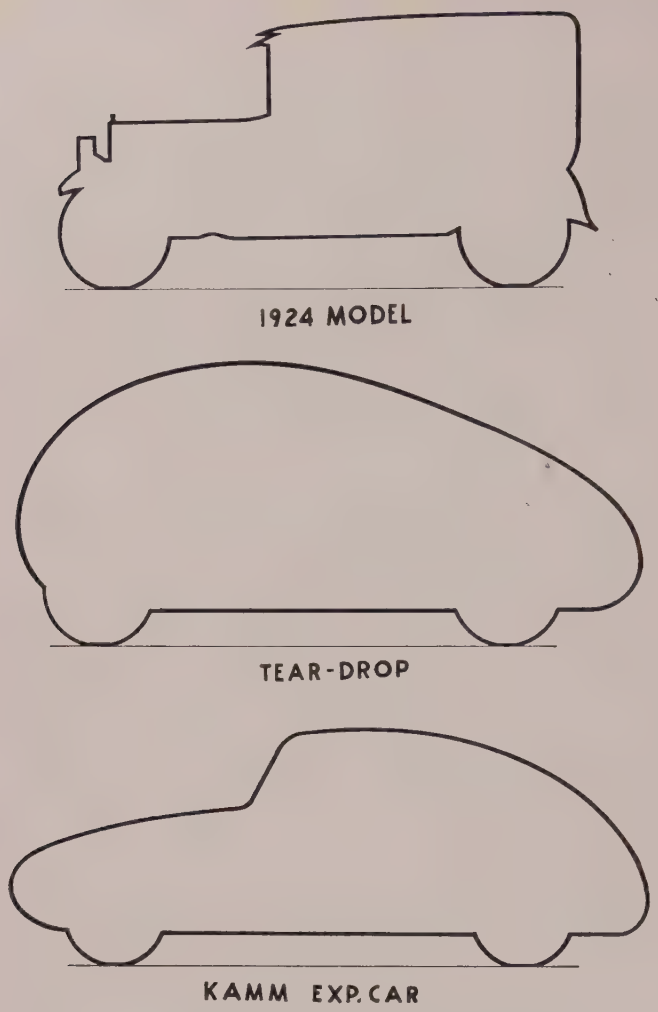


Fig. B – Effect of body shape on aerodynamic stability of car led to design of Kamm experimental model

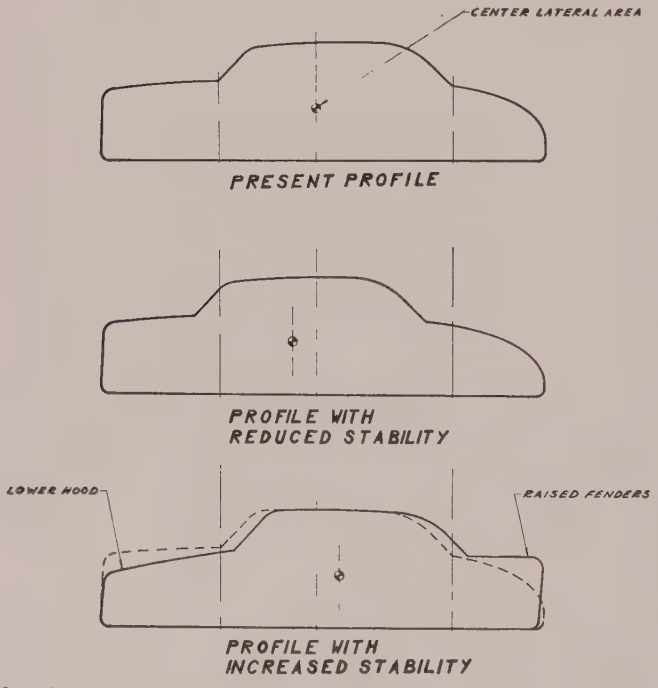


Fig. C – Proper proportioning of car within scope of present style trends may lead to aerodynamic stability

of the front end, thereby shift the center of lateral area to the rear.

To stay within the scope of present styling trends it may be difficult to design a completely stable car, but it is possible to set a trend which is in the direction of an aerodynamically stabilized car by the proper proportioning of the car. If our styling people will keep these observations in mind we will be moving in the proper direction.

Having reviewed the effects of body shape, what now is the effect of car weight on wind stability?

From the paper that has been presented, let us take an example. From Fig. 16 we find that for a resultant wind of 90 mph the lateral wind force is 100 lb and the resultant wind angle is 22 deg. From the curve of Fig. 17 we find that the location of the center of pressure is 20 in. ahead of the center of gravity. Assuming the center of gravity to be at the center of the wheel base, the 100-lb lateral wind force is then divided as follows: 66 lb across the front wheels and 34 lb across the rear wheels. It is obvious that so small a lateral force on the rear or front wheels cannot bring about any skidding or sliding of the tires in respect to the pavement. Even at considerably higher wind speed the side force does not reach the critical value which would cause the skidding of the tires.

What, then, causes yawing of the car? Yawing results from the steering of the wheels due to increasing slip-angles under lateral forces. As the lateral forces are greater across the front wheels than the rear, the slip-angle is greater on the front tires than the rear. This results in a changing of the attitude of the car with respect to its direction of travel. This change in the angle of direction of travel is called yawing. The mechanism of deformation of a rolling tire under lateral thrust is a well-known problem that has received considerable attention in recent years through experimental and analytical investigation. The results of these investigations have borne out the following: For a given constant side thrust the slip-angle increases with (1) reducing inflation pressure, (2) increased vertical tire load for a given tire, (3) increased driving force. Under these conditions any additional weight to the car will not improve the directional stability of the car, all other factors being unchanged. The increased vertical and greater driving forces will increase the slip-angle for the same given lateral wind force, causing more yawing of the car with respect to its direction of travel.

As a conclusion, the weight of the car should not be increased to improve wind-handling characteristics, but the aerodynamic properties of the car under crosswinds can be improved by proper body design, utilizing the information that has been presented in the paper.

Describes Method of Gaining Directional Stability

– W. I. E. Kamm

Stevens Institute of Technology

ROAD tests of vehicles have so far yielded only coefficients of frontal air resistance. Driver feeling has been the only means of evaluating the pushing and turning effects of sidewinds and gusts. Some of the qualitative aspects of the problem have been investigated by an analysis of motion pictures taken from the top of a tower onto a moving vehicle in conjunction with recordings of steering wheel movement. Wind-induced forces and moments have been measured under controlled conditions in wind tunnels where moving belts simulated the relative movement between car and road. However, all such tests have limitations. When the test model is less than full size it is very difficult to make it similar in all details to the actual car; when the model is full size the test facilities required become very expensive. Furthermore, it has not been possible to simulate

gusts under the controlled conditions of wind tunnel testing.

Most important, therefore, is the test procedure of the authors, that is, making quantitative measurements on a full-size vehicle during road travel. When a gust struck the car their procedure made possible the accurate recording of transitory forces and moments due to the wind, and the forces and moments due to the mass of the vehicle being moved by the steering corrections of the driver. This sensitivity would make the method suitable even for demonstrating the effects on stability (with no natural winds acting) of the Karman vortices separating from the rear of the body first on one side and then on the other. At speeds higher than 40 or 50 mph this phenomenon can cause yawing moments and steering corrections that are quite noticeable. The effects of the Karman vortices are evidenced in the calibration data, where the authors calibrated the test car instrumentation by road testing while natural winds were practically nil.

Furthermore, the data collected contain valuable information for determining C_w and $C_w f$ values for various resultant wind angles (C_w = dimensionless coefficient of air resistance; f = the vehicle's cross-sectional area normal to the resultant wind). Also of importance are the data on the location of the center of pressure of the wind striking a vehicle for various resultant wind angles. Many of these car reactions to winds would have been inaccurate on a simplified model tested in a wind tunnel.

Most important from a practical viewpoint is the corroboration of the fact that tendencies toward instability exist in conventional cars. These results emphasize the urgency of improvements for the sake of safety. An exceptionally clear demonstration is given of quickly changing rather large sidewise forces and moments acting when a gust strikes the car. The difficulty in compensating for these forces and moments by the driver's skill as well as the secondary effects of unavoidable oversteering are made evident.

If compensation of sidewind effects on the car were the only thing required, automatic steering correction could be employed. Such a correction could be made by linking front or rear wheels or both to the car in a manner that would cause them to turn the car into the wind when the car was subjected to a sidewise force or tendency to roll due to the sidewind. However, such an arrangement would cause steering in the opposite direction to what it should when the car was subjected to lateral forces coming from road imperfections. If the corrective steering turned the vehicle in one direction due to an external wind force, it would turn it in the opposite direction under the action of the inertia force of the decelerating vehicle mass. Stabilization of a car with regard to wind forces by such means would make it unstable in its reaction to road irregularities and would cause it to swerve more and more, once it had started to spin.

Automatic compensation of sidewind forces and forces due to road irregularities can be obtained by extending and combining the methods mentioned by the authors. This involves (a) moving the center of resultant wind pressure rearward closer to the center of gravity by the addition of stabilizing surfaces to the rear of the car, and (b) the creation of lateral tractive forces between tires and road by operating at some slip angle. The rearward displacement of the center of pressure should be carried far enough that this point is located behind the center of gravity. The center of the lateral tractive forces between tires and road should be located to the rear of the center of gravity also.

The perfect solution to the problem is mechanically simple and easy to understand. Directional stability of the vehicle is secured automatically without steering corrections. The solution to the problem is obtained by making the area of a stabilizing fin such that the center of pressure is moved behind the center of gravity to the point where the resultant lateral force from the tires acts on the vehicle. With equal tire pressures this point is behind the center of gravity if the center of gravity is forward of the wheelbase midpoint of a four-wheel vehicle, or forward of the point that divides the distance between the centers of front- and rear-axle groups inversely proportional to the number of front and rear axles, respectively, where the vehicle has more than two axles. This is due to the fact that lateral tire forces grow larger with an increase in wheel loading but at a rate less than directly proportional. Tire deformation under load is responsible for this. It follows that variation in tire pressures will cause a shift in the point of attack of the resultant lateral tire force—soft front tires tend to move this point rearward.

When the condition is met that the center of pressure of the wind force and the attack point of the resultant lateral tire force are at the same point behind the center of gravity, the yawing moment created by a sidewind will turn the vehicle to an oblique angle that increases until the opposing moment, created by the lateral forces between tire and road, is equal. When the moments are counterbalanced and the center of pressure is located the same distance behind the center of gravity as the attack point of the tire-road resultant force, the resultant wind force and the resultant tire force are oppositely equal. With compensated moments and balanced forces the vehicle gains directional stability and runs at some slip angle, the size of which depends on the magnitude of the wind-induced yawing moment. If this moment changes the angle changes—compensation of the wind-induced moment is accomplished by a corresponding change in the resisting moment between tires and road. Should the sidewind suddenly cease or shift to a frontal direction, the tire force moment reduces to zero the angle at which the vehicle has been moving. At the same time the tire force moment itself disappears. Momentary oblique angles caused by road irregularities are overcome by the moment of the lateral tire forces, which acts to restore the balance between the wind-induced yawing moment and the tire force moment. If the resultant wind is exactly "head-on" the tire force moment will tend to restore the car to a zero angle of obliquity. During the very short periods of transition from one state of balance to another the lateral forces are temporarily uncompensated. The resulting shifting effect is so small that it is imperceptible. It can be shown by calculations that balance is reached after a fraction of a second when strong sidewinds strike suddenly a vehicle of conventional weight stabilized by this method. Shifting movements during this occurrence cause angles no greater than 1/10 deg between the final direction of motion and the direction of undisturbed motion.

These perfect conditions can be realized only approximately in actual operation. Common variations in loading a vehicle displace the center of gravity and the attack point of the resultant lateral tire force. Changes in resultant wind direction affect the location of the center of pressure of the wind on the vehicle as described by the authors. For these reasons the center of pressure and the attack point of the lateral tire forces normally lie at different spots. Fundamentally stable operation can be secured if (a) the center of gravity is located initially so that it is ahead of the wheelbase midpoint for all common distributions of vehicle load, thus placing it in front of the attack point of the lateral tire forces, and (b) if the fins are made large enough to cause the center of pressure to lie behind the center of gravity at any resultant wind angle substantially different from zero.

A slight variation in the conditions for stable operation may be illustrated by the following example: The center of air pressure may lie between the center of gravity and the attack point of the tire forces. The lateral force caused by a sidewind will result in the car's running at some slip angle. The angle increases until the wind moment and tire force moment are balanced. The lateral wind force is greater than the lateral tire force when these moments are equalized, since the distance between the center of pressure and the center of gravity is less than the distance between the attack point of the lateral tire forces and the center of gravity. The sidewind force causes the vehicle

to be shifted laterally. The driver compensates for this movement by adjusting his steering wheel. A steering moment is superimposed on the wind and tire force moments which increases the slip angle at which the car is operating until the lateral forces are compensated by the balance of the three moments. Due to the long lever arm between the steered front wheels and the center of gravity, the steering force necessary at the front wheels for this additional moment is minute. A very small angle at the steered wheels is sufficient. The small angle the steering wheel must be turned cannot be discerned by the driver from the angle required for maintaining a direction of travel. He applies this correction easily and unhurriedly, without being forced to act quickly, with consequent oversteering, which is the case with conventional vehicles, as the authors have demonstrated so clearly.

The method of stabilization described has only recently been proved reliable even on very slippery ice-covered roads. Due to low friction of the tires, larger than normal slip angles were necessary to achieve balance between wind and lateral tire force moments. Under normal driving conditions these slip angles range between ½ and 3½ deg.

Simple Instrumentation for Wind Measurement Suggested

– T. J. Kozier
Ford Motor Co.

THIS paper describes a very ingenious method of measuring wind effects on car stability.

For my own information, however, I am wondering if the authors could clarify for me just what the center of gravity has to do with yawing moments on a car or truck. It would appear to me that a vehicle such as an automobile, bus, or boat, which is not entirely airborne, but is more or less fixed to the ground or water surface, exhibits a tendency to yaw which depends solely on the relative location of the center of pressure and the center of resistance.

Perhaps this point can be clarified with the analogy of a sailboat. If the boat is trimmed so as to keep a constant heading in a crosswind and then the centerboard is raised, the center of resistance of the water to the hull is changed and the boat immediately yaws. We have done nothing at all to the location of the center of gravity. Likewise in an automobile under the influence of a sidewind force, it would seem to me that the distribution of lateral forces between front and rear wheels depends on the location of these front and rear wheels with respect to the center of pressure, and has nothing whatsoever to do with the center of gravity location. Whether or not the vehicle yaws under the influence of the wind force appears to me to depend on the lateral stiffness of the front and rear tires and suspension, and how much they deflect sideways under this force, neglecting for the moment axle steer with roll, which is purely a geometry problem.

On a transit bus or truck, for example, removal of the outside dual wheels in the rear renders the vehicle quite sensitive to wind force. It does this, I believe, because by removing two tires from the back end we have halved the lateral stiffness in the back so that the back end yields more under the influence of sidewind. I don't think the change in bus behavior has anything to do with the very slight change in center of gravity caused by the removal of the wheels.

If the yaw of a vehicle depends on the locations of the center of pressure and the center of resistance due to lateral stiffness of the suspension, and not on center of gravity location, we have a further problem. The lateral deflection of the tire and suspension combination is unfortunately not a linear function of the forces applied. If our tire pressures and suspension stiffnesses are such that both ends of the car deflect sidewise an equal amount under the influence of a given sidewind so that the car maintains its heading without yaw, but with a slight lateral drift, we assume that we have an ideal condition: the center of pressure is coincident with the center of resistance. If we doubt the sidewind forces, however, and the front and the rear deflection rates are unequal and nonlinear, we now find that the center of resistance is somewhere else and the car will yaw. So we are faced with the problem of designing a body in which the center of pressure will shift location properly depending on the magnitude of the side force. This could be quite a problem.

We would like to suggest a somewhat simpler means of instrumenting a vehicle for the measurement of side force which could be applied very easily to a wide variety of vehicles. Disc wheels used on most cars can be equipped with resistance strain gages radially disposed near the rim on inner and outer surfaces of the wheels, and connected so as to indicate the strains produced by force applied at the axle and resisted at the road surface, cancelling out all other extraneous strains. The output of this gage array can be brought out through sliprings and recorded on a multichannel oscillograph. Such an arrangement not only makes the problem of transferring instrumentation from one vehicle to another about as convenient as changing tires, but the data from all four wheels is recorded simultaneously on a time axis with much better frequency response to transient effects than we suspect is obtained with microammeters and movie cameras. We also suspect there is less labor involved than in reading off data from individual frames on a movie film.

Authors' Closure To Kozier Discussion

AN attempt was made in this investigation to consider separately the essential elements of the problem of wind effects on car stability. The applied forces here are wind forces that produce yawing moments. The resisting forces are tractive forces between tires and road. If the lateral forces due to sidewinds are balanced by the opposing lateral tractive forces no drift occurs. If the wind-induced yawing moment is balanced by an opposing moment produced by the resisting tractive forces no yawing occurs. The application of a sidewind force to a body moving through air produces a tendency for the body to rotate about its center of gravity in every case where the center of pressure is not coincident with the center of gravity. Contact with the ground in no way alters this applied moment. Whether the body actually yaws under this applied moment depends entirely on the resisting moment produced by the tractive forces between tires and road. The analogies mentioned of raising the centerboard on a sailboat and removing the outside dual wheels on the rear of a truck result in a tendency to yaw because of the reduction in the resistance. There has been no change in the wind forces and moments, no significant change in the center of gravity, but yawing now occurs because the wind forces and moments are no longer balanced with the reduction in resisting forces.

The problem referred to of designing a body in which the center of pressure will shift location with variations in the magnitude of the side force is more than a problem—practically speaking it is an impossibility. The approach suggested by Dr. Kamm in his discussion would seem to offer more promise toward gaining directional stability.

It was a primary purpose of this work to make evident the significance of body design as it affects directional stability of cars subjected to sidewinds. Disc wheels equipped with strain gages would obviously provide a simpler means of instrumentation, but, at the same time the purpose of the investigation would be defeated. This would occur because wind resistance of the body itself could no longer be measured with the introduction of the suspension between the instrumentation and the body.

SAFE BRAKES FOR PASSENGER CARS

George T. Ladd and **Sidney B. Dew**
Al-Fin Division, Fairchild Engine & Airplane Corp. Wellworthy, Ltd.

Paper presented at SAE Nat'l Passenger-Car, Body, and Materials Mtg., Detroit, March 3, 1953.

IT is generally recognized throughout the industry that present brake-drum designs, particularly those on larger cars, are operating at or near maximum capacity, and that improvement in braking capacity is an early requirement. Specifically, this need is attributable to factors such as higher speeds resulting from increased engine horsepowers and better roads. The increasing use of automatic transmissions and fluid couplings reduces the use of the engine as a retarding factor and places more responsibility on the brakes.

Until recently, brake size was influenced mainly by its holding ability or torque capacity and not so much by a requirement for heat-dissipation capacity.

Fortunately, many are working on the problem, as evidenced by the innovation of highly efficient disc brakes, impeller hubcap aircooling, boosters,

THE bonded bimetallic brake drum has been developed to help solve the increasingly difficult problem of providing satisfactory braking service for modern cars.

These drums have an aluminum-alloy housing bonded to a cast-iron liner.

In this combination, the cast iron provides the wearing surface and the aluminum alloy provides the high heat conductivity and low weight. The result is an improvement in the rate at which heat is dissipated and a lower unsprung weight for the car. These brake drums have also been found to help considerably in eliminating squeal.

In England, it is reported, several car manufacturers are already using these drums successfully.

The Authors

GEORGE T. LADD (M '42) is chief engineer of the Al-Fin Division of the Fairchild Engine and Airplane Corp. He is a graduate of Case Institute of Technology. He has been with his present organization since 1949.

SIDNEY B. DEW is chief chemist and metallurgist of Wellworthy, Ltd., in England. He was appointed to the board of directors of that organization in 1949. He is a graduate of Kingston Senior Technical Science College.

Fig. 1 – Air-flap brake used at LeMans (France) road race

larger external drum areas, modified shoes and shoe action to distribute cooling air and loading more evenly.

At the Fairchild Engine and Airplane Corp. in this country and at Wellworthy, Ltd. in England, we are actively engaged in the development of aluminum-alloy bimetallic drums and disc brakes for passenger cars, buses, trucks, trailers, railcars, aircraft, and high-speed inertia brakes for machinery.[1] Much of our development work has been done in the road-racing and sports-car fields; but we feel that out of this work will come the solution for stopping our high-speed, high-powered, heavy passenger cars.

Henri Perrot, president of the Société des Ingenieurs des L'Automobile, in the 1952 *SIA Journal* comments that cars running in difficult races such as the LeMans 24-hr road race require brakes that are put to very exacting tests, and that new solutions must come to light. Supplementary air brakes, analogous to flaps on airplanes, were mentioned as a possible solution for cars which are capable of 160 mph. Fig. 1 shows an experimental air brake, used at LeMans this year.

Some of you may remember when the speed limit was 30 mph and automobile advertisements claimed that the brakes would stop a particular car operating at 30 mph within a space of 10 ft. By a few simple calculations, we can show easily the fallacy of such a claim. From the expression $V^2 = 2gh$, where V equals velocity in fps and h equals height in ft; it can be calculated that a car moving at 30 mph is developing sufficient kinetic energy to lift itself vertically 32 ft. The minimum distance in which a car with optimum brakes can be stopped on a level road depends to a large extent on the coefficient of friction between the tire and road, which depends in turn on the character of the road surface. Using our example and assuming a tire-to-road coefficient of friction of 0.6, we can equate the kinetic energy of the car to the work done in overcoming friction in a distance, $D = V^2/2g\mu$. Solving this equation for the distance D, we obtain approximately 50 ft instead of the claimed 10 ft.

The distance equation is free from mass and theoretically a heavy car and light car require the same distance for stopping. It should also be noted that the value of the minimum stopping distance depends on the square of velocity. The stopping distance for this example was determined by considering all four wheels locked and sliding, and consideration must be given to the fact that, in many cases, especially when the highway is wet, the friction coefficient is lower than the 0.6 used. Some years ago T. P. Chase presented data[2] to show that the skidding coefficient decreases with speed to the extent that at the highest driving speeds, a coefficient of friction of 0.6 is the maximum that can be expected, and that the maximum rate of deceleration available without sliding wheels is 20 ft per sec per sec. In ordinary passenger-car use, it is, of course, impractical to stop a car by skidding for the full stopping distance and, therefore, the deceleration design factor for passenger cars is usually between the values of 10 to 18 ft per sec per sec.

The B. S. Cunningham Co. reported that their driver stopped the 2600-lb Cunningham sports car, the type used at the LeMans 24-hr race, from 100 mph in something like 6 sec. This car was equipped with a somewhat complex brake- and wheel-cooling arrangement in which the brake drum (Fig. 2)

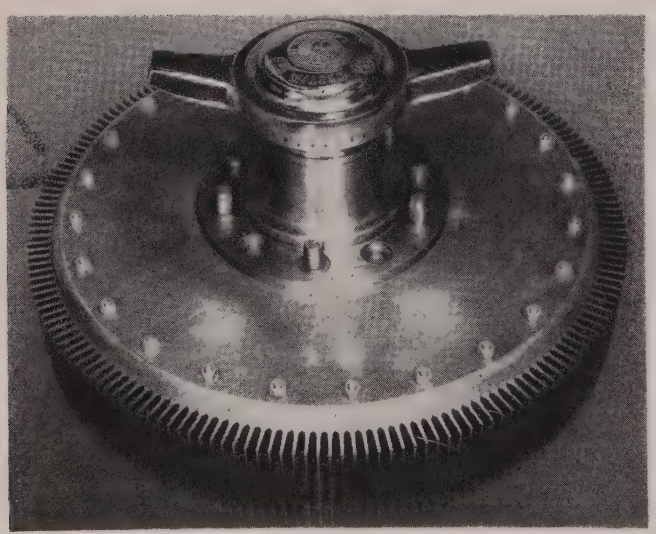


Fig. 2 – Bimetallic brake drum of aluminum and cast iron

[1] The process for joining aluminum alloys to ferrous metals has been patented by the Fairchild Engine and Airplane Corp. under U. S. Patent Numbers 2,455,457; 2,453,772; 2,435,991; 2,396,730.

[2] See SAE *Quarterly* Transactions, Vol. 3, January, 1949, pp. 26-40: "Passenger-Car Brake Performance – Limitations and Future Requirements," by T. P. Chase.

consisted of rather fine-pitch lateral-fin aluminum bimetallic construction. The drum was bolted to a cast-aluminum-alloy wheel which was designed as a centrifugal compressor, with the inlet on the inboard side of the car and the air outlet through the shrouded openings near the wheel rim, as illustrated in Fig. 3. The brake-drum fins, being matched with the inlet vanes of the wheel compressor, acted as an inlet inducer for the cooling air passing over the fins. In fact, Cunningham reported a rather interesting rough road test of braking ability by accelerating to 100 mph as fast as possible and immediately braking to a stop at maximum deceleration. This cycle was completed 30 times, and without interval a similar cyclic performance was started with the maximum speed at 60 mph instead of 100 mph. This time the test was discontinued after 11 cycles because some brake fading was evident. Again without pause, the car was taken to the 100-mph acceleration and full-stop cycles, and the test was stopped after a further 30 cycles. Since there was no fading during the more rigorous braking from 100 mph, an analysis of these conditions will show the important part played by cooling air for thermal recovery. Generally speaking, the convection heat-transfer rate is a square root function of air velocity, and in the case of a brake drum there is the double effect of the forward velocity of the car and the increased air velocity due to the higher rotational speeds of the wheel. This explains the improved recovery when accelerating to 100 mph as compared to accelerating to 60 mph.

The first and all-important design consideration is that the capacity of the brakes must be sufficient for the weight and speed of the vehicle, bearing in mind that the work the brakes must do varies directly as the weight of the car, as well as the square of the car velocity. For the discussion to follow, we will make little reference to other factors that must also be considered by the brake designer, such as uneven skidding, driver reaction, road condition, brake-lining material and its changes in coefficient of friction, force of inertia – weight transfer to the front wheels, and brake mechanism design. Adequate treatment of all these factors would require not one but a series of papers.

An important safety item is brake-pedal pressure, especially in consideration of physical strength requirements and the physical fatigue factor involved in sustained driving. The increasing use of boosters is helping to meet this need. To achieve the desired low pedal pressure without resorting to the use of power devices, it would help materially to use brake drums that will prevent uneven brake-shoe contact under heavy loading conditions and, of course, drums that will not overheat and fade. In the racing-car field, to have a reasonably easy pedal, self-energizing brakes have been used. However, it has been

Fig. 3 – Wheel-cooling arrangement for brake drum shown in Fig. 2

discovered that the self-energizing principle can be dangerous at high speeds. Racing drivers believe that, at high speeds, the driver should be in complete control of the brakes and, therefore, for some designs of larger diameter, more effectively cooled brake drums are being considered. Also, to overcome the difficulties of uneven pressure distribution due to lateral deflection of wide brake shoes, consideration is being given to using two narrow shoes side by side. Possibly, more thought should be given to suitably ribbed cast-aluminum shoes or to standard type, T-shaped welded steel, reinforced as shown in Fig. 4 to overcome brake-shoe deflection.

The ability of a brake drum to dissipate heat is improved by increasing the drum area in contact with the surrounding air, and by causing more air to circulate over the surfaces. Additional improve-

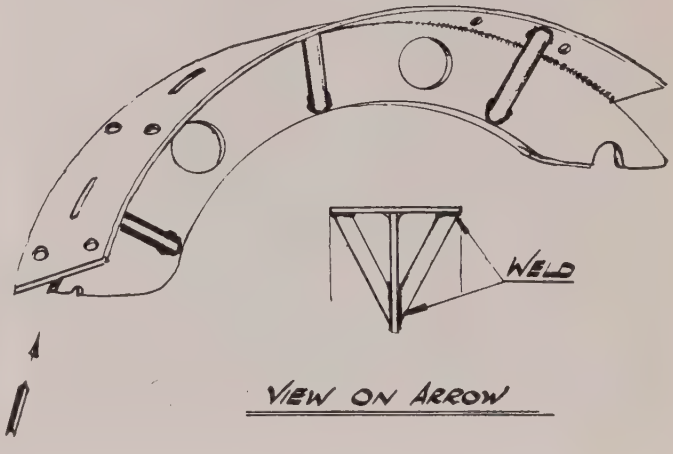


Fig. 4 – Reinforced welded-ribbed T-brake shoe

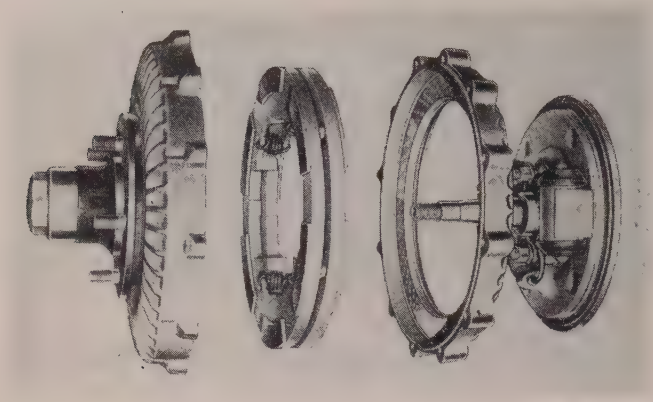


Fig. 5 – Passenger-car disc brake – exploded assembly

Table 1 – Road Test Schedule for Brake-Drum and Lining Endurance

A. Route:
Black top — 3.80 miles

B. Checks:
Brake curves at 0, 1000, and 3000 brake test miles.
Brake-lining measurements at 0, 500, and every 1000 miles thereafter.

C. Detail
First lap: 6 decelerations from 60 to 35 mph.
Second lap: No decelerations, drive at 50 mph to cool drums.
Third lap: 6 decelerations from 30 mph to full stop.
Fourth lap: 6 decelerations from 35 mph to 10 mph.
Fifth lap: 2 decelerations from 70 mph to full stop.
Sixth lap: No decelerations, drive at 50 mph to cool drums.
Stops are to be made approximately 0.6 mile apart, except on 70-mph stops, which are to be approximately 1.9 miles apart.
All decelerations are to be made with clutch engaged, decelerations are to be as fast as possible without sliding tires.

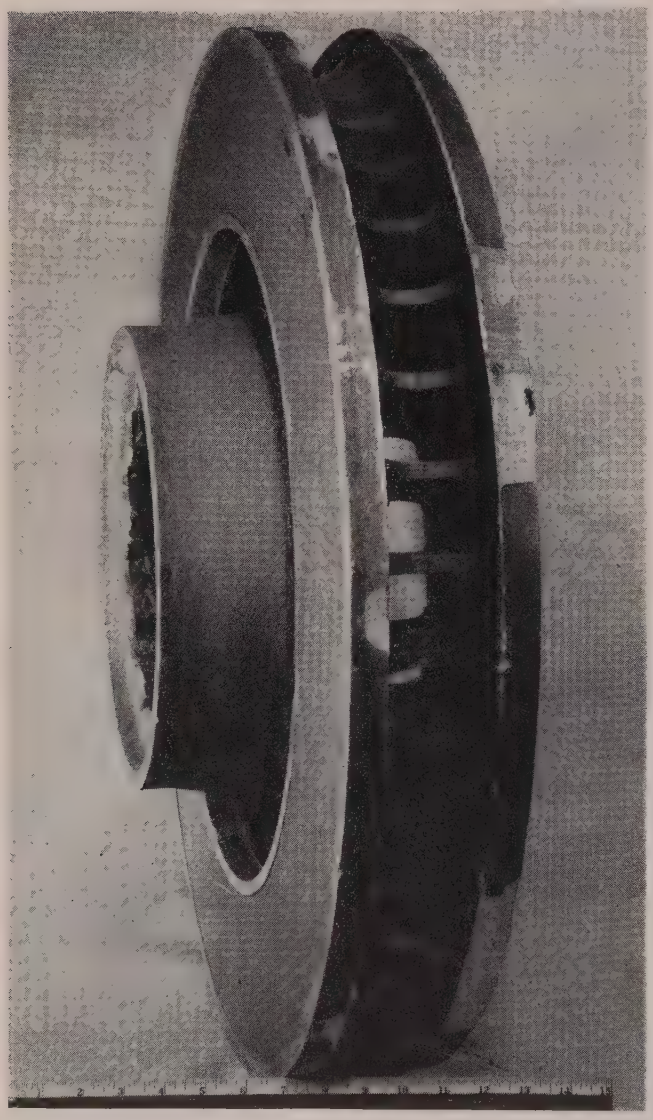


Fig. 6 – Railcar disc brake

ment can be gained by flowing cooling air inside the drum; cooling at the source of heat generation. In cooling internally, another factor of the problem of brake lining is brought to mind, that of brake-lining coverage of the braking surface. If the lining covers the entire braking surface of the drum, very little additional cooling can be obtained from this surface by circulating air in the brake cavity. Ralph K. Super reports[3] that the heat-dissipation capacity of the brake drum can be doubled by removing one-third of the lining and exposing the braking surface to airflow. It is interesting to note that the old-time rule of thumb reasoning, that the greater the lining area the greater the load capacity of the brake, has been proved to be in error. For a long time it was the tendency to increase the size of the brake shoes until the entire braking surface of the drum was in contact with the lining. It is not difficult to realize the impracticability of aircooling the high-temperature braking surface when that surface is almost completely covered by a lining material having excellent insulating properties. Thus, a major part of the heat generated must work its way through the metal of the drum, where it is finally dissipated from the outside by radiation and convection.

Development of Smaller Brakes

For normal severity of usage, production drums of the standard steel and centrifused cast-iron type, with a braking-surface diameter of 11 or 12 in., give the highest deceleration rates possible with present tires and pavement surfaces, under most driving conditions. One possible improvement, with present brakes, would be a 2-master-cylinder assembly with a proportioning linkage that would alter the pressure distribution between the front and rear brakes in relation to prevailing

[3] See *SAE Journal* (Transactions), Vol. 54, May, 1946, pp. 205-214: "Brake Designs and Methods of Rating Brakes for Commercial Automotive Vehicles," by Ralph K. Super.

Fig. 7 – Al-Fin bonded cast iron and aluminum (reduced from photomicrograph magnified 500x – 3% nital etch)

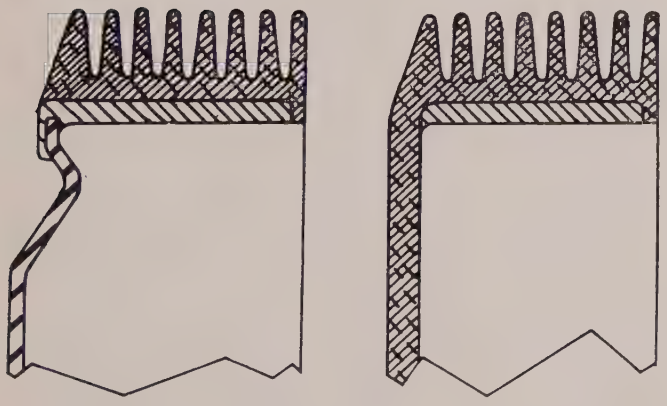


Fig. 8 – Schematic sketches of phase I (left) and phase II (right) bimetallic brake-drum designs

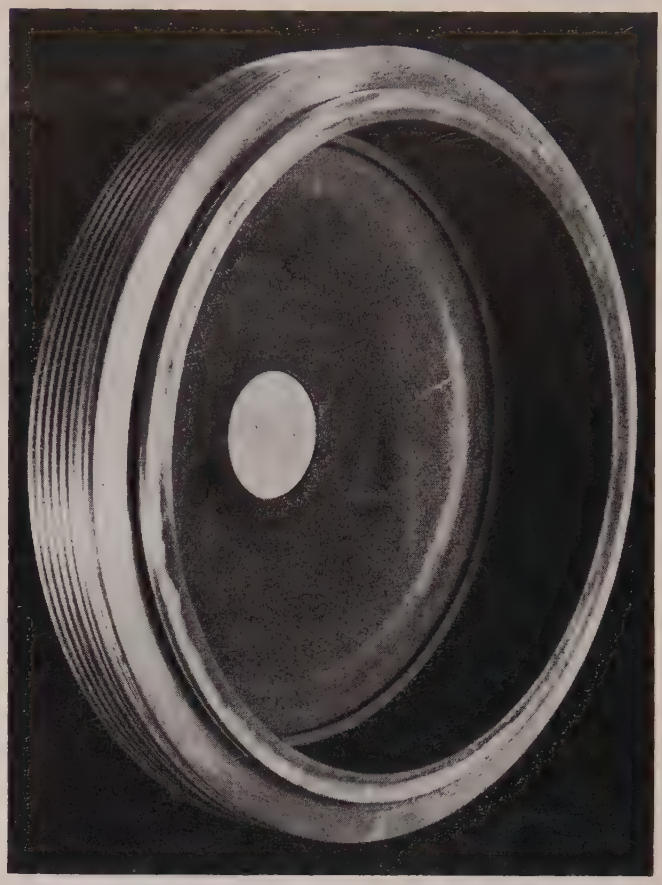


Fig. 9 – Photograph of phase I bimetallic brake drum with circumferential cooling fins

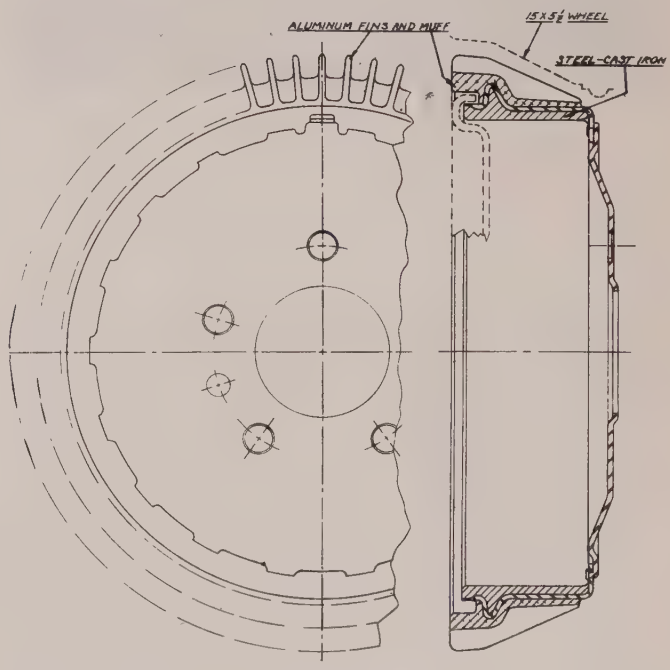


Fig. 10 – Drawing of phase I bimetallic brake drum with radial fins

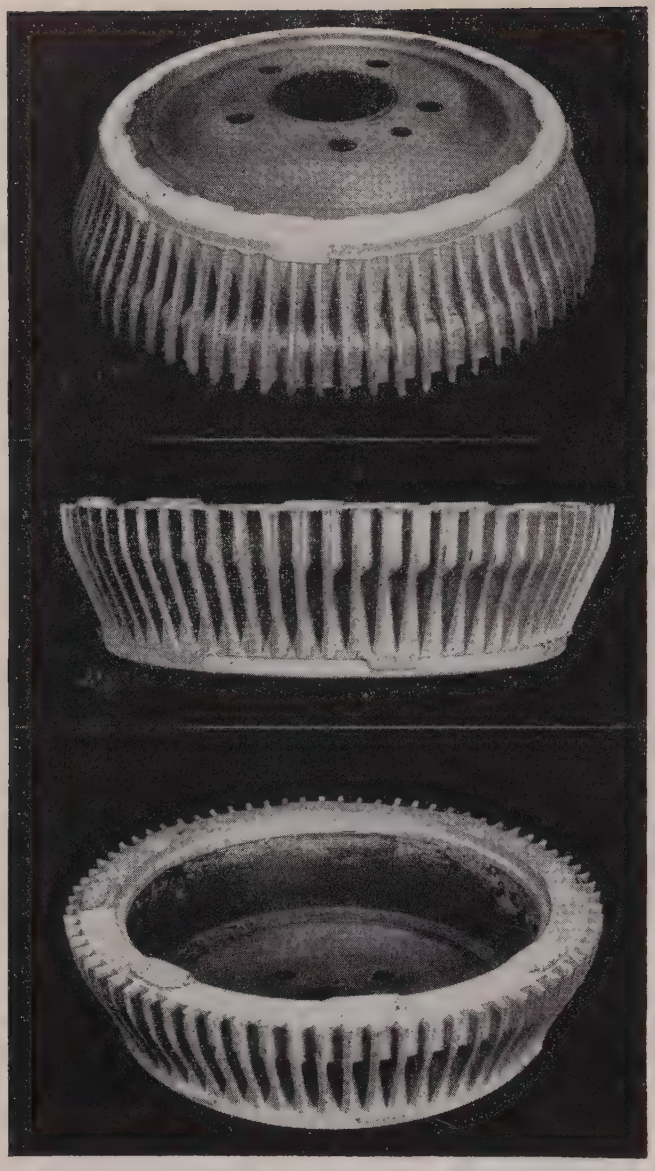


Fig. 11 – Photograph of phase I bimetallic brake drum with radial fins

tire-to-road friction. This would permit higher pressures to be applied to the front wheels for complete utilization of the dynamic weight shift on dry concrete pavement, and possibly it would permit a slight increase in rear over front brake pressure to ensure rear brakes coming in fast on slippery pavement so that steering control could be maintained.

Deceleration versus pedal pressure graphs of 10-in. drums closely conform to the graphs for the currently used steel and centrifused cast-iron drums with diameters of 11 or 12 in. The 10-in. drum was tested with dual-opposed brake shoes, which consisted of two narrow sets rotated 180 deg relative to each other. Running on the proving-ground schedule (Table 1), the standard 12-in. drum lining has a life of from 5000 to 6000 miles. A limited amount of testing on the 10-in. brake indicates a lining life of 8000 to 9000 miles. For the evaluation of brake performance over and above normal duty, dependence is placed on results obtained on mountain tests. In general, cars equipped with 10-in. brake drums make scheduled mountain runs with little or no difficulty. The cars equipped with standard 12-in. drums have been improved to the extent that they also complete the run, but occasional brake failures occur.

To summarize: The 10-in. dual-opposed brake appears to offer better drum and lining life, more consistent output and freedom from fade on very difficult test schedules, with possibly one or more additional stops when making successive stops from 70 mph one-half mile apart.

The improved characteristics of the 10-in. brake are due probably to the following factors:

a. Greater rigidity of drum with smaller diameter.

b. Distortion balance achieved by rotating two sets of shoes 180 deg from each other.

c. Increased air space for cooling between the outside of the drum and the wheel.

d. Minimized effect of high unit pressures due to bellmouthing by the use of two separate sets of shoes.

Bimetallic Brake Drums

Passenger-car manufacturers in the United States, for the most part, use fabricated steel and centrifused cast-iron drums. However, in England several have gone to the use of aluminum alloy bimetallic brake drums with metallurgically bonded-in cast-iron liners. Most important advantages of the aluminum alloy are its high thermal conductivity and light weight, resulting in improvement in heat-dissipation rate as well as in unsprung weight.

It is significant to note that one major American manufacturer of disc brakes (Fig. 5), who has utilized aluminum construction, reports no loss of brake pedal and also the elimination of fade, in spite of frequent and repeated severe use, often on long downhill grades.

Several designs of brakes in current use, in applications other than automotive wheel brakes, utilize the principle of maximum air circulation and maximum exposure of the drum braking surface. One example is the experimental disc brake used for high-speed railcar service (Fig. 6).

In the discussion that follows, bimetallic construction refers to units fabricated by the Al-Fin process of molecularly bonding aluminum and its alloys to iron and its alloys. Fig. 7 shows the metallurgical structure of a typical Al-Fin bond (reduced from a photomicrograph taken at 500 magnification) between cast iron and a silicon-aluminum alloy. The bond, besides having a mean tensile

strength of 15,000 psi, is capable of conducting heat across the bond interface without a temperature drop.

The development of the aluminum bimetallic brake drum has been along two major lines, identified as phase I and phase II. Both designs are shown schematically in Fig. 8. Phase I design, in general, is a bimetallic drum that consists of a standard cast-iron or centrifused production drum with an aluminum muff or finned structure bonded to the outside surface, primarily to improve the thermal recovery of the drum by aircooling. Fig. 9 shows a drum with circumferential fins and Figs. 10 and 11 show a drum with radial fins.

The phase II drum has an aluminum-alloy housing, which carries the torque and dissipates the heat, and a metallurgically bonded cast-iron liner machined to give the surface conditions required for braking use. The high-temperature resistance to wear of cast iron (for the wear surface) is combined with the lightness and good heat conductivity of aluminum (for the body of the drum). Some designs of the phase II drum with circumferential cooling fins are illustrated by Figs. 12 and 13 and with radial fins by Figs. 14 and 15. A pie section, showing the disposition of the castiron liner, is illustrated by Fig. 16.

The phase I construction makes possible the utilization of available tooling and the aluminum can be pressure die cast and bonded, or permanent-mold cast and bonded, to the iron drum in large quantities to save subsequent machining cost, and to keep the increased cost within reasonable limits.

In the past, bimetallic drums have been made with mechanical interlocks between the cast-aluminum drum housing and the cast-iron liner. However, when an aluminum alloy having a high coefficient of expansion is cast without a bond about a ferrous-alloy liner having a relatively low coefficient of expansion, the tendency will be for the aluminum to separate from the iron band during heat-cycling. This may result in plastic deformation of the aluminum, so that when it cools the joint with the iron will not be as close as it was originally. Subsequent heating cycles combined with mechanical load deflections will exaggerate this condition until the iron band becomes loose in the aluminum housing or muff. With the bonded construction, relative motion at the interface is eliminated and the bimetallic unit will maintain original tightness and thermal conductivity.

Design of Brake Drums

The primary purpose of a brake drum is to absorb and then dissipate large amounts of energy without exceeding the temperature at which the linings lose their effectiveness. The large power input that brake drums must absorb is indicated by the following analysis of a standard fade test consisting of stops from 70 mph at 15 ft per sec per sec deceleration at 1-min intervals, for a 4000 lb car.

$$\text{Hp per brake} = \frac{\text{KE}}{t(33{,}000)} = 5 \text{ hp}$$

This figure represents the theoretical continuous power that the brake drum absorbs. Fortunately, in ordinary usage such operation does not continue long, and early recovery is possible during longer periods of operation with no braking at high speed, or braking at much lower energy levels.

Fig. 17, a graph of the stop-start cycle of vehicle and brake, illustrates the velocity and horsepower

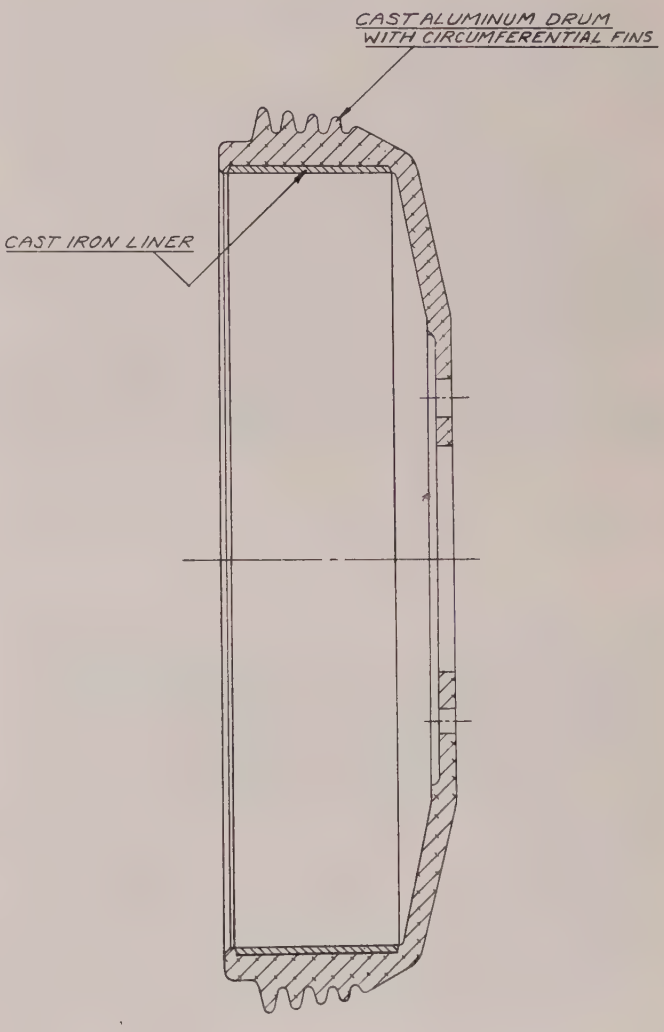


Fig. 12 – Drawing of phase II bimetallic passenger-car brake drum (12 x 2¼ in.)

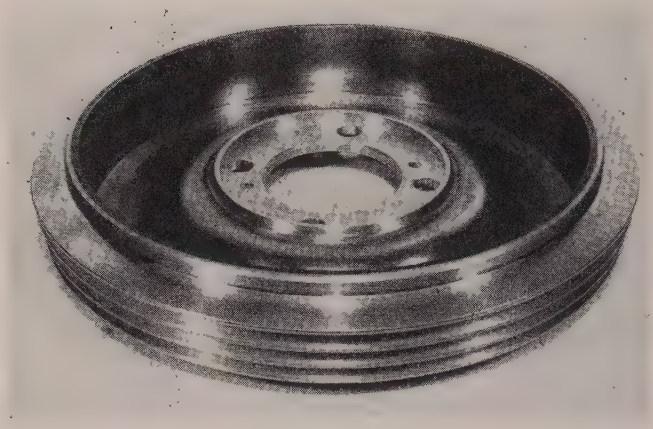


Fig. 13 – Photograph of phase II bimetallic passenger-car brake drum (12 x 2¼ in.)

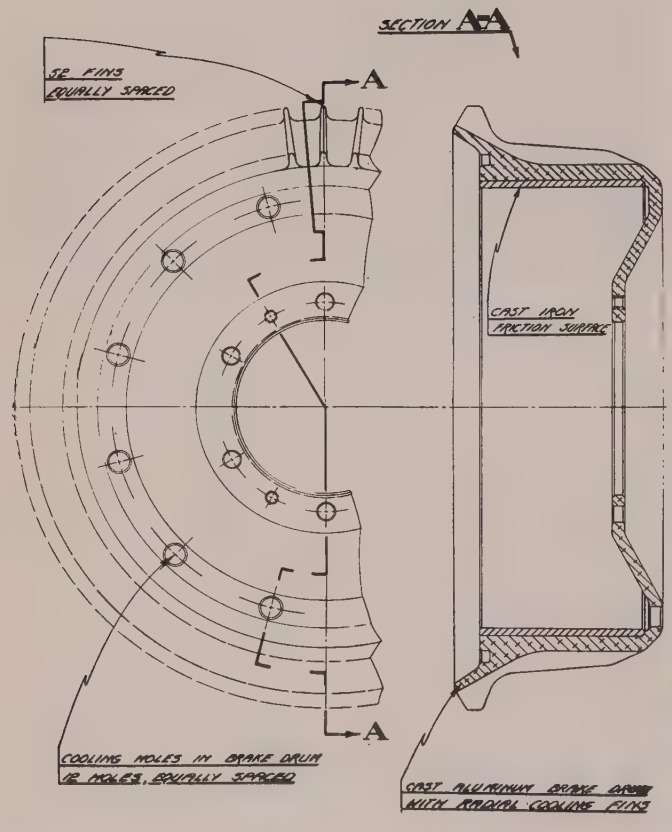


Fig. 14 – Drawing of phase II bimetallic brake drum with radial cooling fins

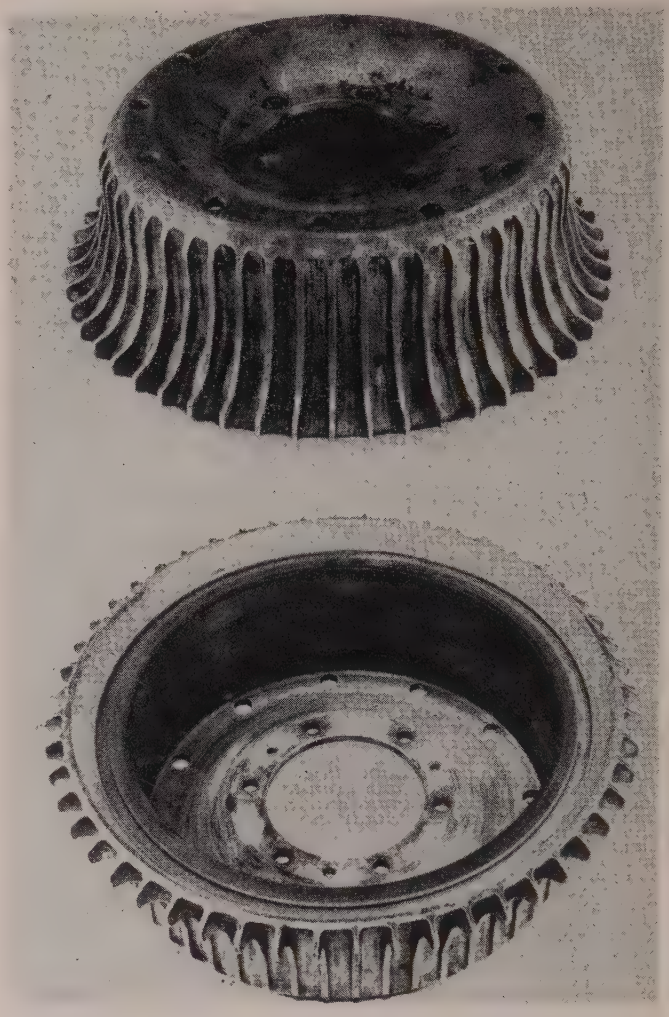


Fig. 15 – Photograph of phase II bimetallic brake drum with radial cooling fins. This type is used on Mercedes LeMans and Pan-American 300-SL

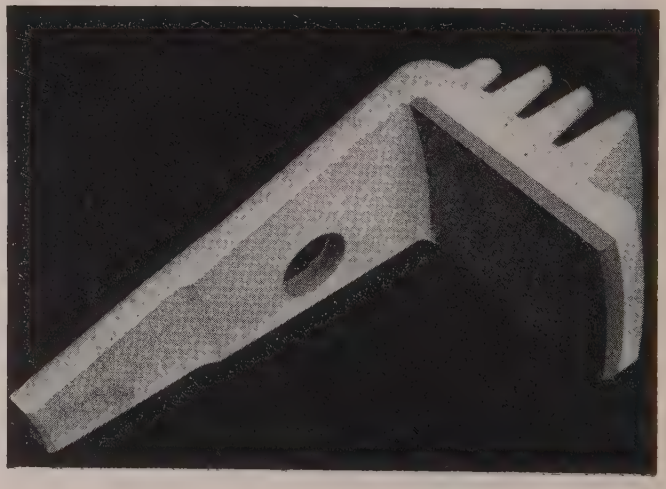


Fig. 16 – Cross-section of phase II bimetallic brake drum

versus time relationships for these conditions. The maximum hp is represented by:

$$\text{Hp} = \frac{\Delta \text{KE}}{\Delta t \times 550} \tag{1}$$

$$\text{Hp} = \frac{\frac{1}{2}\frac{W}{g}V_2^2 - \frac{1}{2}\frac{W}{g}V_1^2}{\Delta t \times 550} \tag{2}$$

$$\text{Hp} = \frac{d}{dt}\left[\frac{W}{2 \times g \times 550}\int_1^2 V dV\right] \tag{3}$$

From, $V = V_i + at$ and $dV = adt$, substituting in (3) we obtain:

$$\text{Hp} = \frac{d}{dt}\left[\frac{Wa}{2 \times 550 \times g}\int_1^2 V dt\right] \tag{4}$$

Where $\int_1^2 V dt$ = area under $(V - t)$ cycle graph (Fig. 17)

continuing:

$$\text{Hp} = \frac{WaV}{8 \times 550 \times g} = \frac{4000 \times 15 \times 103}{8 \times 550 \times 32} = 44 \text{ hp per brake}$$

Another condition that must be considered is long downgrade operation. In many cases it is more important than stops from high speeds because of the continuous heavy braking and low aircooling effectiveness. If we consider a 10% grade, the acceleration component of the gross weight may be taken as $0.10(W)$. For one brake, the torque required to maintain speed without acceleration is:

$$T = \frac{0.1 \times W \times R}{4}$$

where:

W = Gross weight

R = Tire radius (1.2 ft for 15 x 7:00 tire)

The power absorbed is, therefore, $T\omega$, where ω = rotational speed of wheel in radians per sec and

$$\omega = \frac{V}{R}.$$

For a 30-mph dragging speed, the horsepower $\frac{T\omega}{550}$ for the same 4000-lb automobile is 8 hp, which is equivalent to 20,360 Btu per hr. A series of tests was run, using a test rig (Fig. 18) that consisted of a standard passenger-car brake drum and wheel assembly driven by a lathe. Heat was applied to the drum over a 180-deg section by means of a gas burner inside the drum. A small air blower was used to simulate the aircooling of the vehicle velocity and wheel rpm. Inside surface temperatures of the brake drum were measured with a silver-brush thermocouple imbedded in asbestos and ceramic formed to the drum contour, operating

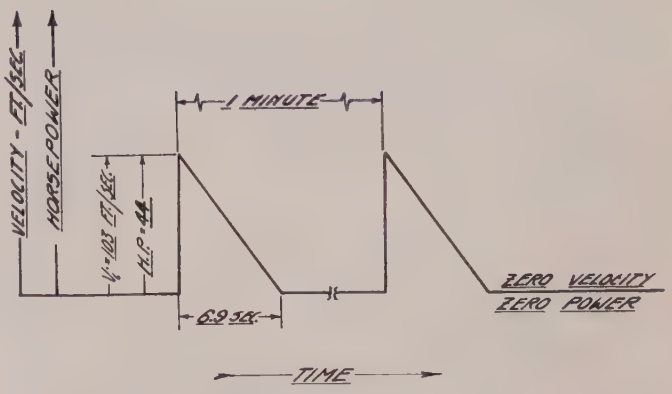


Fig. 17 – Graph of vehicle and brake stop-start horsepower and velocity cycle

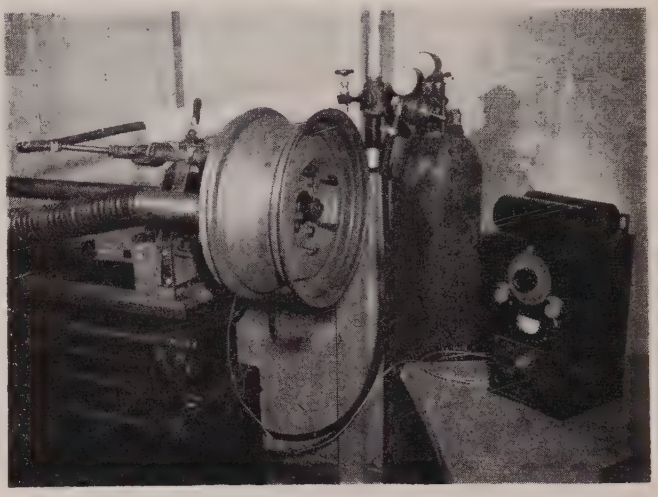


Fig. 18 – Test apparatus for conducting brake-drum heat-cycling test

Fig. 19 – Photograph showing failure condition of 4K6 cast-iron liner

Table 2 – Cast-Iron Liner Composition

Type of Liner-Iron	4K6	BDM 1	BDM 2	BDM 3
Total Carbon	3.5	3.1 - 3.4	3.6	3.1 - 3.4
Silicon	1.8	2.3 - 2.6	1.23	2.3 - 2.6
Sulfur	0.12	0.12 max	—	0.12 max
Phosphorous	0.4 - 0.65	0.2 max	0.05	0.2 max
Manganese	1.6 - 1.2	0.7 - 1.0	0.95	0.7 - 1.0
Chromium	0.3 max	0.1 - 0.3	0.08	0.25 - 0.3
Nickel	0.5 max	0.5 max	0.66	0.5 max

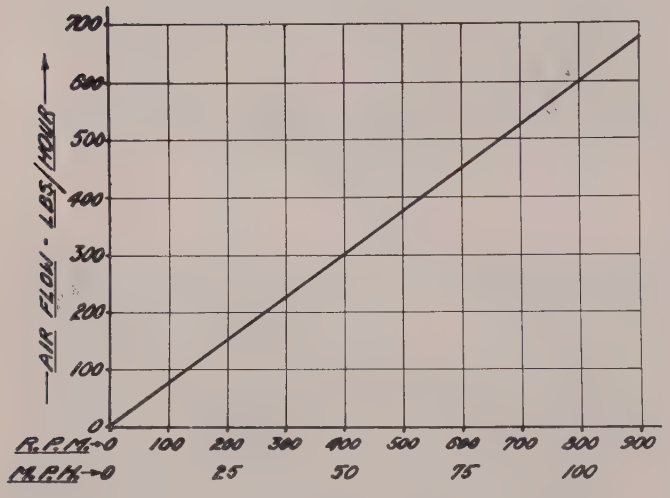


Fig. 20 – Airflow versus car speed for phase II brake drum with radial fins

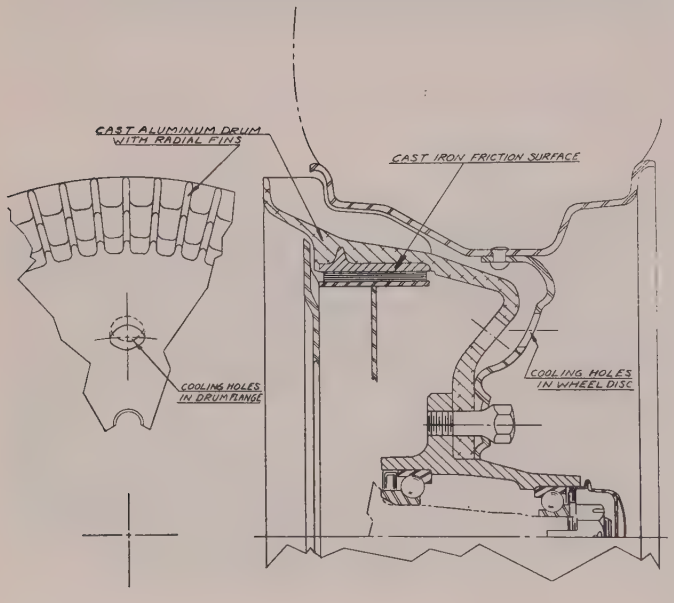


Fig. 21 – Design of phase II bimetallic 12 x 2¼-in. brake drum with radial fins

through a slip-ring arrangement. Data from one phase of the test, in which a constant airflow, simulating a 25½-mph velocity and heat input equivalent to 8 hp, permitted calculation of an overall heat-transfer convection coefficient of 0.3 Btu per deg F per hr per sq in. of inside brake-wall area. Using this heat-transfer factor and a Δt of 600 F (which is based on stabilized temperature readings that were made with a bimetallic, 11x2½-in. brake drum, stopping from 60 mph every 45 sec at a KE of 153,000 ft-lb per stop), we found that the heat-transfer equivalent for a 12x2-in. brake drum of 70 sq in. swept area is approximately 13,800 Btu per hr.

It is reasonable to assume that the balance of 6560 Btu input (20,360-13,800) would be dissipated by conduction to the wheel housing and by radiation to the surrounding air.

When phase II drums were first manufactured by Wellworthy, Ltd., it was thought that centrifugally spun cast iron similar to that being used for engine cylinder liners would be the ideal material for liners, and for a time the results were encouraging. The Aston-Martin car running in the 1949 LeMans 24-hr race with Al-Fin drums used this (British Standards Institution) type (4K6) cast-iron liner, the drums being in perfect condition at the finish. However, heavy checking and sometimes deep axial cracking began to appear on the rubbing surface of the drum after subsequent heavy use (Fig. 19).

In conjunction with the Wellworthy Iron Foundry, dynamometer tests were carried on to find the best liner material. As testing progressed it became apparent that, under heavy load applications, axial cracking was occurring. The bond was not fractured except in the immediate vicinity of these cracks, and it was first thought that the general design was at fault, particularly with regard to the relative thickness of the cast-iron and aluminum sections. In subsequent drums, the thicknesses of these components were varied. It was found that a thicker iron liner and larger mass of aluminum around in the liner could be utilized without fear of high hoop stresses resulting from the differential expansion and contraction between the two materials. However, variations in the design of the drums had no certain effect on the tendency of the iron liner to develop axial cracking, which occurred in some cases without any attendant checking or crazing of the rubbing surface. With these facts in mind, research was instigated with a view to producing a series of cast irons, for test purposes, having properties that would lend greater resistance to thermal shock and subsequent cracking. The first iron developed was called BDM 1, which was followed by a second development BDM 2, which was thought theoretically to be an improvement. BDM 3 cast iron is essentially the same in composition as BDM 1, but with the addition of chromium. The compositions of these various materials are given in Table 2.

The bimetallic drums used in the dynamometer testing of these cast irons were 7 in. in diameter on the braking surface and used iron liners 1¼ in. wide. A torque of 168 ft-lb was supplied to the back plate, and the line pressure was fully automatic to maintain this value. The drums were run up to 2800 rpm and stopped from this speed every 30 sec, the stopping time (4 sec) being constant throughout the test. These test conditions correspond to the degree of braking experienced on a front drum of a 3000-lb car stopping from 125 mph at an average deceleration of 24 ft per sec per sec. Air from a small centrifugal blower was directed into the drum during application.

From these tests it was learned that all three sand-cast-iron materials are superior to the spun 4K6 iron. The sand-cast 4K6, while performing generally well, did continue to show axial crack development. BDM 1 is generally satisfactory, especially when the aluminum alloy bonded to it is age-hardened. No axial cracking was experienced on any drum made of BDM 2, and when the aluminum alloy was age-hardened very little heat-spotting or crazing was noted. In fact, Al-Fin drums (Fig. 15) installed in the Mercedes 300-SL sport cars that did so well in the 1952 Pan-American 2000-mile road race (finishing first and second) and in the 1952 LeMans 24-hr road race, were made with an iron quite similar to the BDM 3.

At the LeMans (France) 1952 race, six out of the first 10 cars, including those placing first, second, third, and fourth, were equipped with Al-Fin bonded bimetallic brake drums. The Le-Mans 24-hr contest is an 8.6-mile closed course, and reportedly the most grueling road race in the world.

Aluminum-Alloy Muff and Fin Design

In general a fairly thick aluminum muff is considered necessary in order to supply a heat reservoir at least equal to that of the standard cast-iron drums, since in most passenger-car applications no appreciable flow of cooling air reaches the brake, due to its being shrouded by the wheel. Under these conditions, circumferential cooling fins cannot operate very effectively. Therefore, we decided to utilize the close coupling of the wheel rim to the brake drum by the addition of radial fins to the drum, and to use the wheel rim as the shroud for the brake, thereby achieving a low-velocity centrifugal air blower. The Mercedes drum (Fig. 15) is such a construction. Fig. 20 shows the airflow over a brake drum (with lateral fins similar to Figs. 14 and 21) for various car velocities. The airflow is not too great. However, along with adding lateral stiffness stability to the brake drum, this design helps to increase the convection conduction coefficient and thereby increases the recovery.

The aluminum alloy used for the body of the

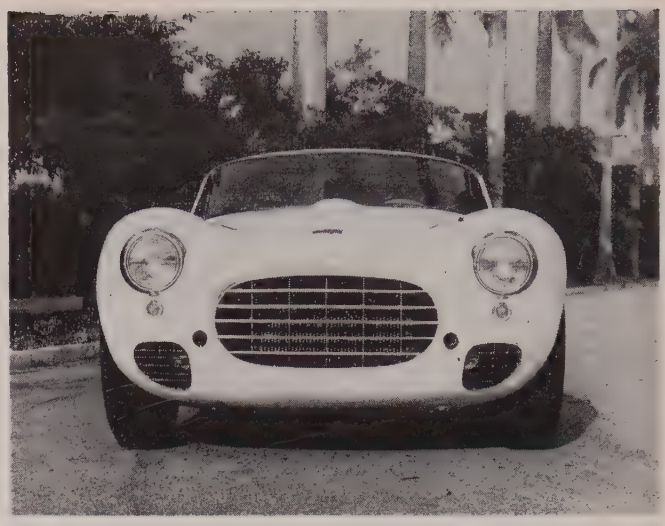


Fig. 22 – B. S. Cunningham sports car, showing air intake for drum cooling

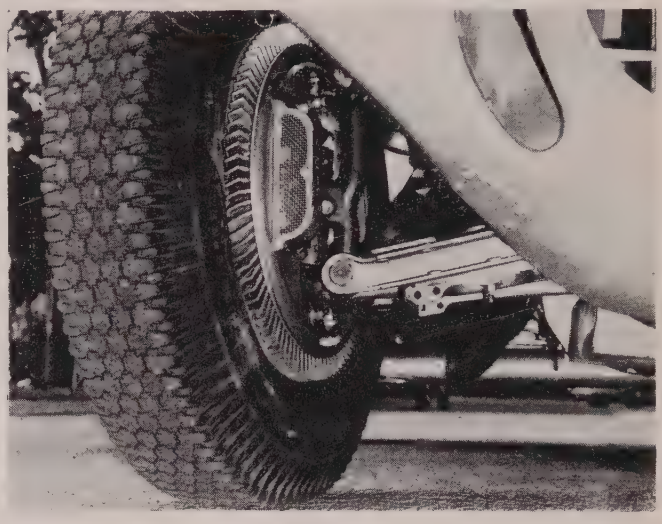


Fig. 23 – Air intake opening in backing plate for internal cooling of brake drum

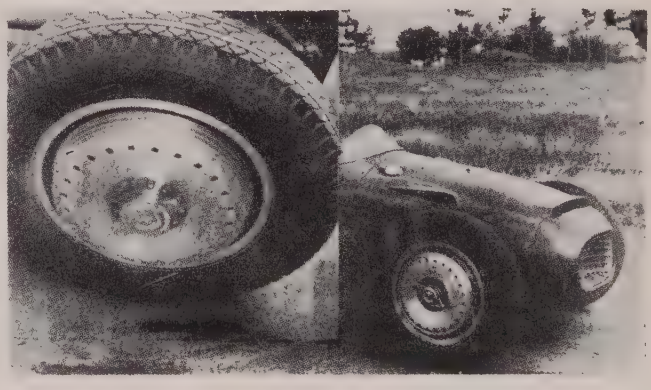


Fig. 24 – Brake drum located on outside of wheel

Table 3 – Aluminum-Alloy Compositions

Type of Alloy	RR 50	DTD 272	Silumin	L-33 Sodium Modified
Composition, %				
Copper	0.8 – 2.0	0.5 – 1.50	—	—
Magnesium	0.05 – 0.2	0.2 – 0.6	—	—
Silicon	1.5 – 2.8	4.5 – 5.5	12.5 – 13.5	11.0
Iron	0.8 – 1.4	0.6 max	—	0.2 max
Nickel	0.8 – 1.75	0.15 max	—	—
Titanium	0.05 – 0.25	0.25 max	0.15 max	—
Manganese	—	—	0.3 – 0.5	—
Aluminum	Remainder	Remainder	Remainder	Remainder
Ultimate Strength (As Cast), psi	24,000	22,000	—	26,000
Elongation, %	3	5.5	—	10
Brinell Hardness	65	94	—	—

drums is usually a medium-strength, medium-ductility alloy such as DTD272, RR50, silumin, or L-33 (refer to Table 3 for chemical and physical properties of these alloys). The basic requirement for a good brake-drum aluminum alloy is that it has high strength in the cast or stress-relieved condition and, of course, good strength properties at high temperatures.

For racing cars, every effort is made to take advantage of air circulation for both external and internal drum cooling. The drum flange is frequently drilled and air scoops attached to the brake back plate (see Figs. 22 and 23).

It is of significance that one builder of racing cars mounts his hydraulic cylinders on an extension of the kingpin housing and does away with back plates altogether. He thus eliminates the plate overhang and makes possible a more rigid drum since no provision is necessary for fitting the back plate to the drum at the inboard face. Another manufacturer goes to the extreme of fitting the brake drum to the outside of the wheel (Fig. 24). This, obviously, gets the brake drum out in the airstream where it will do the most good, although it is difficult to imagine any American passenger-car manufacturer adopting this technique. In such an event the buying public would probably want the stylist to cover it with streamlined sheet metal.

The designer of bimetallic brake drums must consider, along with the mechanical loading stresses, the thermal stresses resulting from the differences in thermal coefficients of expansion of aluminum and cast iron, and make certain that

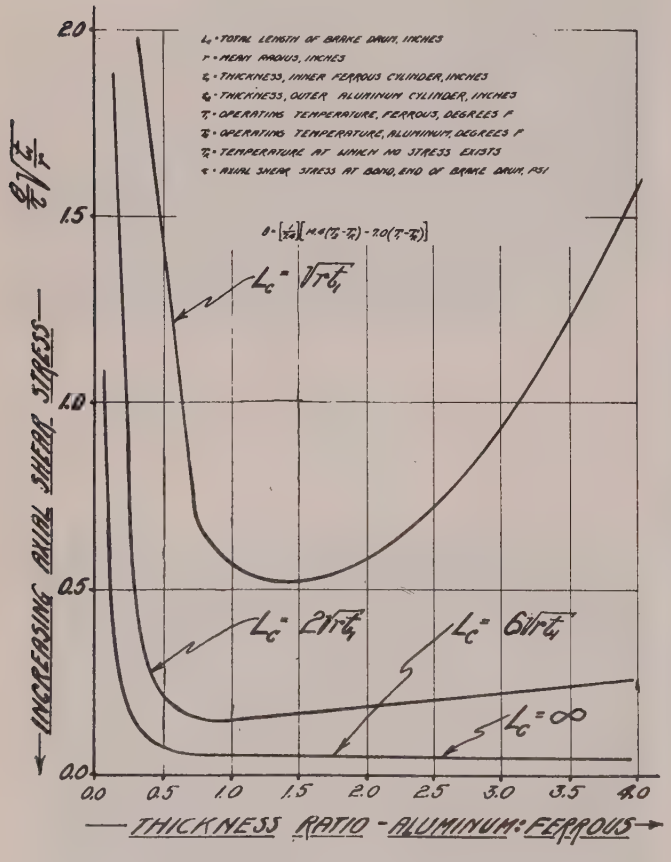


Fig. 25 – Graph showing axial shear stress, design factors for phase II bimetallic brake drum

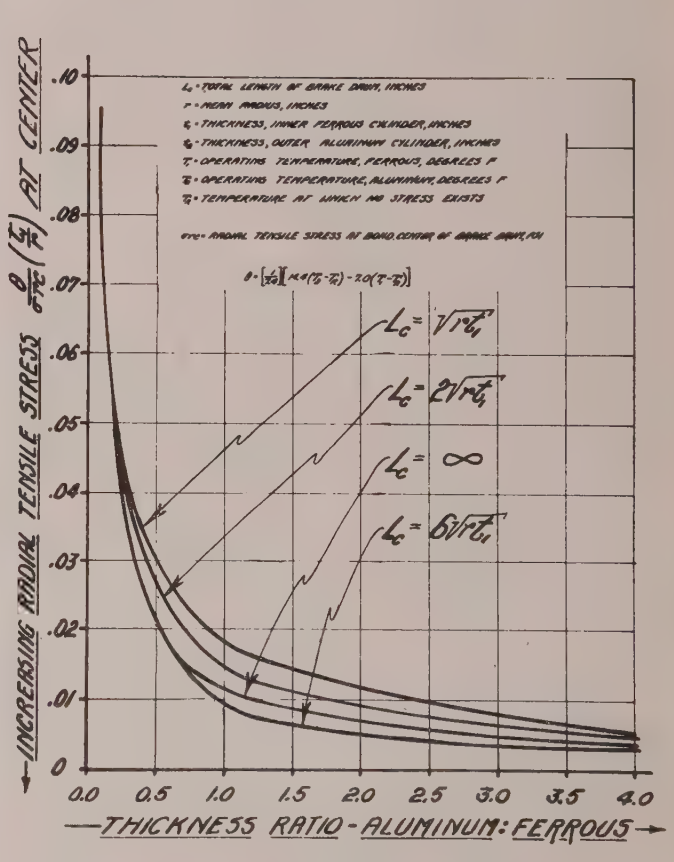


Fig. 26 – Graph showing radial tensile stress at center, design factors for phase II bimetallic brake drum

the optimum thickness relationship between cast iron and aluminum is used. We undertook at Fairchild a theoretical investigation to develop equations for the thermal stresses at the bond. We have been somewhat successful in substantiating this theoretical work by actual test. The theoretical calculations and stress equations are somewhat involved and lengthy and, for this reason, are not included in this paper, but the more important factors are included instead, in the form of graphs. Computed axial shear stresses at the bond are shown in Fig. 25. The curve illustrates, for a given temperature, the reciprocal of stress versus the cylinder thickness ratio for various brake-drum axial lengths. The curve is in parametric form, but the parameters are common ones that are readily derived from the known dimensions of the drum. Knowing the allowable bond shear stress, one can use the curve to determine the permissible operating temperature. Fig. 26 shows the radial stress at the center of the cylinder and Fig. 27 shows the radial stress at the ends of the cylinder. These stresses are tensile if the operating temperature is greater than the "reversal temperature," which is defined as the temperature at which there are no stresses at the interface. An examination of these curves will show that the radial stresses at the center of the cylinder are always greater than at the ends, although for very short cylinders these stresses are approximately equal. It can also be seen that the maximum shear stress occurs nearer the ends and that these stresses vary with the length of the cylinder, but if the cylinder is sufficiently long, they approach limiting values that do not change with further increase in length.

The thickness of the aluminum mounting flange of a phase II bimetallic drum has been more or less standardized at about ¼ in. for drums with brake surface diameters up to 12 to 14 in. That this thickness is adequate is shown by the fact that there have been no recorded failures. Also, in most cases, where holes for internal cooling are required, they are drilled in the aluminum web, with no increase in thickness being necessary.

It has been found best to attach the aluminum drum to the hub or wheel at the smallest possible diameter pitch circle since, at a diameter approaching that of the brake surface, distortion occurs on expansion, resulting in high spots on the brake surface opposite each retailing bolt. No difficulty has been experienced with fretting in the hub stud holes. It is essential, of course, to use a spacer or washer between the nut and aluminum drum housing to reduce the unit stress.

On test as the brakes are applied at regular intervals of time (t), receiving every time the same heat input (Q), the corresponding mean temperature rise per stop is $T = \frac{Q}{Wc}$, where W equals weight of the drum and c equals specific

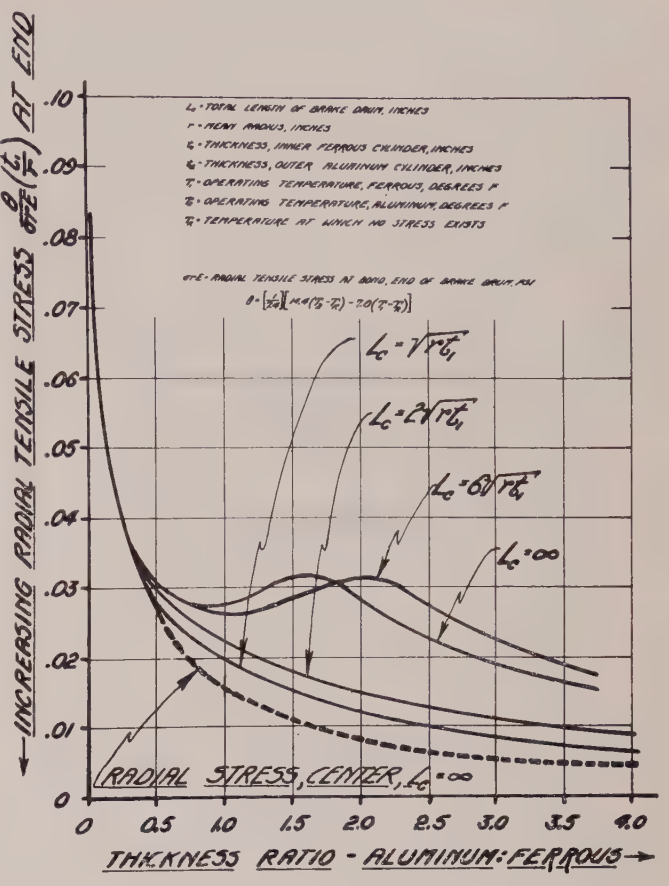


Fig. 27 – Graph showing radial tensile stress at end, design factors for phase II bimetallic brake drum

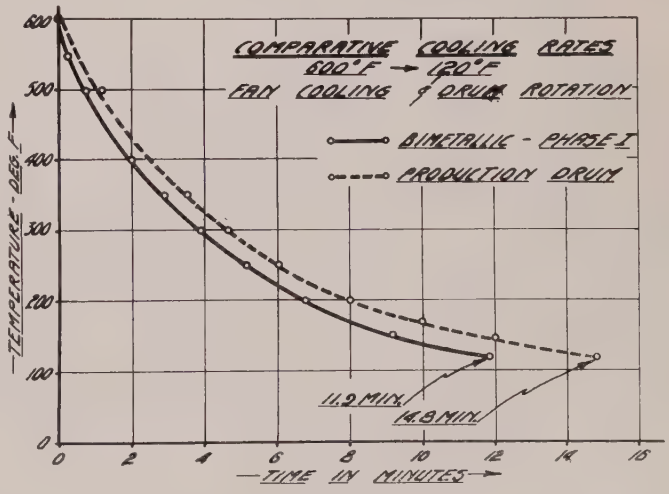


Fig. 28 – Graph of comparative cooling rates for standard production iron drum and bimetallic phase I drum

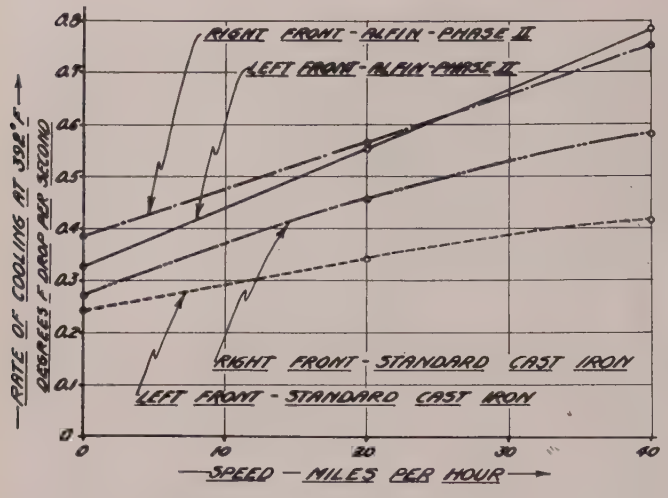


Fig. 29 – Comparative rate of cooling curves for standard cast-iron and bimetallic phase II drums

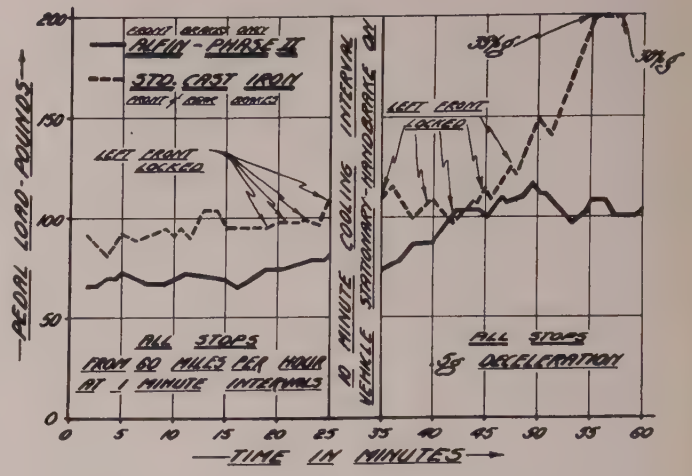


Fig. 30 – Comparative pedal-load graphs for standard cast-iron and phase II bimetallic drums

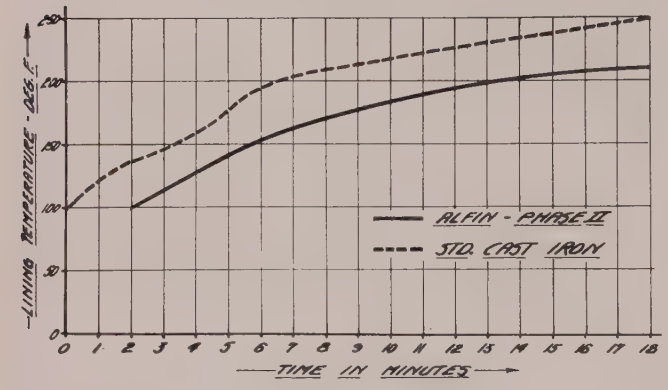


Fig. 31 – Comparative chart of lining temperature for standard iron and bimetallic phase II brake drum – right front brake. Thermocouple located in pocket in lining. All stops at 0.4*g* deceleration from 70 mph to 20 mph; stops made at alternate 1- and 2-min. intervals.

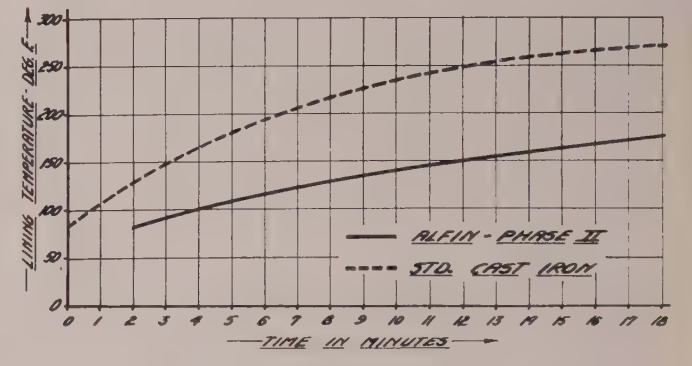


Fig. 32 – Comparative chart of lining temperature for standard iron and bimetallic phase II brake drum – right rear brake. Conditions same as in Fig. 31.

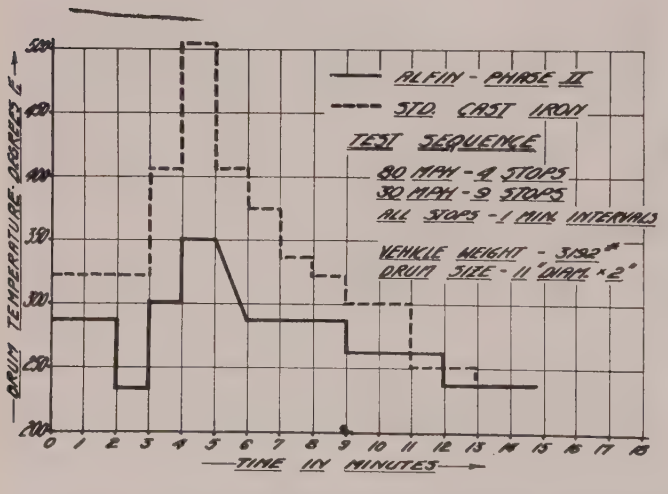


Fig. 33 – Comparative chart showing drum temperature for standard iron drum and phase II bimetallic type – right rear brake

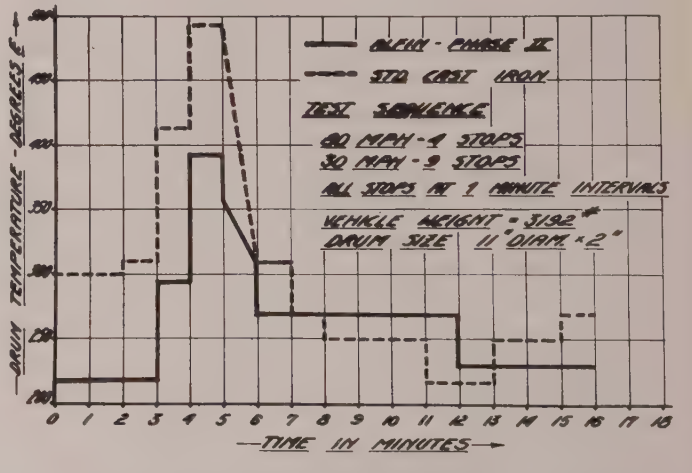


Fig. 34 – Comparative chart showing drum temperature for standard iron drum and phase II bimetallic type – left front brake

heat of the material. Aluminum would appear to have quite an advantage over cast iron because of aluminum's higher specific heat (approximately 2/1). On the other hand, the lower specific gravity of aluminum (approximately one-third that of iron) does modify somewhat the advantage of the higher specific heat. However, we should consider this analysis on the basis of volume of material instead of weight since, to obtain comparative strength and rigidity, the aluminum because of its lower modulus is usually designed heavier in section. Therefore, the combination of these factors still favors the aluminum bimetallic over the iron drum. Of course, mechanical stresses due to shoe loading and deflection dictate to a large extent the cross-section required. We know that the cast-iron liner surface is subjected to various stresses that are functions of the high instantaneous surface temperature. To minimize this high temperature, it is necessary to spread the contact evenly between brake shoe and drum. This explains the necessity for a concentric drum and the utilization of high-conductivity material, since the latter permits rapid flow of the surface heat into the brake-drum housing. We believe that this ability of aluminum to conduct heat rapidly from the hot contact surface is responsible, to a large extent, for the efficient operation of aluminum bimetallic drums, especially in installations where optimum cooling by air convection is not possible.

Unfortunately, the amount of heat lost through convection during the actual process of braking is negligible. To confirm this fact, consider the case of an 18 ft per sec per sec stop from 60 mph, ideally requiring 4.6 sec to stop. The cooling curves (from 600 F) for a standard iron drum and a bimetallic phase I drum (Fig. 28) show that the temperature loss during this interval is less than 20 F. This helps to illustrate that the property of fast recovery between stops is an extremely valuable one in that it permits the drum to resume braking at a lower temperature level and not rise to extreme temperatures with continued use. These curves also show that the bimetallic drum returned to a temperature of 120 F approximately 19.5% faster than the standard drum. Both drum temperatures were taken at a location 0.010 to 0.015 in. from the friction surface.

Temperature Comparative Tests

Comparative tests on a Riley 2½-litre car were run to decide whether the fade characteristics of the aluminum bimetallic circumferentially finned drum were different from those of a standard cast-iron drum. These tests indicate that at 392 F the bimetallic drums have the highest rate of cooling (Fig. 29). The results of fade tests with bimetallic drums on the front, together with standard cast-iron drums on the rear, are shown in Fig. 30. These results are more conclusive and indicate that the bimetallic phase II drums required less pedal effort

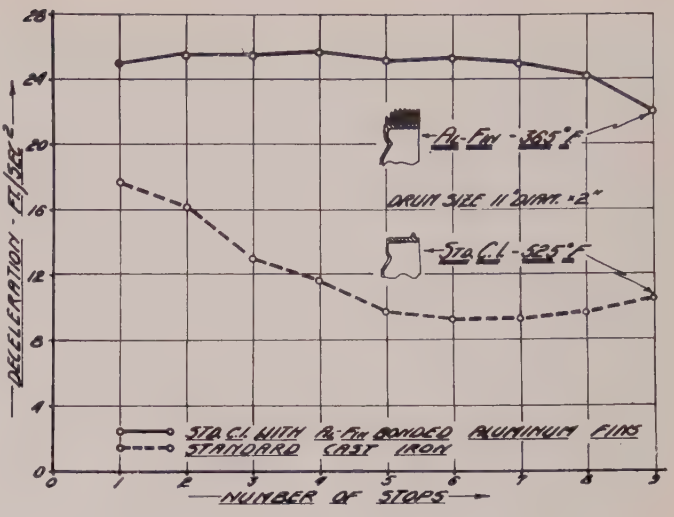


Fig. 35 – Comparative deceleration curves for 3-min cooling cycle, comparing standard iron drum with phase I bimetallic type. Decelerations from 60 mph, with successive stops; 600-psi line pressure; 3-min cooling in still air between stops

since they did not fade, and that the standard cast-iron drums faded badly during the second half of the test.

Figs. 31 and 32 represent additional comparative cooling data for 0.4*g* decelerations from 70 mph to 20 mph made at 1 and 2-min intervals. These test results also indicate the cooling superiority of the Al-Fin phase II brake drum. Additional comparative efficiency and recovery tests are illustrated by Figs. 33 and 34.

Supplementary comparative heat-dissipation tests were conducted with standard iron 11x2-in. drums and with the same type of drum, machined to a uniform rim thickness before bonding and

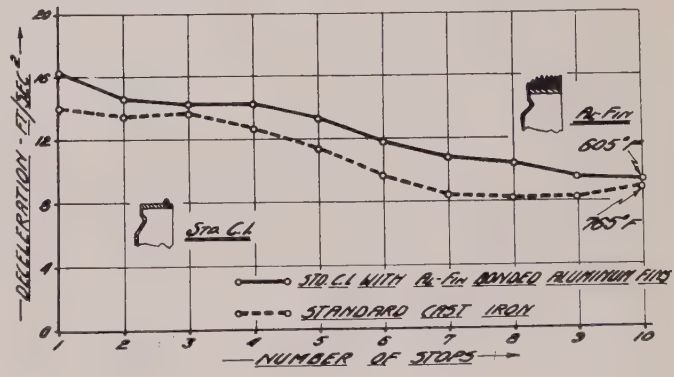


Fig. 36 – Comparative deceleration curves for 1-min cooling cycle, comparing standard iron drum with phase I bimetallic type. Deceleration from 60 mph, with successive stops; 600-psi line pressure 1-min cooling in still air between stops; 11 x 2 drum size

casting on the aluminum circumferential fins (phase I type). These tests were made to evaluate any marked improvement in brake performance under adverse cooling conditions, and the brakes were cooled in still air. Simulated stops were made from 60 mph. At the completion of each stop the drum was accelerated to 60 mph and allowed to coast for a given period of time. The results of these tests are shown in Figs. 35 and 36, the former for a total cooling time of 3 min from the end of one stop to the beginning of the next, and the second graph for a cooling time of 1 min. It will be noted that the heat dissipation and deceleration properties of the finned drum are better than for the cast-iron drums when plotted with the same number of consecutive stops under similar conditions. Also note the large advantage of recovery time, when comparing the 3-min to 1-min graphs.

Antisqueal Properties

It has been found that the bonded bimetallic brake drums, both on passenger cars and buses, help considerably in the elimination of squeal. Brake squeal is quite a problem at the present time, especially on city buses. Al-Fin drums have, in many cases, completely eliminated squeal; and in England quite a few sets of bimetallic brake drums are sold for this feature alone, even though the standard cast-iron drums were giving adequate braking service. The ability of the bimetallic structure to eliminate objectionable audio frequencies is probably due to the superior dampening characteristics of aluminum as compared to cast iron.

The test data that have been presented are the results of the interest of many companies in this development. The assistance and encouragement of B. S. Cunningham Co.; Bristol Aeroplane Co., Ltd. (Car Division); Kelsey-Hayes Wheel Co.; Motor Wheel Corp.; Lockheed Hydraulic Brake Co., Ltd.; Ferodo, Ltd.; Aston-Martin, Ltd.; Daimler-Benz A.G. (Mercedes); Buick Motor Division and Cadillac Motor Car Division of General Motors Corp.; Auto Specialties Mfg. Co., Inc.; Karl Schmidt G.m.b.H.; and Wellworthy, Ltd. are gratefully acknowledged. The authors also wish to acknowledge the assistance of D. Pearce of the Fairchild Engine Division for his help in the preparation of design calculations.

DISCUSSION

Considers Design Approach To High-Speed Braking Problem

—C. R. Lupton
Bendix Aviation Corp.

IT was interesting to note the relative effectivness of forced aircooling at 60 and 100 mph in the case of the Cunningham sports car. This has some real significance in our problem of high-speed braking.

In reference to the 2-master-cylinder arrangement, it is not clear just how the change in front to rear brake proportioning is adjusted to accommodate the change in tire-to-road coefficient of friction. We believe, however, that it has been quite generally and consistently established that, from a practical standpoint, optimum brake distribution is attained on all road surfaces when, as is universally done today, brake proportioning is based upon "dynamic" axle loading.

Considering relatively wide brakes of small diameter, we are evaluating the centermount type of drum having its support on the outside diameter at the midpoint of the drum ring, rather than at the edge of the ring, in an attempt to minimize bellmouthing.

The need for better means of dissipating heat from the brakes of modern passenger cars has spurred us into action on the problem, particularly in the past two years.

Formerly, vehicle manufacturers were reasonably content with braking systems capable of stopping the car about 7 consecutive times from 70 mph without complete loss of pedal or requiring pedal effort in excess of 200 lb.

Now, in view of the increased potential speed of passenger cars, some of which will attain an honest 100 mph, the demands are for braking systems that will provide a number of consecutive stops from 100 mph.

Although the difference in velocity is only 30 mph, the difference in the kinetic energy to be dissipated is in the ratio of 1/2. For example, the kinetic energy stored in a 5000-lb vehicle traveling 70 mph is 820,000 ft-lb, while at 100 mph it is 1,670,000 ft-lb.

Some of the limitations on the number of consecutive high-speed stops obtainable are:

1. Brake pedal to the floor. This can be accounted for by a number of conditions, some of which are:

A. Thermal and mechanical expansion of the brake drums.

B. Mechanical deflections in the pedal-to-master-cylinder hookup.

C. Fluid displacement losses in the hydraulic system due to high line pressure.

D. Vaporization of fluid in the hydraulic system.

2. Exceedingly high pedal effort required.

A. This is generally caused by fading of the brake lining.

3. Unbalanced braking.

A. Dangerous pulls to right or left can be caused by:

1. Unstable friction material.

2. Unequal heating or cooling of the brakes due to crosswinds.

In most cases the above conditions can be minimized by rapid dissipation of heat from the brakes by:

1. Ribbed, finned, or bimetallic drum construction.
2. Forced aircooling of the brake drums.
3. Moving the brake and drum out of the wheel cavity into the slipstream.

Considering the brake design approach to the problem, our experience with various designs of radially expanding drum brakes and axially expanding disc types has shown that definite improvement can be made. However, without close attention to the drum construction as a means of dissipating the heat generated, the brake itself may be disappointing. We have found that an unusually high number of fade stops can be made with a powered, relatively low-effectiveness brake, only to find that the friction material has burned to a crisp. We recognize the need of an improved drum in this case.

Now, considering the brake-drum approach to the problem, a dynamometer test was made comprising three drum types, all tested with a standard duoservo 11-in. brake.

The drums were:

1. Conventional cast-iron ring with stamped-steel drum back, weighing 16 lb.
2. Heavy-section, single-rib, cast-iron ring with stamped-steel drum back, weighing 26 lb.
3. Axially or radially ribbed, all-cast-iron drum with ribs extending down the drum back to the mounting flange, weighing 21 lb.

One phase of the work was a 100-mph fade and recovery test.

Three 460,000 ft-lb KE stops from 100 mph at 15 ft per sec per sec were made at 2-min intervals, followed by 3 min of cooling, then 10 stops from 30 mph at 15 ft per sec per sec at 1-min intervals.

At the end of the third 100-mph fade stop, drum temperatures taken at 0.090-in. from the rubbing surface were:

815 F for the conventional drum.
760 F for the heavy drum.
600 F for the ribbed drum.

Temperatures after the 3-min cooling period were:

485 F for the conventional drum.
470 F for the heavy drum.
340 F for the ribbed drum.

Temperatures after the 10 recovery stops were:

340 F for the conventional drum.
380 F for the heavy drum.
240 F for the ribbed drum.

Actually, the curves are rather parallel. The significant fact is that the ribbed drum did not attain nearly as high a temperature as the other two by the end of the third fade stop. All tests were run with a moderate airstream from a fan.

In conclusion, it appears that an ideal drum is one having the capacity to absorb and store considerable heat. This is of value during low-speed stops, when forced cooling is ineffective, but appears to be even more desirable as a means of "ironing out" the effects of crosswinds and other such factors that bring about unequal cooling tendencies and consequent unequal brake effectiveness at high speeds. The drum should also have maximum cooling area in the form of ribs or vanes that would serve as an impellor to deflect the airstream over the drum surface, either through or from ventilating holes or louvers in the wheel for high-speed braking.

This ideal, of course, can be recognized as a rather specific compromise, arrived at through a consideration of cost and mass-production problems, as well as those of performance under all of the operating conditions of the American automobile. There may well be a number of other compromises of equal merit, and this will not be surprising because, after all, we have to sell the brake, brake drum, and wheel combination, or no benefit will be derived by anyone except the engineers.

ORAL DISCUSSION

Reported by M. A. Forester
Packard Motor Car Co.

O. J. Wolfer, Pure Oil Co.: Have you done any work using magnesium?

Mr. Ladd: We have not done any work with magnesium as we have not found a way of bonding magnesium and iron together to form a very high strength bond.

A. V. Lorch, Kaiser Aluminum & Chemical Sales, Inc.: Will you please give us your thought on the relative value of circumferential versus radial ribbing?

Mr. Ladd: If the drums were allowed to operate in open air with no intereference from parts of the wheel-tire assembly or sheet-metal body work, circumferential fins would operate very effectively. However, because in most American passenger-car designs the brake drums are placed well inside the wheel structure, radial ribbing does a better job of cooling because we can design the radial type of brake drum to act as a low-velocity air blower and thereby create a higher convection conductive heat-transfer coefficient.

C. L. Burton, Aluminum Company of America: Our experience has shown that aluminum drums do a good job when they are located so as to be in the path of air movement. They do not have any particular charcteristics when used in still air.

Mr. Wolfer: How about using water on the drums to dissipate heat?

Mr. Ladd: If this could be done it would be wonderful. I know that some large trailers on the West Coast carry 500 gal of water and spray water on the brake drum. However in freezing cold weather, the trailer behind sometimes gets into trouble because of ice formation on the road resulting from the water condensation from the preceding vehicle.

Question: Have you ever noticed any great difference in squeal characteristics of bimetallic drums versus standard steel centrifused cast-iron drums?

Mr. Ladd: Yes, we found a definite improvement in squeal characteristics of the bimetallic aluminum drums. They operate much more quietly. Unfortunately, all of our evaluations have been made with buses. I believe that this is due to the phenomenon of aluminum having greater dampening characteristics.

TITANIUM-ALLOY

THE object of this paper is, first, to review the metallurgical principles governing the development of titanium alloys, and, second, to present some preliminary results of work in progress at the Armour Research Foundation on the development of titanium alloys for use at elevated temperatures.

Commercially pure titanium will undoubtedly find many useful applications in areas where its excellent resistance to corrosion and good forming and welding characteristics are most important. However, the major utility of titanium will be in applications where the strength-to-weight ratio of its alloys will be the governing factor.

The present commercial titanium-base alloys demonstrate the advantages to be gained by their substitution for other ferrous and nonferrous alloys in many applications. Strengths of up to 170,000-180,000 psi can be obtained in some of the current titanium-base alloys; however, the higher strength levels are usually accompanied by low ductility. Problems in the forming and welding of the stronger alloys also tend to reduce their acceptability. Many of these problems will be solved as production techniques are developed; however, drawing an analogy from the history of other alloy systems, alloys having superior properties in cer-

---

The purpose of this paper is twofold:

1. To review the metallurgical principles governing the development of titanium alloys.

2. To present some preliminary results of work in progress at Armour on the development of titanium alloys for use at elevated temperatures.

To develop superior alloys requires, the authors say, an intelligent and systematic approach based on the following:

1. Knowledge of the equilibrium phase relationships of titanium with other metals.

2. Information regarding the kinetics of reactions in the solid state.

Use of this approach has, they add, resulted in the development of titanium-base alloys having excellent properties at temperatures up to 1020 F.

The Authors

M. HANSEN is chairman of the Metals Research Department of the Armour Research Foundation of Illinois Institute of Technology. He received his Ph.D. degree in Germany. He joined the staff of the Illinois Institute of Technology in 1947, and has had his present position since 1951.

H. D. KESSLER is supervisor of Nonferrous Metals Research, Armour Research Foundation of Illinois Institute of Technology. He received his bachelor's degree in metallurgical engineering from Case Institute of Technology and his master's degree in the same field from Illinois Institute of Technology. He has been with Armour since 1946.

DEVELOPMENT

M. Hansen and H. D. Kessler,
Armour Research Foundation of Illinois Institute of Technology

This paper was presented at the SAE Annual Meeting, Detroit, Jan. 16, 1953.

tain applications will join and possibly even replace some of the present commercial alloys.

The development of superior alloys requires an intelligent and systematic approach based on the following: (1) knowledge of the equilibrium phase relationships of titanium with other metals, and (2) information regarding the kinetics of reactions in the solid state. Thanks to the farsighted research programs of the Air Force and the Ordnance Department, much of this fundamental information is available and more is being obtained in the various research laboratories throughout the country.

Phase Relationships

Most of the important binary phase diagrams and some of the ternary diagrams of titanium either have been or are in the process of being determined. As the ternary phase relationships can be generally quantitatively predicted from the binary diagrams, knowledge of the binary systems is of prime importance. The following discussion will, therefore, outline the three basic types of solid-state phase relationships obtained in binary titanium-base alloys. These are classified principally as to the effect the addition element has on either raising or lowering the transformation temperature (885 C) of the pure metal, and the phase changes that immediately follow at the high titanium end of the systems. Above 885 C, the beta phase (body centered cubic) is at equilibrium. Below that temperature the alpha phase (hexagonal close packed) exists.

The first type of system is represented by Fig. 1. The binary systems of titanium with molybdenum,[1] columbium,[1] vanadium,[2] and tantalum[3] are known to belong to this group. This phase diagram is characterized by a continuous series of beta solid solutions with the formation of an alpha plus beta field, which grows wider with decreasing temperatures.

The photomicrographs included in the figure illustrate the structural appearance of the various phase fields. It should be noted that the structure of the two alloys quenched from the beta field is different. The structure of the alloy to the left of the broken vertical line (R) shows acicular transformed beta, generally known as alpha prime; whereas the structure of the alloy to the right of R consists entirely of retained beta. In the former case the supersaturated beta phase partially decomposed, to give alpha prime, which, similarly to the martensite of steels, is formed by a diffusionless transformation. X-ray examination shows this nonequilibrium product is a distorted version of alpha, apparently of the same composition as the beta solid solution from which it is formed. All alloys quenched from the beta field to the left of R will have structures showing alpha prime. Also, all alloys heat-treated in the alpha plus beta field at temperatures above T will consist of isothermal alpha plus alpha prime. Alloys heat-treated below T in the alpha plus beta field will consist of alpha plus retained beta.

The second type of system (represented in Fig.

[1] See *Journal of Metals*, Vol. 3(191), October, 1951, pp. 881-888: "Systems Titanium-Molybdenum and Titanium-Columbium," by M. Hansen, E. L. Kamen, H. D. Kessler, and D. J. McPherson.

[2] See Transactions of ASM, Vol. 44, 1952, pp. 990-1003: "Titanium-Vanadium System," by H. K. Adenstedt, J. R. Pequignot, and J. M. Raymer.

[3] See *Journal of Institute of Metals*, Vol. 81, 1952/1953, pp. 73-76: "Constitution of Tantalum-Titanium Alloys," by D. Summers-Smith.

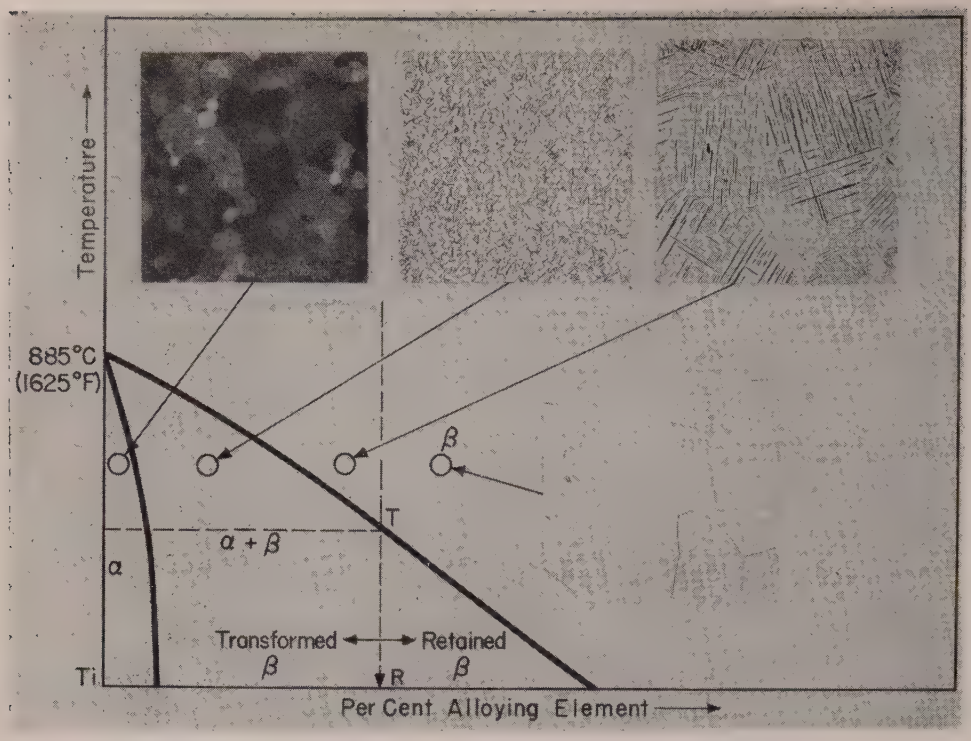


Fig. 1 – Solid-state equilibria for type 1 binary titanium-base alloy systems showing typical microstructures of various phase fields

2) is typical of the binary systems of titanium with chromium,[4] iron,[4] silicon,[5] nickel,[6] copper,[7] and hydrogen.[8] For most practical purposes this type of system is similar to that of Fig. 1, and similar structural features are noted. However, below the eutectoid temperature the beta phase decomposes into alpha plus an intermediate phase, gamma. In most alloys of these systems, the eutectoid reaction is of little practical importance, as it does not occur within the period of practical heat-treatment. The eutectoid reaction must be considered, however, if the alloys are to be exposed to elevated temperatures for long periods.

The third type of phase diagram is shown in Fig. 3. The known members of this group are the binary systems of titanium with aluminum,[9] oxygen,[10, 11] nitrogen,[11] and carbon.[11] Whereas the first three members of this group show extensive solubility of the addition element in alpha titanium, the maximum solubility of carbon is less than 0.5%. Aluminum, the only metal in this group, is also the only member showing substitutional solubility; oxygen, nitrogen, and carbon being interstitially soluble in titanium. The major characteristics of this type of system is that the addition element raises the transformation temperature of titanium and thereby stabilizes the alpha phase to higher temperatures. As noted in the photomicrographs, which are only typical of titanium-aluminum alloys, the beta phase is not retained, regardless of composition, on quenching from the beta field. In the titanium-aluminum system the serrated type of transformation product shown is produced on quenching; whereas in the oxygen, nitrogen, and carbon alloys an acicular product is obtained on similar heat-treatment.

Transformation Characteristics

The key to understanding the heat-treating principles in titanium-base alloys lies in recognizing that the same solid-state reactions occur that are basic to established commercial ferrous and

[4] See Transactions of ASM, Vol. 44, 1952, pp. 974-989: "Systems Titanium-Chromium and Titanium-Iron," by R. J. Van Thyne, H. D. Kessler, and M. Hansen.

[5] See Transactions of ASM, Vol. 44, 1952, pp. 518-538: "Titanium-Silicon System," by M. Hansen, H. D. Kessler, and D. J. McPherson.

[6] See U. S. Bureau of Mines Publication RI 4463 (1949), "Tentative Titanium-Nickel Diagram," by J. R. Long E. T. Hayes, D. C Root, and C. E. Armantrout.

[7] See *Journal of Metals*, Vol. 4, July, 1952, pp. 766-770: "Titanium-Copper Binary Phase Diagram," by A. Joukainen, N. J. Grant, and C. F. Floe.

[8] See Proceedings of Royal Society of London, Series A, Vol. 204, 1950, pp. 309-323: "Experimental and Thermodynamic Investigation of Hydrogen-Titanium System," by A. D. McQuillan.

[9] See *Journal of Metals*, Vol. 4, June, 1952, pp. 609-614: "Titanium-Aluminum System," by E S. Bumps, H. D. Kessler, and M. Hansen.

[10] See Transactions of ASM, Vol. 45, 1953, pp. 1008-1028: "Titanium-Oxygen System," by E. S. Bumps, H. D. Kessler, and M. Hansen.

[11] See *Journal of Metals*, Vol. 188, October, 1950, pp. 1261-1266: "Alloys of Titanium with Carbon, Oxygen, and Nitrogen," by R. I. Jaffee, H. R. Ogden, and D. J. Maykuth.

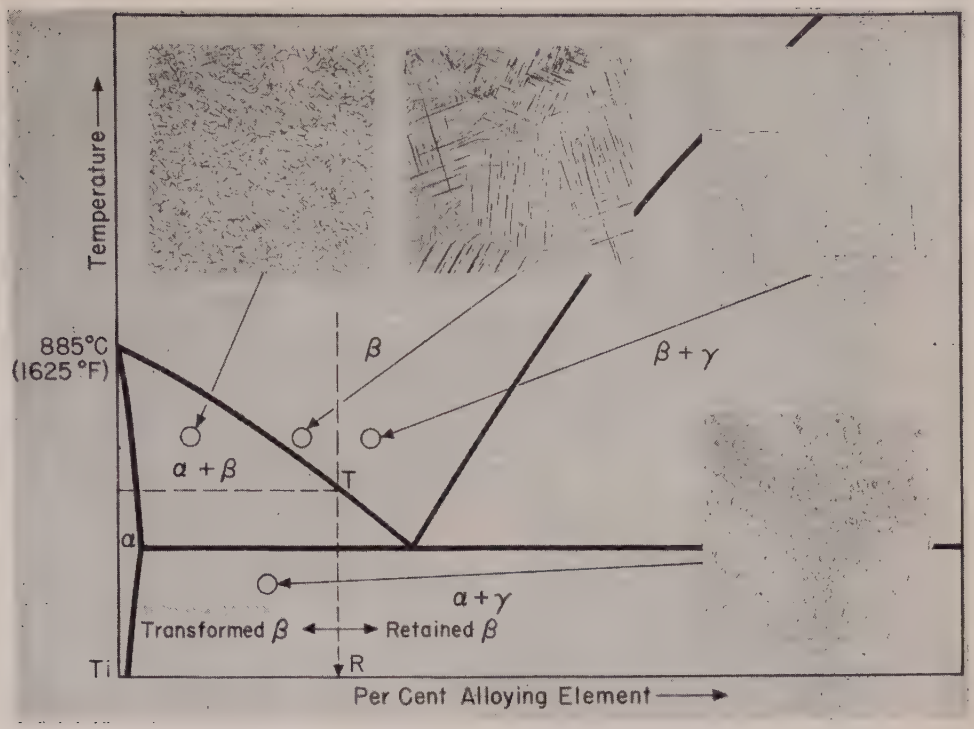


Fig. 2 – Solid-state equilibria for type 2 binary titanium-base alloy systems showing typical microstructures of various phase fields

nonferrous alloys. Response to heat-treatment results from the instability of a high-temperature phase at lower temperatures. This instability can manifest itself in three possible ways, depending upon the alloy system and composition: (1) by precipitation of a new phase from supersaturated beta; (2) by the eutectoid decomposition of beta to two new phases; and (3) by transformation to some metastable structure such as alpha prime. One or more of these reactions can be recognized and utilized in each of the heat-treatable titanium alloys.

Fig. 4 shows schematic hardness versus composition curves for two types of binary titanium-

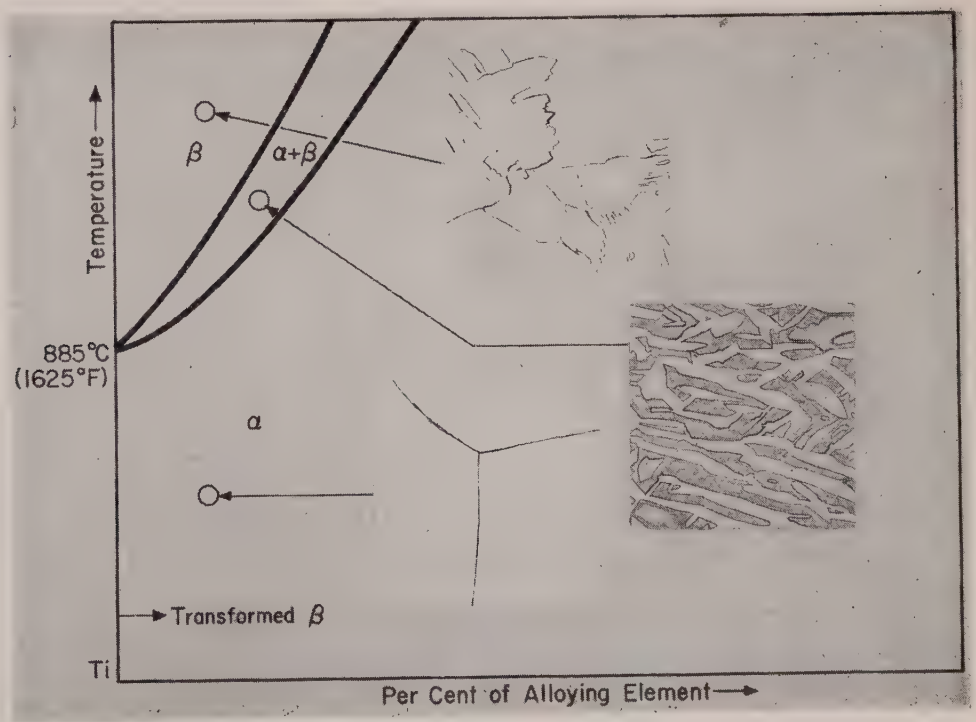


Fig. 3 – Solid state equilibria for type 3 binary titanium-base alloy systems showing typical microstructures of various phase fields for titanium-aluminum alloys

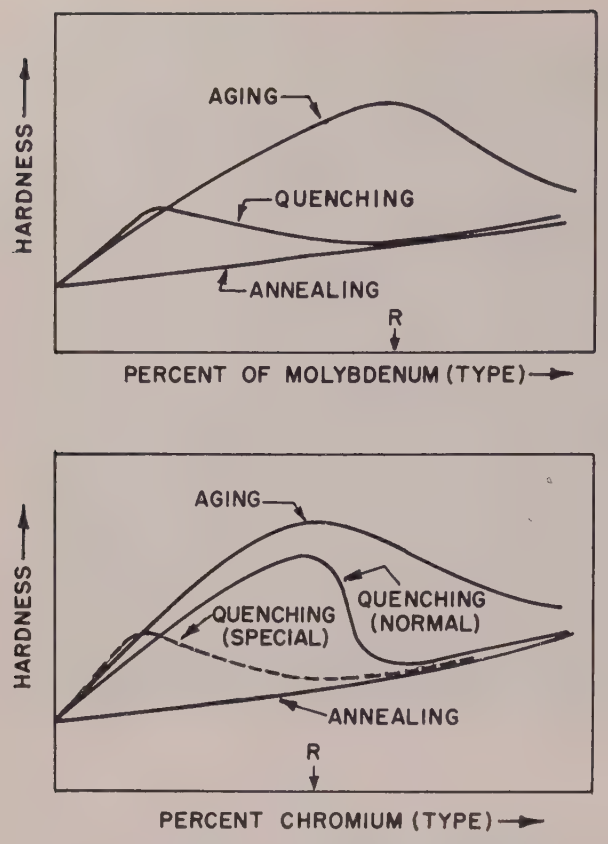


Fig. 4 – Hardness versus alloy content curves illustrating influence of various heat-treatments on properties of molybdenum (type 1) and chromium (type 2) alloy systems

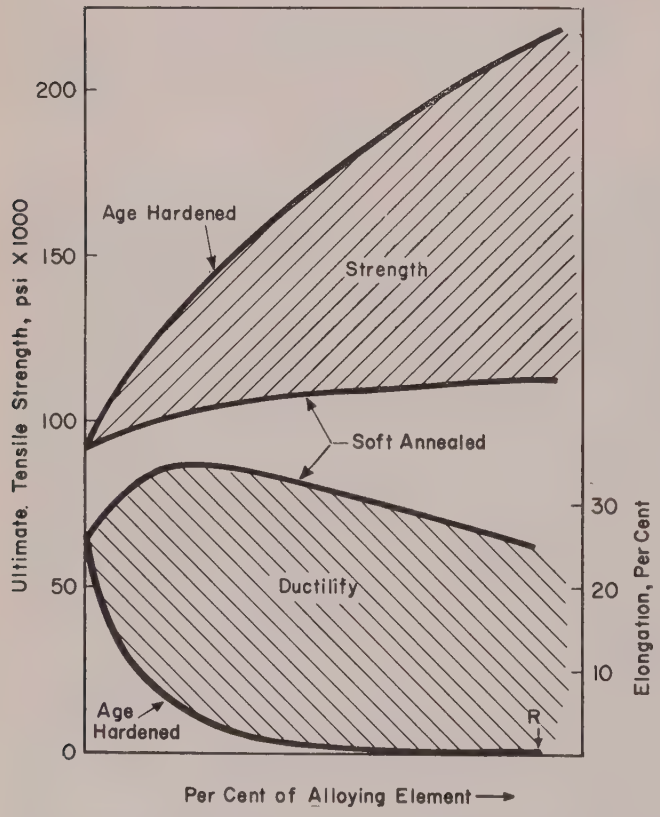


Fig. 5 – Tensile properties versus alloy content illustrating range of properties obtainable by heat-treatment for either type 1 or type 2 alloy systems

base alloys. The two diagrams illustrate the hardness changes that can be produced by various heat-treatments of the type 1 and type 2 phase diagram groups. The upper curve, shown for the titanium-molybdenum-type system, indicates that very little hardness increase can be expected by water quenching from the beta field. The peak that occurs at low alloy contents represents only a small hardness increase with respect to the "annealed" curve. This small hardness increase can be attributed to the formation of alpha prime, indicating that this martensitic product is relatively soft. To the right of the quenching peak, increasingly greater amounts of soft retained beta coexist with alpha prime to the composition *R*, where the minimum hardness occurs and the structure consists entirely of retained beta. The heat-treatability of this group of alloys is entirely dependent on the rejection of alpha from the metastable beta phase. To obtain an optimum combination of strength and ductility, either quench-and-age or isothermal-quench-and-age types of treatment could be applied depending on the composition of the alloy.

The lower diagram of Fig. 4 shows that, on water quenching titanium-chromium-type alloys, a hardness peak occurs in the region of *R*, and that this hardness can be only slightly increased by an additional aging treatment. For the sake of definition, the hard phase obtained at the quenching peak will be referred to as beta prime. The high hardness is believed to be associated with the precipitation of alpha from the normally soft retained beta phase. The fact that no alpha precipitate is observed in beta prime structures, as the equiaxed grains appear identical to soft retained beta, indicates that the high hardness may result from a preprecipitation phenomenon of some sort. In the titanium-chromium system, the normal water-quenching process is not sufficiently drastic to suppress the preprecipitation reaction. In some recent work at the Armour Research Foundation, only by a drastic quench in iced brine was it possible to suppress the beta prime reaction. The broken curve in Fig. 4 illustrates the results. Similar experiments performed with equally small samples of titanium-chromium alloys based on magnesium-reduced titanium gave only the hard beta prime structure, indicating that contamination by the interstitially soluble elements increases the rate at which beta prime is formed.

By step quenching, beta prime has also been observed to occur in the titanium-molybdenum system,[12] as preceding the appearance of precipitated alpha phase. On the basis of these results it would appear that differences in the heat-treatment

[12] "Progress Reports on Flight Research Laboratory Contract No. AF 33(038)-16347," by D. J. LeLazaro and W. Rostoker. Unpublished work at Armor Research Foundation.

between the titanium-molybdenum and titanium-chromium alloys are due to variations in reaction rates rather than the actual mechanism of hardening.

Fig. 5 shows the range of tensile properties possible for the type 1 and type 2 alloys. The degree of heat-treatability increases considerably with increasing alloy content to the composition *R*. The maximum strengths are obtained by age hardening; however, the higher strengths are accompanied by very poor ductility values. One of the main objects of the present research on the transformation kinetics of titanium-base alloys is the determination of the particular type of heat-treatment that will give the best combination of tensile strength and ductility.

Fig. 6 illustrates the range of tensile properties possible in titanium-aluminum alloys. As yet there is no information available on the heat-treatability of the other binary alloys of this phase diagram group. As shown in Fig. 6, titanium-aluminum alloys can be hardened somewhat by water quenching. The process by which this hardness increase occurs is not yet fully understood. In general, there is a moderate strength increase with increasing aluminum content, and a commensurate loss in ductility.

Borrowing a tool popular among ferrous metallurgists, it has been found that a standard Jominy end-quench test is very useful in giving a broad picture of transformation structures and hardnesses with progressively slower cooling rates. (See Fig. 7.) The various types of hardening processes can be detected by the end-quench procedure. A quench-hardenable alloy in which alpha prime is the only contributor to hardness gives a curve similar to that obtained for steels and shows the highest hardness at the quenched end. An alloy in the region of the composition of *R* (Fig. 4), which hardens only by aging, gives a curve having a hardness minimum at the quenched end. Alloys of intermediate compositions, between the alpha prime and aging peaks, show a hardness maximum a short distance from the quenched end.

Isothermal transformation studies are of considerable value in evaluating the heat-treating characteristics of titanium alloys. A time-temperature-transformation (TTT) chart for a 7% molybdenum alloy[13] is shown in Fig. 8. Tensile property trends indicated in the diagram are based on unpublished work being done at the Armour Research Foundation[12] and illustrate the useful information obtainable with this type of study. Typical structures obtained for various holding times at one temperature level are shown above the diagram. The results of tensile tests run on the iso-

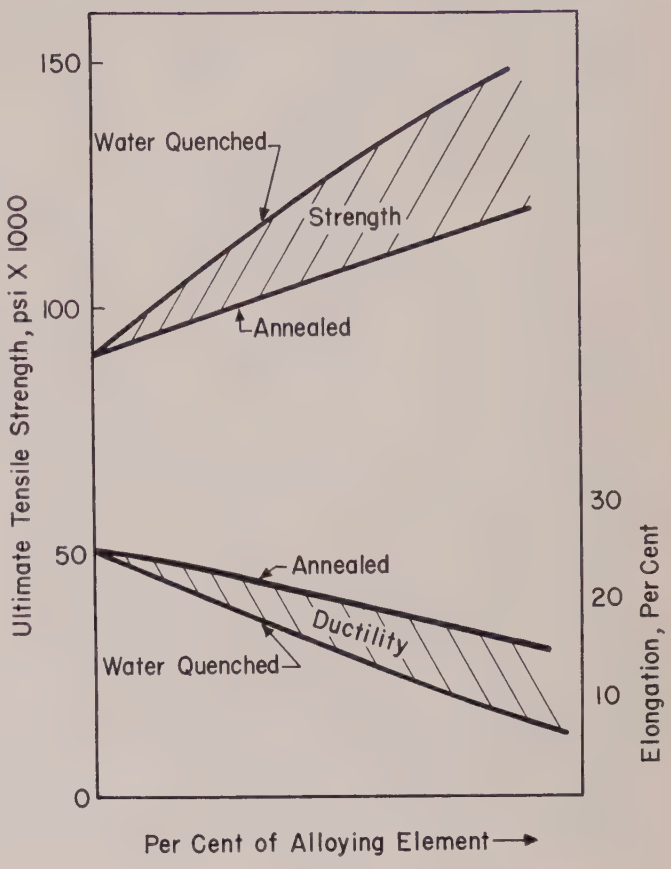


Fig. 6 – Tensile properties versus alloy content illustrating range of properties obtainable by heat-treatment for type 3 alloy system, titanium-aluminum

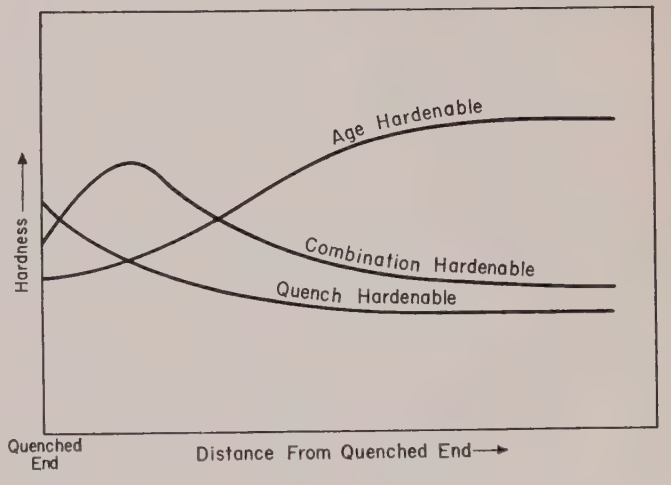


Fig. 7 – Schematic representation of types of hardenability curves obtainable in titanium alloy systems

[13] See *Journal of Metals*, Vol. 4, March, 1952, pp. 265-269: "Time-Temperature-Transformation Characteristics of Titanium-Molybdenum Alloys," by D. J. DeLazaro, M. Hansen, R. E. Riley, and W. Rostoker.

Table 1 – Tensile Properties of Isothermally Transformed 7% Molybdenum Titanium-Base Alloys

Heat-Treatment		Ultimate Tensile Strength, psi	Reduction in Area, %	Elongation, %
Temperature, C	Time, min			
550	5	178,000	0	0
	15	179,000	3	1.5
	60	180,000	2	1.5
600	15	176,000	2.5	0
	60	173,000	3	1.5
	120	166,000	5	3
650	5	165,000	5	1.5
	15	144,000	11	6
	60	140,000	12	6
700	5	131,000	8	5
	15	122,000	16	6
	60	127,000	24	12
750	5	122,000	5	3
	15	130,000	9	9
	60	109,000	18	19

thermally transformed 7% molybdenum alloy are shown in Table 1.

Alloys treated in the region between the two *C* curves – that is, between the start and completion of alpha precipitation – are generally quite brittle. Alloys treated below the nose of the *C* curve generally possess very high strengths, but are characteristically of very poor ductility. At all temperatures ductility improves with increasing isothermal annealing times. For this particular alloy the best combination of strength and ductility is obtained on heat-treating between 650 and 700 C. The best ductility and lowest strength values are obtained by heat-treating for an hour or longer at temperatures above 700 C and the beta/alpha plus beta transformation temperature indicated by the upper horizontal lines at 820 C. The lower horizontal line represents the M_s temperature below which alpha prime forms on quenching from the beta field.

Alloys for Use at Elevated Temperatures

The excellent strength-to-weight ratio obtainable with titanium and its alloys offers great promise for aircraft applications, particularly where somewhat elevated temperatures are encountered. Recognizing this fact, the Wright Air Development Center is sponsoring an investigation on "The Development of Titanium Alloys for Use at Elevated Temperatures" at the Armour Research Foundation. Of particular interest are alloys that can be substituted for aluminum alloys and stainless-steel parts in the gas turbine. This portion of the paper serves as a brief progress report of the accomplishments being made in this investigation.

The first step in the development of heat-resistant titanium alloys was the selection of a stable, high-strength, single-phase matrix. Subsequent work was then directed toward increasing the resistance to creep and improving the strength by the addition of other elements.

The alloys, no matter how simple or complex in composition, must be stable structurally at the temperature of service or at lower temperatures.

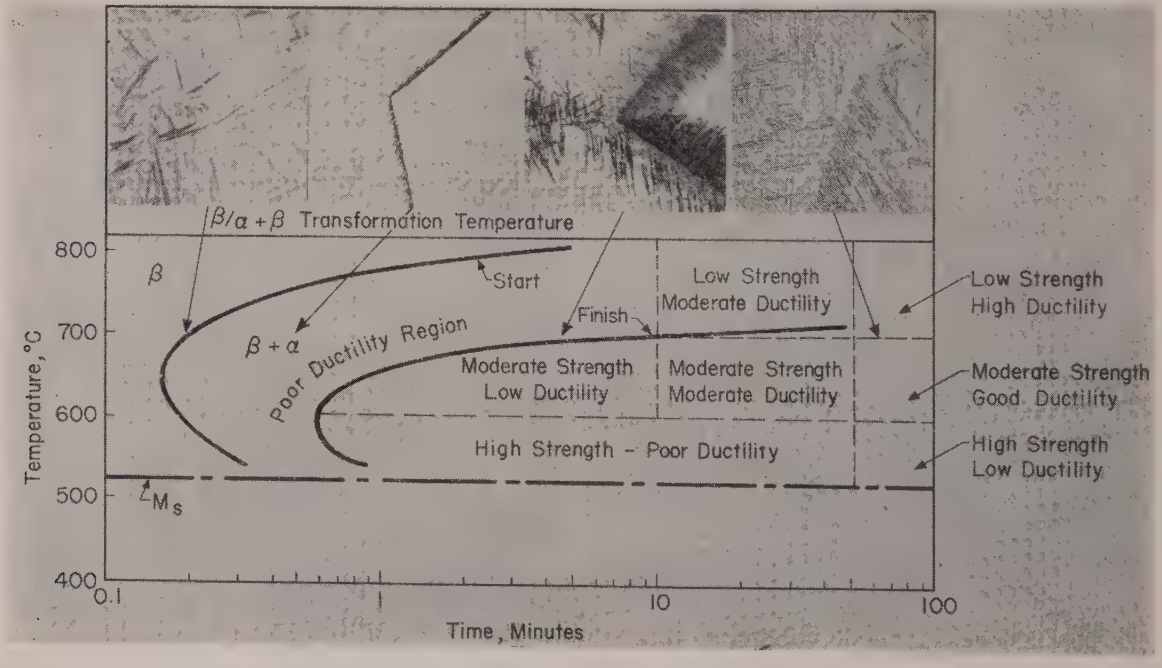


Fig. 8 – TTT diagram for 7% molybdenum alloy illustrating range of properties and microstructures obtained during course of isothermal heat-treatment

This requirement follows from the need for the structurally sensitive mechanical properties not to change in a way that would be detrimental to service life. Ideally, no precipitation or eutectoid decomposition should take place in the temperature range of the application. Such structural changes may result in a substantial variation of mechanical properties, and would cause volume changes that could be detrimental to the application. Heat-treatments should be applied with the primary object of achieving structural stability and a maximum resistance to creep.

Knowledge of many of the binary phase diagrams played an important part in the selection of alloys for the initial studies. Particular attention was focused on the titanium-aluminum system, as the phase diagram indicated that these alloys would be stable over a wide range of compositions due to the extensive solubility of aluminum in alpha titanium.

Although knowledge of the phase relationships considerably reduced the number of alloys required for the initial studies, the volume of alloys was still great; therefore a good screening technique was required. The hot hardness test was selected; for, as shown in Fig. 9, there is a good correlation between hardness and strength, but ductility can vary considerably for a particular hardness level. Although the data used in the preparation of Fig. 9 were based on room-temperature tests, the results of hot hardness and elevated-temperature tensile tests have indicated that the hot hardness and hot strengths fall on the same curve.

Fig. 10 shows the results of hot hardness tests carried out on binary alloys containing 5 and 6% additions of various elements. Based on these results and much additional information not shown here, the titanium-aluminum system was selected as the stable, high-strength base for the ternary alloy studies. As shown in Fig. 10, the 6% aluminum alloy showed the best stability at temperatures above 400 C. The strength-composition relationships for titanium-aluminum alloys at elevated temperatures are presented in Fig. 11. Although the strength increases with increasing aluminum content, compositions of 10% or more aluminum are extremely brittle at room temperature; therefore, 8% was selected as the maximum aluminum content for subsequent alloy development studies.

The selection of ternary alloys based on the titanium-aluminum system was considerably simplified by the availability of the binary phase diagram information. Systems where precipitation hardening could be expected were selected for this portion of the investigation. The hot hardness test was again used for the initial screening. An example of the results is shown in Fig. 12, which presents data for alloys containing 6% aluminum and the addition of from 1 to 6% of other elements. It will be noted that considerable improvement is

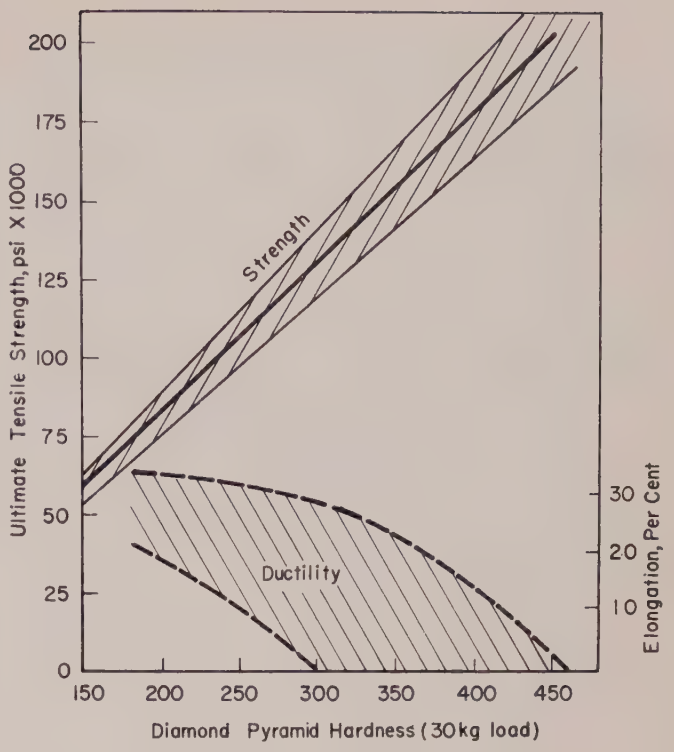


Fig. 9 – Tensile properties versus hardness correlation that generally applies to all titanium alloys, illustrating spread in tensile values obtainable at a given hardness value

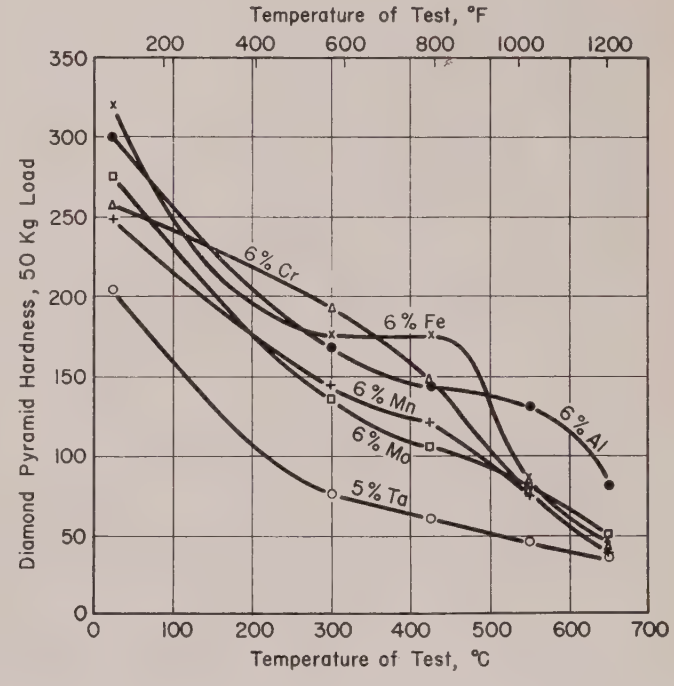


Fig. 10 – Hardness versus temperature curves illustrating comparative hot hardness of various binary titanium-base alloys

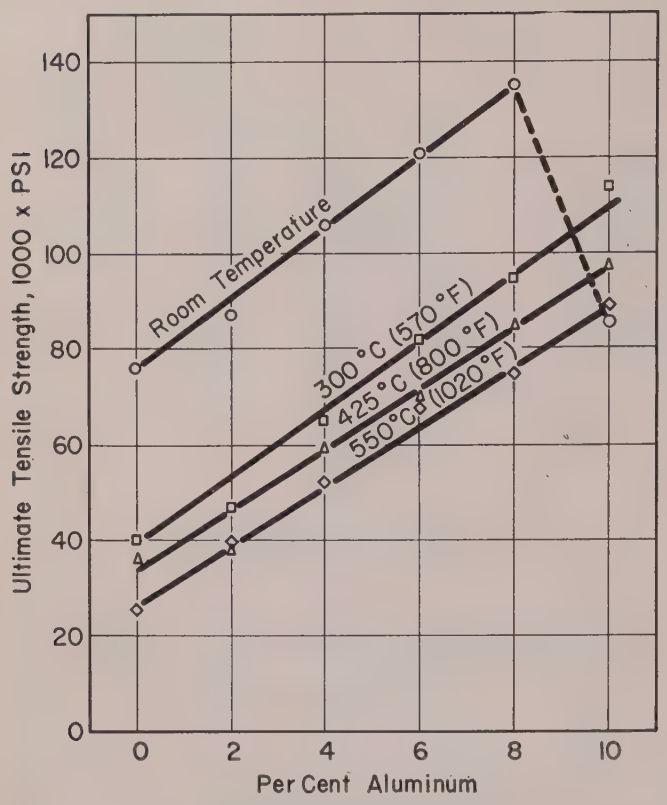


Fig. 11 – Tensile strength versus aluminum content at various temperature levels for titanium-aluminum binary alloys

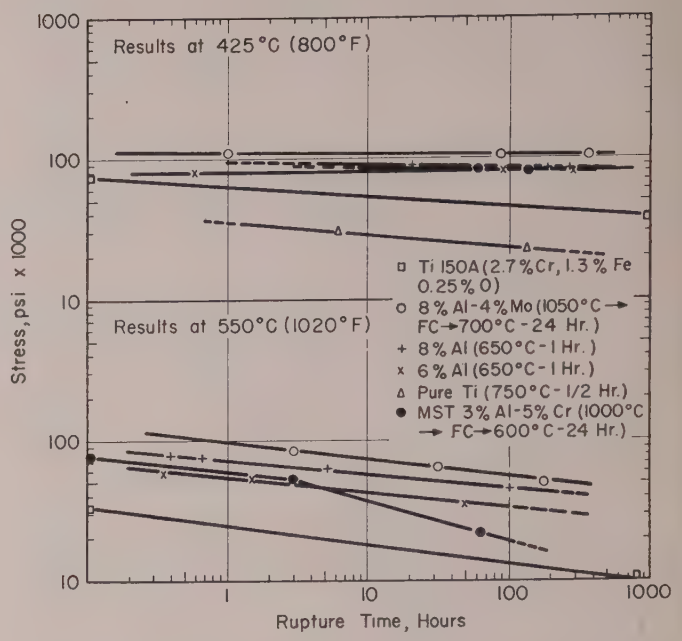


Fig. 13 – Creep rupture test results at 425C and 550C, showing comparison of properties between several experimental alloys, commercially pure titanium, and two commercial titanium-base alloys

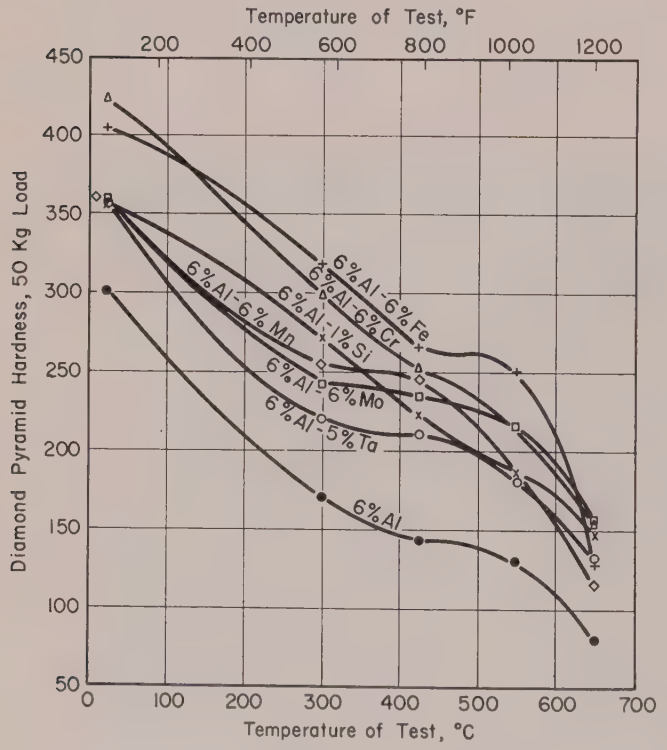


Fig. 12 – Hardness versus temperature curves illustrating hot hardness of various ternary alloys based on titanium-aluminum system, as compared to 6% aluminum binary alloy

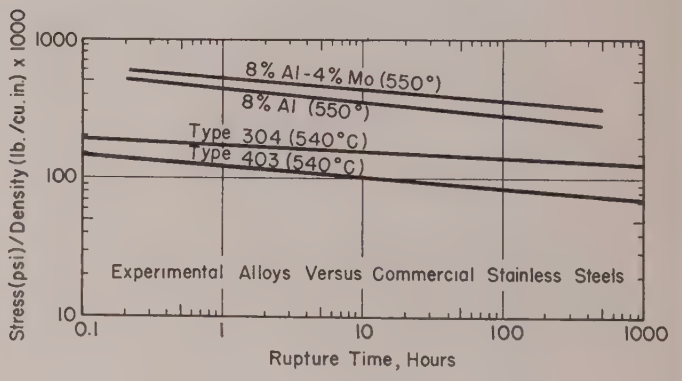


Fig. 14 – Creep rupture test results showing comparison on strength-weight basis between two experimental alloys and two commercial stainless steels tested at 550C and 540C, respectively

obtained in hot hardness by the addition of other elements to the 6% aluminum base. The results of the hot hardness studies of ternary alloys based on titanium-aluminum indicated that additions of iron, chromium, molybdenum, and silicon were of prime interest. Further studies were then carried on using creep rupture testing as a basis for evaluation of the more promising alloys.

With the exception of the titanium-aluminum alloys, many of the binary and ternary alloys studied had complex structures consisting of two or three phases at the temperature of testing. As noted previously, properties of the binary alloys of the titanium-chromium- and titanium-molybdenum-type systems can vary within wide limits of strength and ductility. Similarly, many of the Ti-Al-X alloys investigated were heat-treatable within wide limits of properties by the process of the precipitation of alpha or gamma from the supersaturated beta phase. When the known transformation characteristics of typical alloys were used, it was possible to predict heat-treatments that would render both the binary and the ternary alloys structurally stable at the temperature of testing. Only by the application of such stabilization treatments was it possible to obtain a fair comparison of the elevated-temperature properties of the various compositions tested.

Fig. 13 and Table 2 illustrate the creep rupture properties of some of the characteristic experimental alloys, as compared with the commercial alloys MST 3Al-5Cr[14] and Ti-150A (2.7% Cr, 1.3% Fe, 0.25% O).[15] At 425 C (800 F), the experimental alloys are as good, if not better than, MST 3Al-5 Cr. The curves lie considerably above that of commercially pure titanium. The results at 550 C (1020 F) show that the curves for the experimental alloys 8% aluminum-4% molybdenum, 8% aluminum, and 6% aluminum lie considerably above those for the two commercial alloys; and titanium does not even appear in the diagram.

Fig. 14 shows a comparison on a strength-weight basis between two experimental alloys and annealed types 304 and 403 stainless steels[16, 17] at 540-550 C.[16] These data certainly indicate that titanium-base alloys offer excellent properties for application to temperatures of at least 550 C (1020 F). At higher temperatures, oxidation becomes a problem with the present alloys, although aluminum has been shown to improve the oxidation resistance of commercial titanium. It is quite possible that, if titanium-base alloys have good strength properties at even higher temperatures, they could be applied, provided they are protectively coated or properly alloyed to prevent oxidation.

Table 2 – Creep Rupture Test Results for Alloys Tested at 425 C (800 F) and 550 C (1020 F)

Alloy Composition and Heat-Treatment	Testing Temperature, C	Tensile Stress, psi	Time to Rupture, hr	Total Elongation, %	Stress to Rupture, 100 Hr	Stress to Rupture, 500 Hr
6% Al (650 C – 1 hr)	425	80,000	0.58	13	79,500	78,000
		79,500	91	22		
		79,000	288	20		
	550	58,000	0.25	23	32,000	26,000
		53,000	1.5	30		
		35,000	49	36		
8% Al (650 C – 1 hr)	425	87,500	21	14	86,500	84,000
		86,000	183	6		
		83,000	270	17		
	550	77,500	0.40	19	45,000	38,000
		75,000	0.67	8		
		62,500	5.3	9		
		45,000	102	9		
8% Al – 4% Mo (1050 C – 1 hr – F.C. 700 C – 24 hr)	425	110,000	4.7	14	105,500	102,000
		108,500	48	5		
		107,000	83	5		
		106,000	87	14		
		103,000	360	5		
	550	84,800	3.5	13	56,000	45,500
		65,000	33	9		
		50,000	181	5		
3% Al – 5% Cr (Mallory-Sharon Alloy) (1000 C – 1 hr – F.C. 600 C – 24 hr)	425	81,500	60	11	80,500	77,000
		80,000	137	8		
	550	80,000	0.08	...	19,000	12,000
		52,800	3.0	36		
		22,000	64	14		
304 (annealed)	540	...			36,500	33,000
403 (annealed)	540	...			32,000	31,000

In conclusion, it should be stated that titanium metallurgy is in the ideal position of developing concurrently or even in advance of industrial practice. The important physical metallurgical information on phase diagrams and transformation kinetics is of considerable aid to alloy development and should help to bring about the development of optimum alloys for various applications in the shortest possible time. Use of this approach has resulted in the development of titanium-base alloys having excellent properties at temperatures up to 550 C (1020 F). Some of these alloys show better creep rupture properties at this temperature than types 304 and 403 stainless steel.

Acknowledgments

The authors are indebted to the Flight Research Laboratory and the Materials Laboratory, Wright Air Development Center, which sponsored much of the reported work, for permission to publish this paper. The help of many associates at Armour Research Foundation is also appreciated; thanks are particularly due to W. Carew and F. Crossley, who performed the studies on elevated-temperature properties; and C. A. Johnson, who supervised the metallographic work.

[14] Produced by Mallory-Sharon Corp. Tested at Armor Research Foundation.

[15] See "Handbook on Titanium Metal." Pub. by Titanium Metals Corp. of America, New York, 1952.

[16] See pp. 34 and 46 of "Working Data for Carpenter Stainless Steels." Pub. by Carpenter Steel Co., Reading, Pa., 1946.

[17] See pp. 54 and 160 of "Steels for Elevated-Temperature Service." Pub. by United States Steel Co., Chicago.

Instrumentation

THIS paper – which received the SAE Manly Memorial Medal for 1952 – presents a survey of special-purpose instrumentation required for development testing of aircraft gas turbine engines and their components. They are divided into:

1. Instruments to investigate the mechanical operation of the engine.

2. Instruments to measure the quantity and state of the working fluids – fuel, air, and combustion gases – as they pass through the engine.

R. E. GORTON is development engineer of the Instrument Group, Pratt & Whitney Aircraft, Division of United Aircraft Corp. He received his B.S. and M.S. degrees in aeronautical engineering from Rensselaer Polytechnic Institute in 1937 and 1938, respectively, and joined Pratt & Whitney upon graduation.

B. E. MILLER is assistant project engineer of the Airflow Measurement Section of the Instrument Group, Pratt & Whitney Aircraft, Division of United Aircraft Corp. He received his B.S. degree in mechanical engineering from Michigan College of Mining and Technology, and joined Pratt & Whitney upon graduation.

THE array of instrumentation used for gas turbine engine development is to the development engineer what a pair of spectacles is to the nearsighted man with severe astigmatism. Both are expensive, readily subject to damage, and troublesome to use and maintain. However, without extensive instrumentation the development engineer is blind. He can neither see where he is going nor find small chunks of thrust or horsepower that are frequently lost.

Twenty to twenty-five years ago, early in the history of aircooled radial engine development, typical test stand instrumentation was not expensive, as shown in Fig. 1. A few pressures and temperatures, engine speed and torque, oil comsumption, and fuel flow were all the variables measured. About the only special-purpose portable instrumentation that could be brought to the aid of the few instruments shown was the ear of the experienced mechanic, who could pass upon the "smoothness" of the engine.

As aircraft engines have become more highly developed, more instrumentation has been necessary to provide the necessary design information. In piston engine development most of the specialized instrumentation has been applied to the mechanical problems of vibration, fatigue, or lubrication. During the last decade special airflow instrumentation has been devised and applied to supercharger development, permitting significant improvement in piston engine altitude performance. Practically all of the instrumentation used during piston engine development has been also applied to gas turbine engines. Fig. 2 illustrates some of the portable instrumentation used during initial running of one new type of gas turbine engine. In addition, the shift in emphasis from mechanical problems to the aerodynamic and combustion studies necessary to improve turbine engine performance has introduced new requirements for instrumentation.

Instrumentation for Mechanical Problems

The discussion of turbine engine instrumentation may conveniently be subdivided into two parts: instrumentation for mechanical problems, and instrumentation of the working fluids, that is, air and fuel. On the mechanical side, one of the first questions to be answered is, "How much power have we?" For a propeller-turbine engine the shaft torque output can be measured by a torquemeter built into the engine reduction gear, by an external torquemeter, or by a cradled dynamometer. The problem is different from that existing for piston engines only in that the power to be handled is greater. This is particularly true when testing a turbine or a compressor separately, as required in development of each component. A 5000-hp propeller-turbine engine may develop 15,000 hp in its turbine, and put 10,000 into its compressor. An example of the apparatus required for such testing is the 24,000-hp compressor drive stand at the Willgoos Turbine Laboratory, shown in Fig. 3. The motor and speed-increasing gearbox are cradled as a unit, weighing about 80 tons, supported on oil films in four spherical hydrostatic bearings. With a maximum rating of 16,000 ft-lb torque, there is so little friction in the suspension that a variation of less than 30 ft-lb can be detected.

Most of these torque-measuring devices convert torque to a force at the end of a lever arm of some known length. With a turbojet engine, on the other hand, the output thrust force is the quantity to be measured. To measure this thrust the bed to which the engine is attached must have axial freedom, restrained by a force-measuring device. Test

For Aircraft Gas Turbine Development

R. E. Gorton and B. E. Miller,
Pratt & Whitney Aircraft, Division of United Aircraft Corp.

This paper was presented at the SAE Annual Meeting, Detroit, Jan. 17, 1952.

beds to accomplish this may be suspended by means of a parallelogram linkage, by means of thin flexible metal straps or plates, or they may slide on axial shafts that are rotated to reduce friction. For measuring either thrust or torque, the end instrument must measure force, and it usually must indicate at some remote point. A wide variety of instruments are available for this purpose. Most elementary is a mechanical weighing scale. This may be located at the position of the force, or it may be remotely located, with the force transmitted by rods, bell-cranks, and knife edges. The turbine test stands of Rolls-Royce, Ltd., in England use the all-mechanical, remote, force-measuring system, and achieve greater reliability than do any of the more complex systems that have been tried. The indication of a mechanical weighing scale is also frequently transmitted electrically to a remote point by a selsyn link.

The force resulting from torque or thrust may also be directly measured by nonmechanical means. Two common methods are the strain-gage load cell and the hydraulic load cell. In the first, a stiff member carrying wire strain gages is deflected elastically by the applied load and the output of the gages is read remotely by suitable electrical instruments. In the hydraulic load cell the load acts against a piston or very flexible diaphragm terminating one end of a liquid-filled system. At the remote end of a tube is a precision pressure gage, usually of the Bourdon type. Pressure generated in the closed system by mechanical load on the diaphragm is indicated on the gage, calibrated in force or torque units.

A hydraulic or pneumatic servosystem may also be used to generate a pressure proportional to the load. The applied load is opposed by pressure in a chamber closed by a piston or diaphragm. A pressure storage reservoir is maintained at some pressure above that corresponding to maximum load rating. A two-way valve connects the load-opposing chamber to either the pressure reservoir or to a drain. This valve is mechanically linked to the piston or diaphragm in such a manner that motion of the piston under load operates the valve to increase or decrease the chamber pressure to balance the load, returning the valve nearly to its neutral position. The chamber pressure is remotely indicated on a precision pressure gage. Such servosystems must be carefully designed and maintained to avoid hysteresis or instability.

Having instrumented a turbojet test stand to measure thrust, the instrumentation engineer can expect sooner or later a complaint that the thrust measurement is incorrect; that nothing can be found wrong with a particular engine to explain a slightly low measured output. Perhaps the thrust measurement does need checking, but more likely the engine speed measurement would bear investigation. The thrust output of a turbojet engine at maximum rated speed is very sensitive to small speed changes. For several types of engines for which experimental data were plotted on logarithmic coordinates, the slope of the thrust versus speed curve at maximum rated speed showed the thrust to increase with the sixth to eighth power of the speed. Stated differently, for one type of engine, a 1% change in speed caused a 7% change in thrust. Military specifications call for thrust measurement accurate to 1%. To repeat thrust measurements to this accuracy the speed must be set to one-seventh of one per cent. Let us say that speed-measuring instrumentation is desired accurate to 0.1%, and see what complications arise.

The general class of indicating tachometers that, by means of magnetic drag, centrifugal force, or

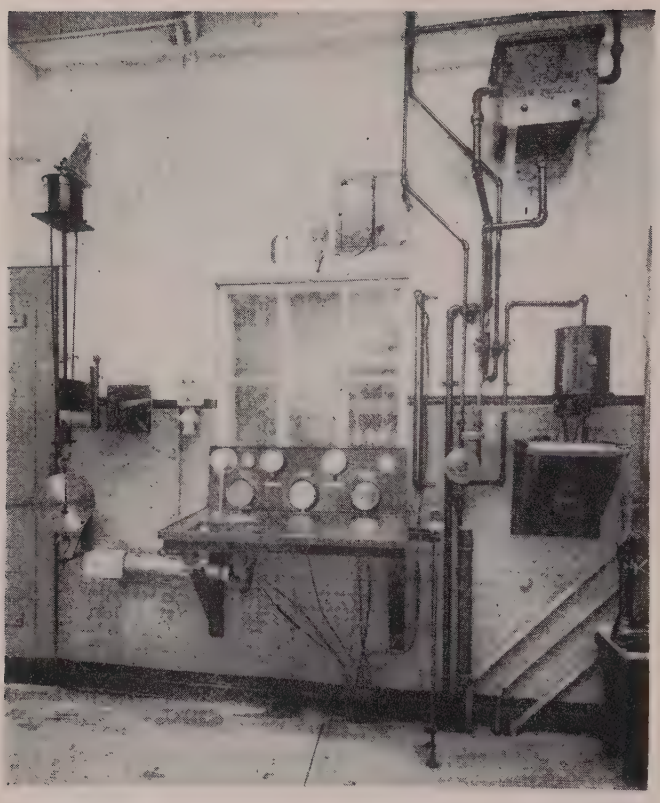


Fig. 1 – Aircraft-engine test instrumentation used prior to 1930

other physical phenomena convert speed directly to a pointer deflection can be dismissed as not providing better than 0.5% accuracy.

Next, we may consider tachometers that measure speed by comparison with some reference speed or frequency. This technique can be made to give an accuracy of speed measurement comparable to the accuracy of the reference frequency. The most obvious frequency for such a purpose is that of the 60-cps a-c power systems used throughout most of the nation. One of the most common tachometers uses an electric clock to trigger a revolution counter. Engine revolutions are counted for a fixed time interval, say one minute, giving an average speed over that interval. However, before the accuracy of the result can be stated the accuracy of the 60-cps power source must be known for that same interval. Frequency deviations vary with the size and operating practices of the utility or power plant providing the power. Few, if any, stay within 0.1% of 60-cps frequency all the time. Most large power plants maintain a frequency recorder. A study of the deviations recorded over several weeks should enable the user to decide whether or not his power frequency is adequate for reference. Another possibility is for the user to maintain a frequency recorder, so that

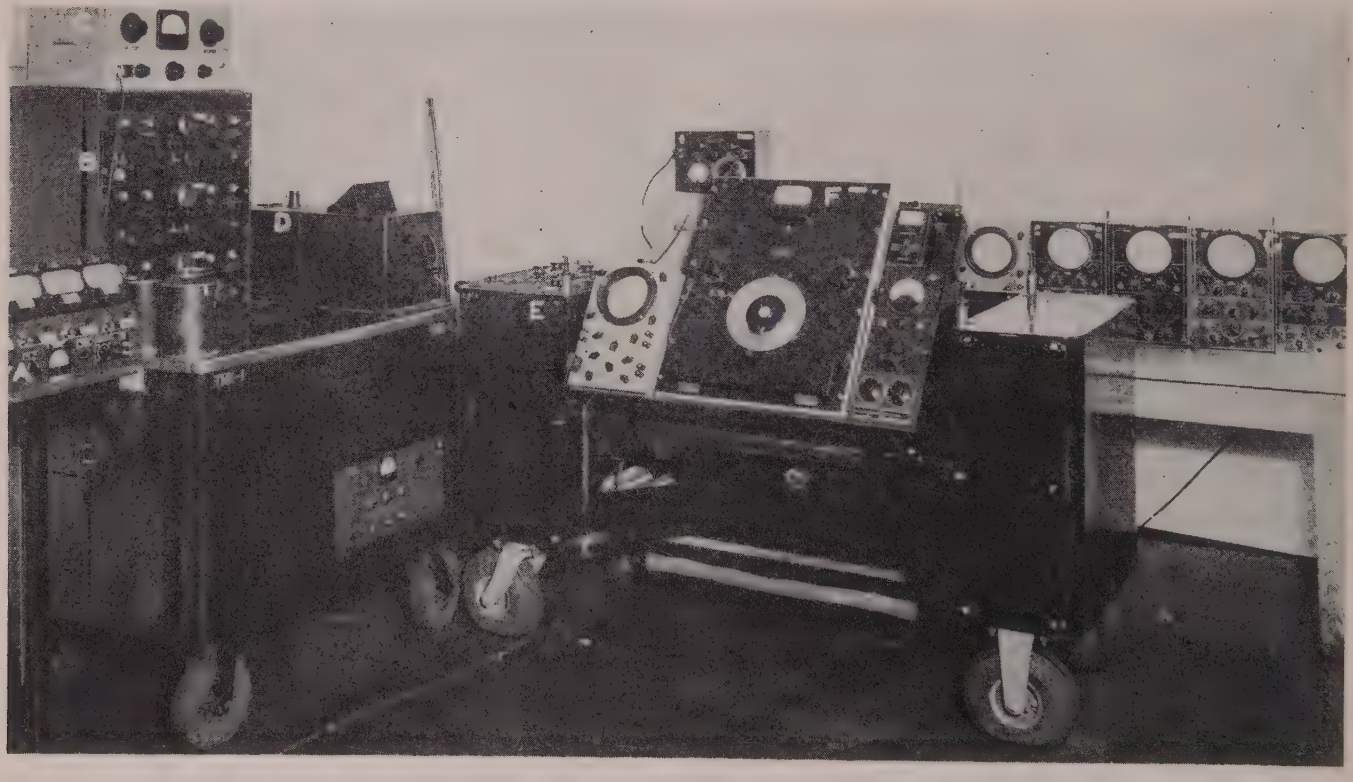


Fig. 2 – Some of the special-purpose instrumentation used on one turboject engine test

A – Four-channel vibration meter
B – Amplifiers for oscillograph
C – Frequency meter to indicate rotor speed
D – Twelve-channel oscillograph for recording compressor blade vibration
E – Strain-gage bridge and switch panel
F – Wave analyzer
G – Cathode ray oscilloscopes for visual monitoring of blade vibration

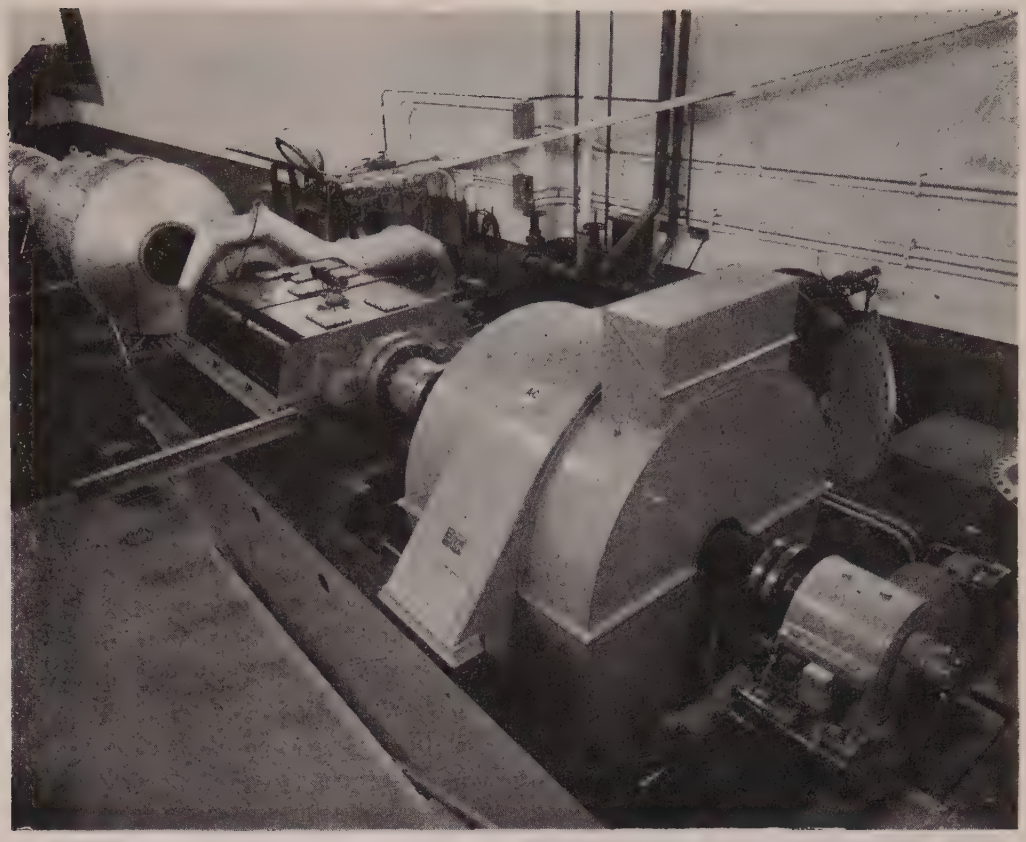


Fig. 3 – Motor and gearbox cradled for torque measurement – one of the larger instruments

speed measurements can be discarded or corrected during intervals of inaccurate frequency.

In many cases the power frequency stability is so poor that its use as a speed-measuring reference is out of the question. It is then necessary to resort to a tuning fork or crystal-controlled oscillator to generate a stable frequency. Accuracy of the reference frequency to 0.1% is easily achieved by this means. The problem then becomes one of devising a comparison instrument that will operate from the low power output of such oscillators or of amplifying the output. One solution is to generate a 60-cps reference frequency, then, by power amplification and a special distribution system, obtain a source of accurate 60-cps power throughout the engine test area, adequate for tachometer drive. Another simple means of checking discrete engine speeds is to drive a rotating disc in the instrument in synchronism with the engine by a self-synchronous repeater. The reference frequency source is then made to flash a lamp to illuminate the disc stroboscopically. At fixed ratios of engine speed to reference frequency the disc appears stationary, at which instant the engine speed is determined within the accuracy of the reference frequency. Another modification of this arrangement (as shown in Fig. 4) has been used for setting take-off speed in fighter aircraft, where space is limited. Instead of a stroboscopic disc, a synchroscope such as is employed for synchronizing the two engines of a twin-engined airplane is used. One winding is energized from the a-c output of the tachometer generator of the engine, which also drives the pilot's indicating tachometer. The other synchroscope winding is energized through a small amplifier from a tuning fork. This fork is tuned at manufacture to the frequency corresponding to the tachometer generator output at take-off rpm. When the tachometer generator frequency equals the tuning fork frequency, the synchroscope indicator will become stationary and take-off speed will be set within the accuracy of the tuning fork.

These methods have certain disadvantages. The revolution-counter, electric-clock combination gives an average speed over an interval of the order of one minute – a longer time than is acceptable in some cases. The stroboscopic or synchroscope arrangements usually permit accurately setting only certain discrete speeds. By using electronic-counter techniques these disadvantages can be overcome. An electronic-counter tachometer does basically the same thing as the electric-clock, revolution-counter instrument. A slotted disc driven by the engine produces many impulses per revolution by photoelectric or electromagnetic action. This permits the electronic counter to count fractions of revolutions. An adapter containing the slotted disc and an electromagnetic pickup may be constructed, as shown in Fig. 5, to fit between the aircraft tachometer generator and its pad. The time base is generated by a crystal oscillator at radio frequency. A second electronic counter counts a preset number of cycles of this oscillator signal to determine the time interval

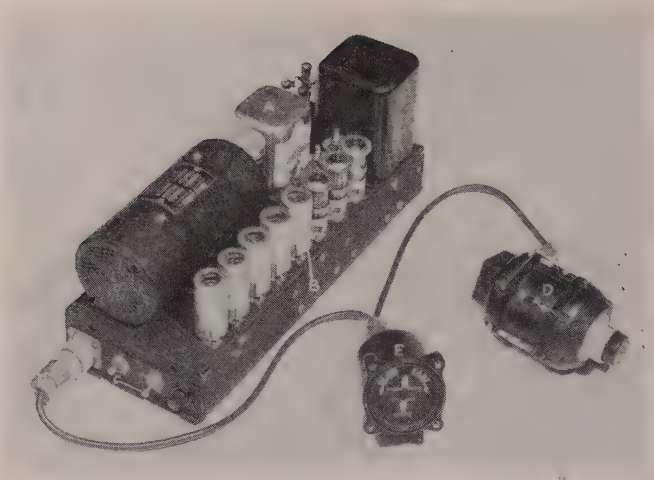


Fig. 4 – Accurate engine speed setting indicator for flight use
A – Precision tuning fork – 824.94 cps
B – Divider and amplifier section
C – Frequency selector switch – 91.66 or 45.83 cps
D – Tachometer generator
E – Pilot's indicator – pointer becomes stationary at take-off rpm

during which engine revolutions are counted. This time interval may be as short as one second. After a few seconds for reading the count, the operation is automatically repeated. By thus speeding up the counter operation, readings may be obtained every few seconds, with an accuracy of better than 0.05%. This type of instrument has the disadvantages of expense and complication, but it will provide speed-measurement ability not otherwise obtainable.

The instruments mentioned for measuring torque, thrust, or speed can be made adequate for indication or recording of steady-state engine conditions. New requirements arise as soon as these variables change rapidly with time, as during an engine acceleration or during unsteady operation with unsatisfactory stability. When, under such conditions, torque or thrust is required as a function of time, the natural frequency of the measuring system and the rate of response of the recording instrument must be considered. For a smooth acceleration of large aircraft turbine engines, the rate of change of engine variables is sufficiently slow so that good results are obtained from photographs of the steady-state indicating instruments taken at a rate of two or more pictures per second. This technique is inaccurate in test stands with unusually long pressure lines to indicating gages or when applied to turbine engines having unusually high acceleration rates.

Fig. 5 – Tachometer adapter incorporating electromagnetic pulse generator
A – Adapter housing
B – Electromagnetic pickup
C – Tooth wheel
D – Standard tachometer generator

When more rapid response is required, say up to 50 cps, various direct-writing oscillographs, recording on strip charts by ink, hot wire, or other means, are available for recording of voltage signals. Transducers whose frequency response is adequate for the desired investigation are used to convert torque, thrust, or other variables into electrical form. A typical four-channel recorder of this type in use for recording of engine transient response is shown in Fig. 6. With transducers of higher frequency response and with galvanometers recording optically on photosensitive film or paper, time response in the range of one millisecond can be achieved. These methods are adequate for study of most cases of fluctuating output, due to instability of combustion, fuel system, or control.

Engine speed versus time is of interest during acceleration tests or for investigation of control instability. For smooth accelerations, photographic recording of an indicating tachometer with two or more pictures per second is frequently used to get the general shape of the acceleration curve. The tachometer can be expected to lag by 1 to 2% during the most rapid part of the acceleration, and possibly to overshoot by a similar amount. Photographic recording of the indication of an electronic-counter tachometer would give more accurate readings, eliminating the lag and overshoot; but any stock instrument would not give readings more frequently than at one-second intervals – an inadequate rate. A d-c voltage proportional to engine speed may be generated with sufficient accuracy and recorded upon a recording voltmeter or oscillograph. As long as the recording instrument has a time response of one-tenth of a second or better, the shape of the voltage-time curve should be adequately reproduced. Such instruments are likely to have a scale length so limited that reading accuracy is no better than 1%. For acceleration over a limited speed range or for study of speed fluctuation about a steady average speed, the scale may be expanded by bucking out the average d-c tachometer voltage by an equal and opposite external stable source of voltage and using the full scale of the recording instrument to show the variations. This technique is also commonly used in engine control development testing for expanding the scale for other variables than speed.

There is occasionally a requirement for a speed versus time record over a wide speed range, with better than 1% accuracy. Such a record can be obtained from a multichannel oscillograph record taken at a film speed of five or more feet per second. One channel is used to record a fork or crystal-controlled constant-reference frequency, which may be from 100 to 1000 cps. The other channel simultaneously records an impulse generated by the engine one or more times per revolution. At the expense of considerable time and eye-strain, this record can be analyzed to give the desired data. The electronic-counter tachometer could be made to do this same job by increasing its sampling rate and photographically recording its indications.

If the "one simple rotating member" that is supposed to make the turbine engine so attractive compared to the reciprocating engine always behaved itself, there would be little need for any further instrumentation of the mechanism. But, particularly in the larger axial-flow compressors, the rotor is not a single, simple, structural member, but is an assembly of various blades, discs, spacers, hubs, shafts, and bolts. This assembly of parts may cause many kinds of mechanical trouble before being developed to a satisfactory state.

One type of rotor trouble is runout, or bending of the rotor axis away from the axis of its supporting bearings. A stationary pickup device located with a small clearance radially outward from the blade tips can be made to give an indication of the variation in that air gap caused once per revolution by rotor runout. Most blade materials have a magnetic permeability somewhat greater than that of air, so an electromagnetic sensing device can be used. Fig. 7 shows a small, U-shaped electromagnet enclosed in a tubular holder, which may be adjusted by a micrometer mount to the desired clearance from a row of blade tips. As each blade passes the magnet, a positive, then a negative, pulse is generated in the output. If the rotor has runout, then the envelope of the puise amplitudes is modulated, becoming larger at the point of minimum clearance, smaller when the rotor has rotated 180 deg to the point of maximum clearance. By applying this modulated wave to the electronic modulation meter also shown in Fig. 7, a meter reading somewhat proportional to runout is obtained. This combination of pickup and modulation meter is capable of determining small runout to about 2% accuracy under carefully controlled conditions. Under actual engine test conditions, an accuracy of from 10 to 25% is estimated. This is adequate to get a measure of rotor bending as affected by changes in design or operating conditions.

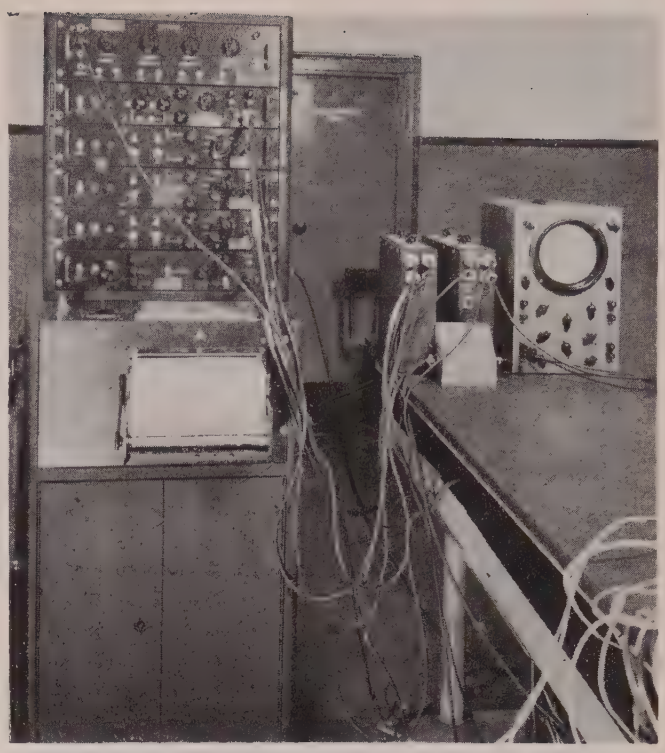


Fig. 6 – Direct-writing oscillograph for recording engine transient response
A – Four-channel recorder
B – Interchangeable d-c or carrier-type amplifiers
C – Voltage bucking circuits to permit scale expansion

A capacitance pickup has been described[1] for measuring blade tip clearances while the rotor is slowly turned. This device could very likely be developed to work with a modulation meter as a runout pickup for application to completely nonmagnetic blades.

If there is a fear that blades may rub, due to runout, warping of the engine cases, or rotor growth with heat or centrifugal force, a compressor may be fitted with electrical contacts to trigger a "rubbing indicator." The surface against

Fig. 7 – Electromagnetic pickup and modulation meter for indicating rotor runout
Insert: Electromagnetic pickup to be inserted through engine case opposite blade tips

[1] See Royal Aircraft Establishment Report EL-1474 (1950), "Measurement of Blade Tip Clearances in Aircraft Turbines by Capacitance Method," by I. A. Mossup and F. D. Gill.

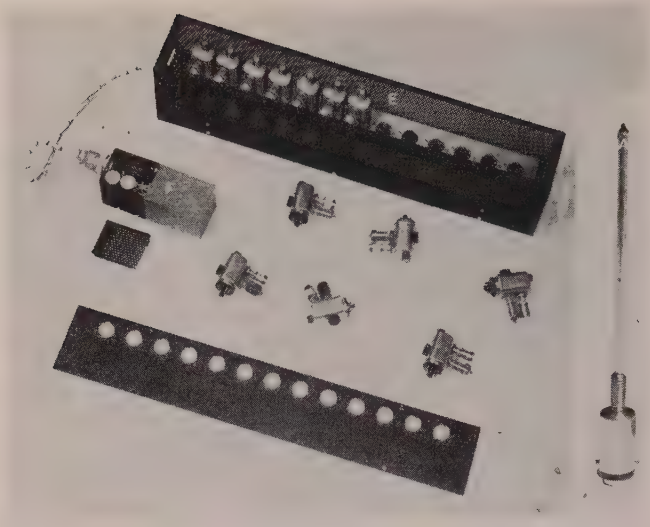


Fig. 8 – Contact and warning circuit for detecting blade rubbing
Insert: Enlarged view of probe having insulated contact A
B – Plug-in relay assembly
C – Trigger tube
D – Sensitive relay
E – Rack for 13 plug-in units
F – Power supply

which rubbing is feared is fitted with several insulated electrodes of fine wire, which project a few thousandths of an inch from the surface toward the blades. A blade touching any one of the electrodes makes a contact that is used to flash a warning light. Since the actual time of contact may be extremely short, a means is provided for holding the light on for several seconds. The electrical circuit through the rotor bearings may be of quite high resistance unless the rotor is especially grounded. The short duration and high resistance of the rubbing signal require a high-speed, sensitive relay circuit, accomplished by using a trigger tube to operate a relay. The resistance of contact at which the tube will fire and the time delay before it resets, extinguishing the warning light, are readily adjustable. In test rigs with excessive oil or water, the electrodes may become partially shorted, giving false signals. In such cases the electrode is coated with an insulating varnish to prevent shorting, or the rotor may be grounded by a slip ring and brush, permitting a much less sensitive setting of the trigger tube. Fig. 8 shows a 13-channel rubbing indicator as constructed to warn of rubbing in any one of the 13 stages of a compressor.

In some single-stage compressor test rigs, the rotor is mounted on a shaft protruding from a gearbox, while the housing and stator vanes are mounted on a separate pedestal to the bed plate. If the compressor is run with small blade tip clearances, the differential thermal expansion between the gearbox pedestal and the housing pedestal is sufficient to cause blade rubbing. Continued operation is made possible by steam heating the cooler of the two pedestals, with the amount of heating controlled by reference to rubbing indicators at the blade tips.

Closely allied to the problem of rotor runout is the question of vibration. The vibration of any new and untried test rig or engine involving a high-speed rotor must be monitored to assure safe operation. The pickups themselves are, in most cases, seismic, electromagnetic generators operating above their natural frequency. These are similar in principle and frequently similar in construction to the first widely used electromagnetic pickups, described in 1936.[2] Descriptive material on vibration and motion measurements may also be found in other texts.[3]

There are a few problems peculiar to turbine engines that are met in applying vibration pickups. One problem arises when the designer or analyst wants to know the vibration of a rotor-supporting bearing in an axial-flow engine. The structure of struts between the bearing and the outer case may be so flexible that motion on the outside of the engine cannot reliably be assumed to be identical to bearing motion. The pickup must be put inside, on the bearing housing. If the bearing in question is at the discharge end of a compressor or on the turbine, the temperature around it may be that of "cooling" air obtained from the compressor discharge. No pickups are at present known to be available that will withstand the temperatures over 500 F reached by this "cooling" air. The pickups may be enclosed in a heat-insulated metal box, with an external source of cool air supplied to the inside of the box. In order to be able to use the ordinary shop-compressed air line as a source of this cool air, a discharge line from the pickup box must be led outside the engine, as compressor pressures around the pickup will probably exceed the air line pressure. Getting a pickup, insulated box, air supply and return lines, and electrical leads into the restricted internal spaces of an engine is not a simple task. After it has been accomplished, the cooling effect that can be obtained through the small air lines required may be found marginal, so that pickup life is still short. Vibration pickups that could withstand temperatures above the 200 to 250 F limits of present pickups would be of great advantage in such applications.

The output of velocity-type pickups may be observed on a cathode ray oscilloscope, recorded on an oscillograph, studied with a wave analyzer to determine the frequency components, or applied to the input of an indicating vibration meter. The first three instruments are commonly used during initial running of a rig or engine, while its normal

[2] See *Journal of Aeronautical Sciences*, Vol. 4, May, 1937, pp. 281-285: "M.I.T.-Sperry Apparatus for Measuring Vibration," by C. S. Draper, G. P. Bentley, and H. H. Willis.

[3] See "Handbook of Experimental Stress Analysis," by M. Hetenyi. Pub. by Wiley, New York, 1950.

vibration performance is being determined. For later running the indicating meter is used to monitor the amplitude to see that no change occurs. Small meters with integrating amplifiers are available, the scale reading amplitude in thousandths of an inch. Such meters have been built to P&WA specifications as shown in Fig. 9. They incorporate the following features:

1. Small size. Four meters can be installed in a panel space less than one foot square.
2. Self-contained electronically regulated power supplies.
3. Plug-in high-pass frequency filters. For any given engine or rig, filters can be chosen to pass rotor-unbalance frequency and to eliminate lower frequency shaking of the rig. Two filters of different cut-off frequency can be inserted at one time, while a selector switch permits choice of either of those filters, or no filter.
4. Ten-turn potentiometer for accurately setting the calibration constant of any vibration pickup.
5. Simple built-in calibration circuits for setting amplifier sensitivity.
6. Choice by selector switch of two different full-scale meter ranges.
7. Warning light and relay contacts that are actuated by vibration exceeding a preset limit.

For some applications the 6- to 11-oz weight of velocity-type pickups is excessive. Such is the case when local motion of a sheet-metal duct or motion of some relatively light appendage to the engine is desired. Very small and light accelerometers, employing piezo-electric crystals as the sensitive element, have recently become available. Accelerometers with weights from one-quarter to two and one-half ounces may be obtained, with usable frequency ranges as high as 50,000 cps. Accelerometers are also required for such tests as that of dropping an engine shipping container to determine its effectiveness in protecting the engine.[4]

While velocity-type vibration pickups or accelerometers are adequate for motion measurements of a vibrational nature, there are other motions whose frequency or acceleration is so low that these pickups are not applicable. The linear differential transformer finds considerable use for detection of such motions. This device, operated with an a-c carrier amplifier, is sensitive to relative motion between a coil assembly and a magnetic core that slides inside the coils.[5] One ten-thousandth of an inch may easily be measured. Typical applications for the differential transformer include: measurement of the motion of levers, pilot valves, and diaphragms in engine controls; measurement of distortion of engine cases under load or due to thermal expansion; and measurement of motion between parts of test rigs that are suspected of going out of alignment.

No discussion of instrumentation for vibration in turbine engines would be complete without mention of wire strain gages. The control of turbine and compressor blade vibration is one of the major requirements for engine reliability. Techniques of strain measurement applicable to blade vibration studies have been described.[6, 7, 8] Little more need be said here about gage installations. With proper care in choosing the gages to suit the temperature range expected, the space available, and the data desired, proper care in locating and installing them,

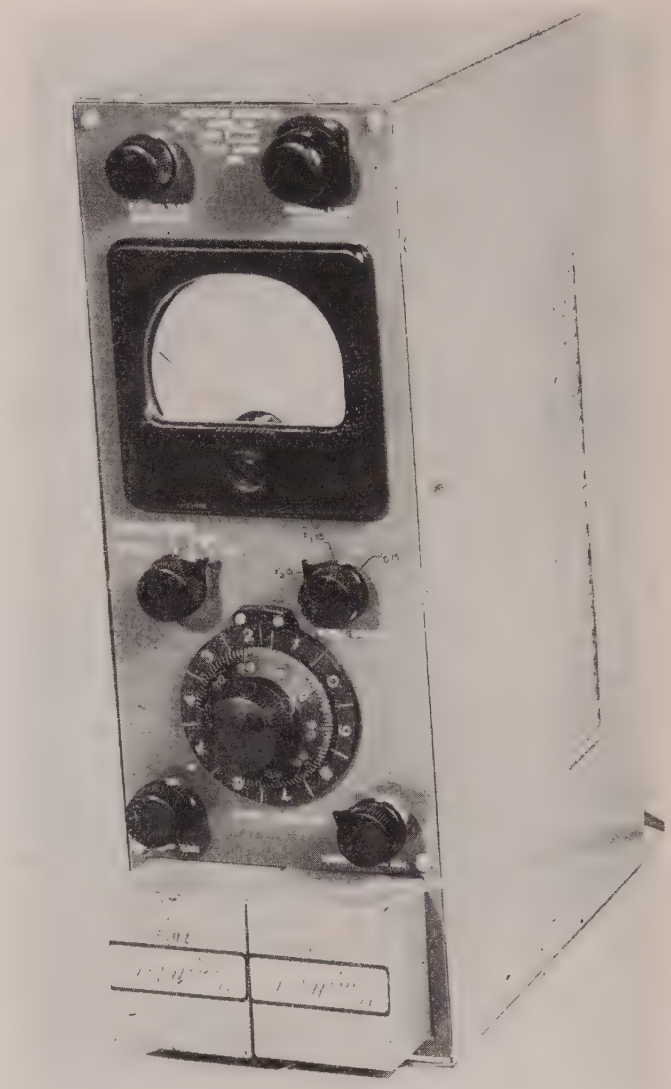


Fig. 9 – Vibration meter with plug-in filters
A – Filters

[4] See U. S. Navy, David Taylor Model Basin Report 720, February, 1951, "Evaluation of Selected Shock Instruments," by T. A. Perls and H. L. Rich.

[5] See Proceedings of Society for Experimental Stress Analysis, Vol. 4, No. 2, 1947, pp. 79-88: "Linear Variable Differential Transformer," by H. Schaevitz.

[6] See SAE *Quarterly* Transactions, Vol. 3, October, 1949, pp. 540-556: "Strain Measurements on Rotating Parts," by R. E. Gorton and R. W. Pratt.

[7] See Proceedings of Society for Experimental Stress Analysis, Vol. 8, No. 2, 1951, pp. 209-228: "Advances in High Temperature Strain Gages and Their Application to Measurement of Vibratory Stresses in Hollow Turbine Blades during Engine Operation," by R. H. Kemp, W. C. Morgan, and S. S. Manson.

[8] See Proceedings of Society for Experimental Stress Analysis, Vol. 9, No. 1, 1952, pp. 163-176: "Development and Use of High-Temperature Strain Gages," by R. E. Gorton.

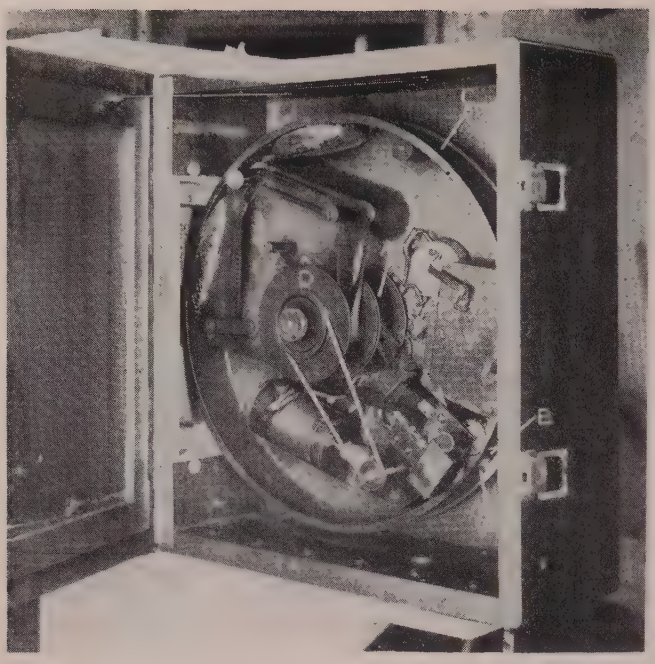


Fig. 10 – Drum with internal film supply for high-speed cathode ray oscillography

A – Film supply spool
B – Slot through which film enters and leaves drum
C – Film on outside diameter of drum in position for exposure
D – Film take-up spool
E – Film advance drive motor
F – Film advance sprocket

and suitable slip rings, there is little trouble in getting the desired signals from the gages. One thing still lacking at present is a source of commercially available high-temperature gages adequate for turbine blade application.

Having obtained electrical signals from strain gages, we cannot consider the instrumentation problem completed. The recording of some blade vibration signals requires special apparatus, due to the extremely high frequency of the vibrations present. Appreciable stresses at frequencies as high as 30,000 cps have been observed.

Assume for the moment that recording apparatus covering the frequency range to 30,000 cps is required, capable of resolving individual cycles so they may be counted to find the number of cycles of vibration per revolution of the rotor. All galvanometer-type oscillographs can be discarded, as they cover only a small part of that frequency range. Cathode ray recording oscillographs will resolve 30,000 cps easily, assuming adequate film speed. Fifty cycles per inch of record can be counted with the assistance of a low-power magnifier. To obtain this with the 30,000-cps signal requires a film speed of 600 in. per sec, 50 fps. The only practical way of finding all the blade resonances of an engine or test rig is to perform a very slow acceleration through the speed range, thus making certain that no speed is missed. Such an acceleration commonly takes over 30 sec. This would require a 1500-ft supply of film run at 50 fps. No oscillograph camera is available with such a film transport mechanism.

An alternate method is first to run an acceleration making visual observations of a cathode ray oscilloscope to find speeds at which appreciable stresses occur, then to attempt to hold the rig speed constant long enough at each stress peak to permit taking a short record at high film speed. Cameras capable of doing this are available. One type, as shown in Fig. 10, has a drum on which a 3-ft strip of film is wrapped. The drum is rotated to get a film speed of 50 fps past the lens. This speed could be made higher with no extensive changes. Inside the drum are supply and take-up reels for 100 ft of film. The film transport inside the drum is motor driven, and can be operated between records while the drum is rotating at full speed. When this camera is used, records 3 ft or less in length can be taken in rapid sequence at a drum speed high enough to resolve 30,000 cps. The disadvantage of this technique is that one is never certain that the stress was at its peak value during the fraction of a second that the record was taken.

Developments in magnetic tape recording within the last few years have provided instruments capable of doing the required recording job. With present recording heads a tape speed of 15 in. per sec will record 15,000 cps; a tape speed of 30 in. per sec will record 30,000 cps. Even the smallest standard reels of tape will run continuously for 7½ min at 30 in. per sec. Recorders with two or more channels are available. The most serious disadvantage of tape recording of amplitudes at the present time is the presence of fluctuations in sensitivity along the tape. Specially selected tape should be used to minimize errors due to these fluctuations. Another precaution is that all the circuits, including the amplifiers, between the strain gages and the recording head must be carefully designed for uniform frequency response to 30,000 cps. Ordinary commercial equipment designed for the audio-frequency range may be inadequate. A two-channel tape recorder has been satisfactorily used, both on the ground and in flight, to record two strain-gage signals with engine-speed and time-marker signals superimposed upon them.

Another advantage of the tape recorder appears when the time comes for playing back the tape to study the signals recorded there. The tape drive speed may be reduced for playback by as much as an 8 to 1 ratio. The assumed 30,000-cps signal then appears converted to a 3750-cps signal, and may be rerecorded on common oscillographic equipment. A large percentage of the tape may be discarded as of no interest, only rerecording from the short sections on which appreciable stresses are found. The complete set of apparatus as used for tape recording of strain-gage signals and rerecording on 35-mm film is shown in Fig. 11. A

sample record obtained from the tape is shown in Fig. 12.

Some other uses for tape recorders may be mentioned, though somewhat outside the subject of instrumentation for mechanical problems. A portable tape recorder is very useful for study of the noise generated by aircraft engines. The aircraft industry has standardized instrumentation and practices for noise measurement[9] to permit comparison between results of different groups. This standard requires an octave-band frequency analysis of the noise. In many cases the noise may be transient, giving no time for such analysis. Or it may be inconvenient to conduct the frequency analysis at the time of test. The tape recorder is the best readily portable instrument with which the noise can be stored without distortion, for later analysis. Noises from various sources can also be placed in sequence on the tape and carried to an engineering conference for auditory comparison.

[9] See "Uniform Practices for Measurement of Aircraft Noise." Pub. by Aircraft Industries Association, Washington, 1952.

Temperature Measurement on Rotors

An extremely important mechanical feature of the rotor, particularly the turbine, is its resistance to centrifugal loads causing creep or bursting. In order to know the strength of the material, its temperature must be measured under engine or rig operating conditions. For the determination of turbine blade and disc temperatures on operating engines or component rigs, we employ thermocouples attached to the part under test. The thermocouple voltages are taken from the rotating members by a transfer device and the temperature indicated on standard potentiometer-type instruments.

Prior to the adoption of the thermocouple as the measuring element, the possible use of radiation pyrometry methods for making these observations was considered. The inability to measure at

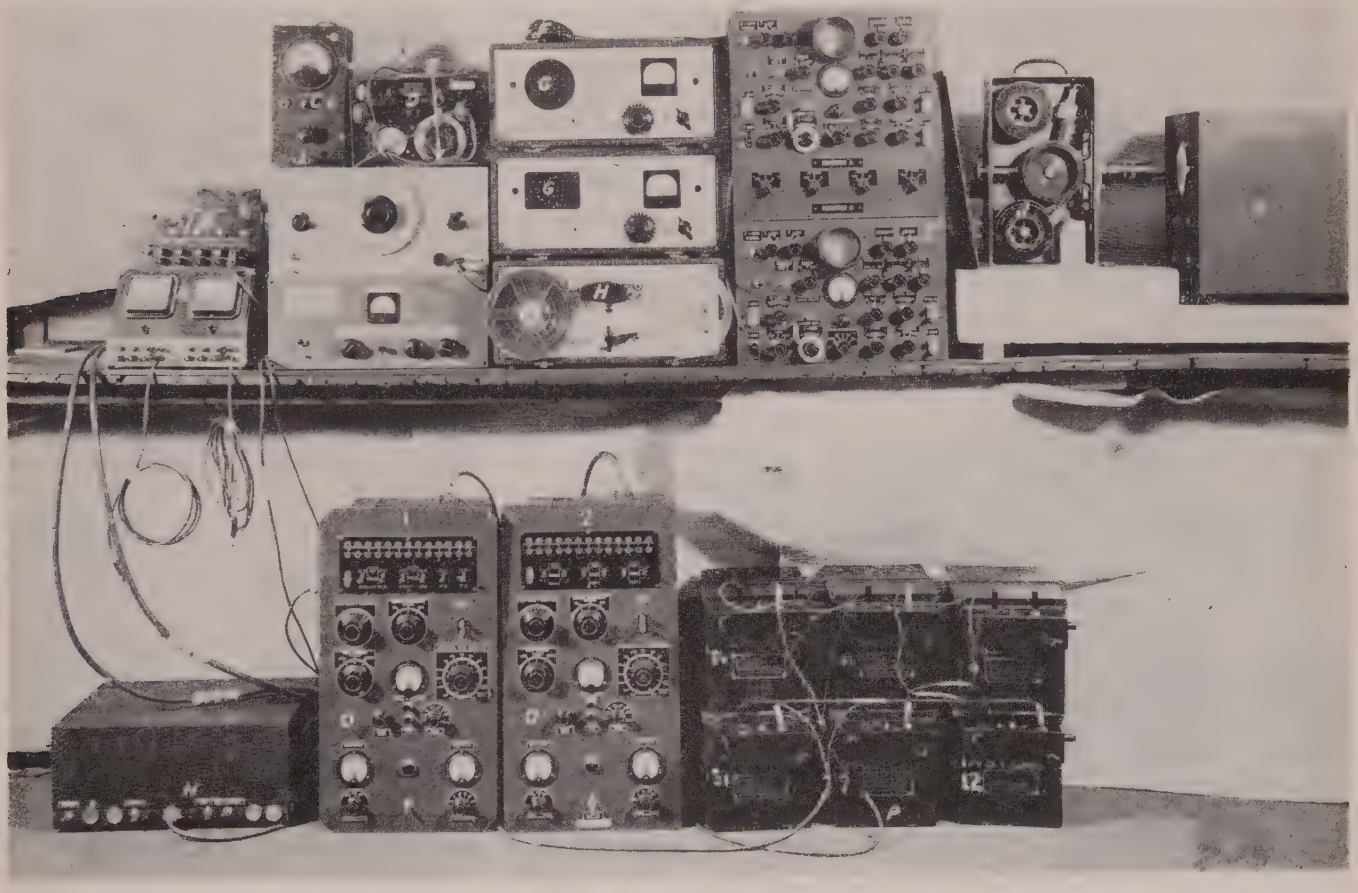


Fig. 11 – Equipment used to measure and record high-frequency vibratory strain signals

A – Cathode-follower strain-gage panel
B – 100/1000-cps signal generator
C – Electronic voltmeter
D – Microvolter
E – Impedance-matching calibration input cable
F – Identification microphone
G – Tape recorder amplifiers
H – Magnetic tape recorder
I – Dual-beam oscilloscope monitors
J – Continuous record camera
K – Dual-beam oscilloscope
L – Pip mixer and shaper unit
M – Audio oscillator
N – Power supply, cathode-follower strain-gage panel
O – Transformer-output strain-gage panels
P – 24-v batteries, strain-gage circuit power source

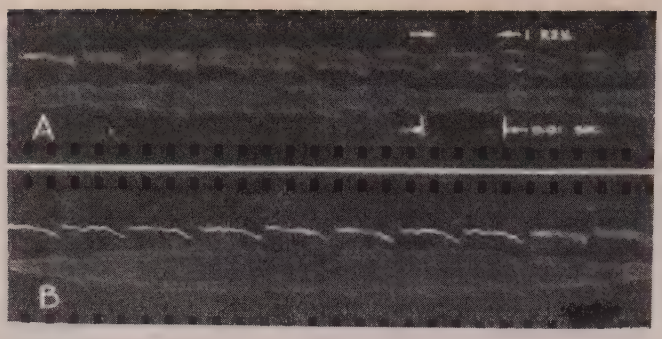


Fig. 12 – Records of turbine blade stress obtained from magnetic tape recording by playback into two-beam oscilloscope (all stresses at 48 cycles per revolution)

Record A: Engine speed 8085 rpm; upper trace: Blade A, radial gage at trailing edge 1⅛ in. from tip; lower trace: blade E, axial gage at tip, ⅝ in. from trailing edge

Record B: Engine speed 7955 rpm; lower trace: blade E, same gage as above

a specific finite point and at a fairly large number of such points on any one run was a drawback. In addition, errors possible through changing emissivity of the target during test, reflected radiation from burners or other sources, and changing transmission or reflection coefficients of the attendant optical system were all factors that complicated the problem. As a result, it was decided that using thermocouples offered the simplest and the quickest means of obtaining the desired measurements.

The technique for installing thermocouples on blades and discs was adapted from previously established methods developed for the use of high-temperature strain gages.[8] Chromel and alumel ribbon 0.005 in. by 0.030 in. is used in all areas where the leads are to be cemented to the blades or disc. This ribbon is extended by condenser-discharge welding to commercial 28-gage glass-insulated duplex chromel and alumel wire for leads to the transfer unit. The weld is made at a point sufficiently cool for use of glass-insulated wire and where a suitable means of anchoring the spliced length of wire to the disc is available.

Particular attention should be paid in forming the chromel and alumel ribbon so that it lays snugly against the surface of the cement precoat before the cement overlay is put on. The less the wire and cement project above the surface of the parts the better the chance of a successful installation. The actual thermocouple junction is formed by condenser-welding the chromel and alumel ribbon to the part at the point of desired measurement.

A review of the available literature regarding the transfer of thermocouple voltages from rotating machinery revealed such means as slip-ring brush methods, slip-ring compensation methods, or the induction method.[10, 11, 12] While some of these methods produced satisfactory results, we endeavored to construct a transfer unit that would permit continuous operation without the complication of controlled reference junctions.

Fig. 13 indicates the arrangement of the temperature transfer unit employed. Chromel and alumel rings are mounted on an insulated shaft and are connected by chromel and alumel wire to a chromel and alumel pin connector at one end of the shaft. Bearing against these rings are chromel and alumel wheels having a diameter twice the ring diameter, running in bearings in an insulated arm with loading provided by torsion springs. Cones of chromel and alumel material are spring-loaded against a recess in the center of their corresponding wheels, and chromel and alumel wires attached to these cones provide the connection to the indicating instrument. Thus, the unit has complete thermocouple-material continuity. The assembly is enclosed with a cover, and oil mist lubrication is provided for the wheel and shaft bearings and the cone contacts.

In operation, the limiting speed is determined by the increase in circuit resistance of the roller and cone contact points, which causes loss of sensi-

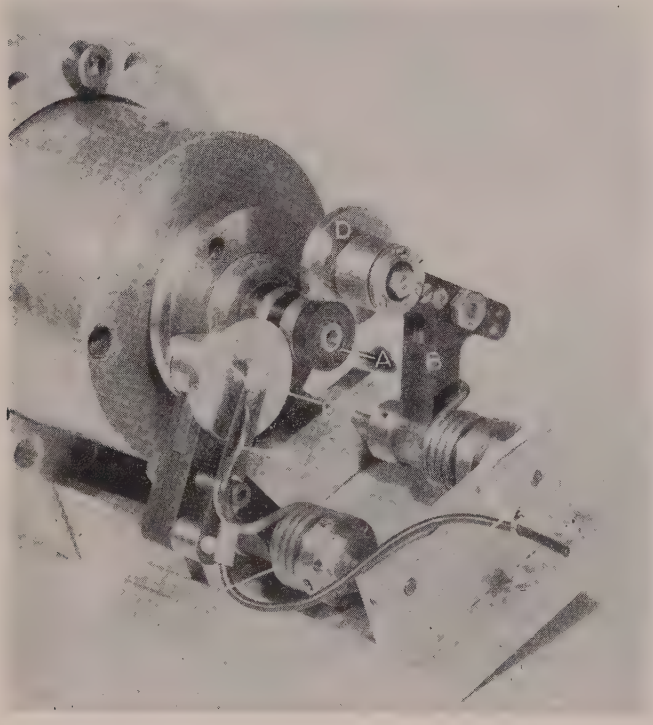


Fig. 13 – Temperature-transfer unit construction

A – Shaft with chromel and alumel rings
B – Plastic follower arm
C – Torsion spring
D – Chromel and alumel wheels
E – Chromel and alumel cone contacts at wheel centers
F – Leaf springs
G – Chromel and alumel lead wires

[10] See NACA TM-1080 (1945), "New Apparatus for Measuring Temperature at Machine Parts Rotating at High Speeds," by E. Gnam.

[11] See "Determination of Gas Turbine Blade and Rotor Temperatures at High Speeds." Pub. by Office of Naval Research, 1947.

[12] See NACA RM E50J23A (1951). "Methods for Connection to Revolving Thermocouples," by P. R. Tarr.

tivity in the self-balancing potentiometer-type indicating equipment. With a single roller follower per ring, this speed is approximately 5000 rpm, and with two roller-follower assemblies per ring, approximately 15,000 rpm. Since most testing is done between these limits, two roller assemblies per ring make up the standard configuration. Two sizes of transfer unit assembly are used, one capable of handling a single thermocouple at a time and the other, two thermocouples.

Connection of the transfer unit to the turbine shaft is accomplished through a plug-in type of connector shaft developed for strain-gage slip-ring applications.[6] This shaft was modified to include a remotely actuated selector switch,[12] which permits the measurement of 12 thermocouples, one at a time in one configuration, or 28 thermocouples, two at a time in a second configuration. Arrangement of the transfer unit and remotely actuated selector switch connector shaft is shown in Fig. 14.

Typical data taken from a turbine test are shown in Fig. 15. Six thermocouples were installed on the blade, three radially at the thickest point on the airfoil section and three circumferentially on the blade root. This particular installation was run approximately two hours and the desired data were obtained on a single run. Other instrumented turbine wheels run in component rigs have accumulated as much as five hours of continuous operation. Thermal shock on starting, particularly with a "hot" start, contributes greatly to shortening the useful life of high-temperature thermocouple or strain-gage installations.

The selector switch shaft has also been used with success on high-temperature strain-gage measurements, permitting the use of more gages than the strain-gage slip-ring assembly can handle at one time in a single setup. This is particularly advantageous when the life of the gage at operating conditions is marginal and the number of engine starts must be reduced.

Fig. 14 – Selector switch shaft and temperature-transfer unit
A – Selector switch actuator
B – Selector switch shaft
C – Transfer-unit cover

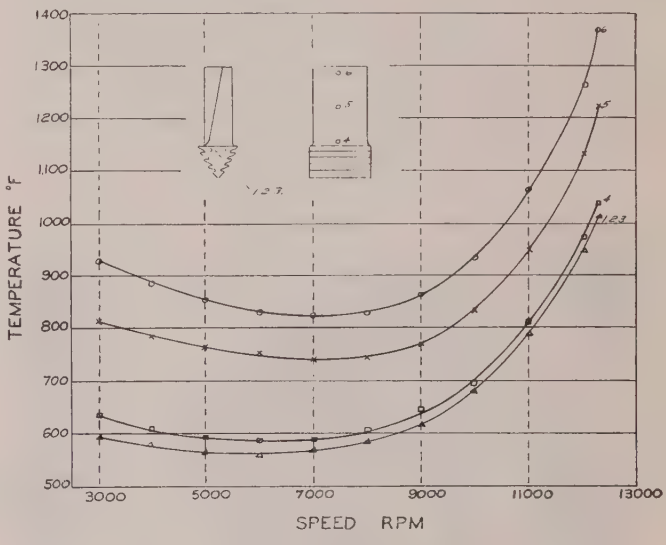


Fig. 15 – Turbine blade temperature versus rpm obtained with temperature-transfer unit

Instrumentation for Working Fluids

The introduction of the turbine engine, particularly the axial-flow multistage machine, increased greatly the amount of aerodynamic data required. With performance characteristics extremely sensitive to temperature and pressure distribution, flow angle, and velocity, determination of these factors and many others requires a multitude of accurate measurements.

Many of the pressure and temperature probes employed at P&WA for fluid flow measurements have previously been described.[13] Variations of these basic types are frequently used, the physical arrangement being modified to suit requirements at the particular point of measurement, but with

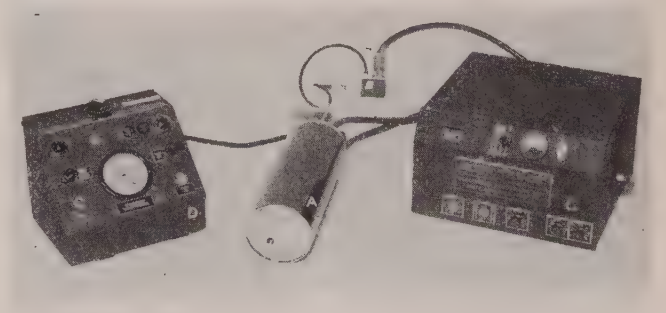


Fig. 16 – Automatic yaw-balance equipment
A – Probe positioner
B – Differential-pressure transducer
C – Automatic yaw-balance unit
D – Manual traverse control unit

[13] See SAE *Quarterly* Transactions, Vol. 2, January, 1948, pp. 104-116: "Instrumentation for Development of Aircraft Powerplant Components Involving Fluid Flow," by S. J. Markowski and E. M. Moffatt.

no modification of performance characteristics. Further information has been published on means of reducing errors in thermocouples used for high-temperature gas measurements. A recent bibliography on gas temperature measurement[14] contains these data as well as many other useful references.

A two-orifice probe was recently put into use for temperature measurements above the usable limit for chromel and alumel thermocouples and in locations such as burners where active combustion is taking place. This principle, originally proposed by Uehling,[15] is based on the gas flow through two orifices in series. Gas at a steady rate is made to flow through two orifices in series with an indeterminate amount of heating or cooling between them. The mass flow rate through the probe can be determined by measurement of pressures and temperature at the second orifice; hence, the temperature at the first orifice can be obtained by calculation from measurement of its pressures.

Comparison of probe readings at 3600 F with sodium-line-reversal temperature measurements has indicated a difference of approximately 50 F, while room temperature measurements can be made to about 3 F. Inherently, the probe has no radiation error.

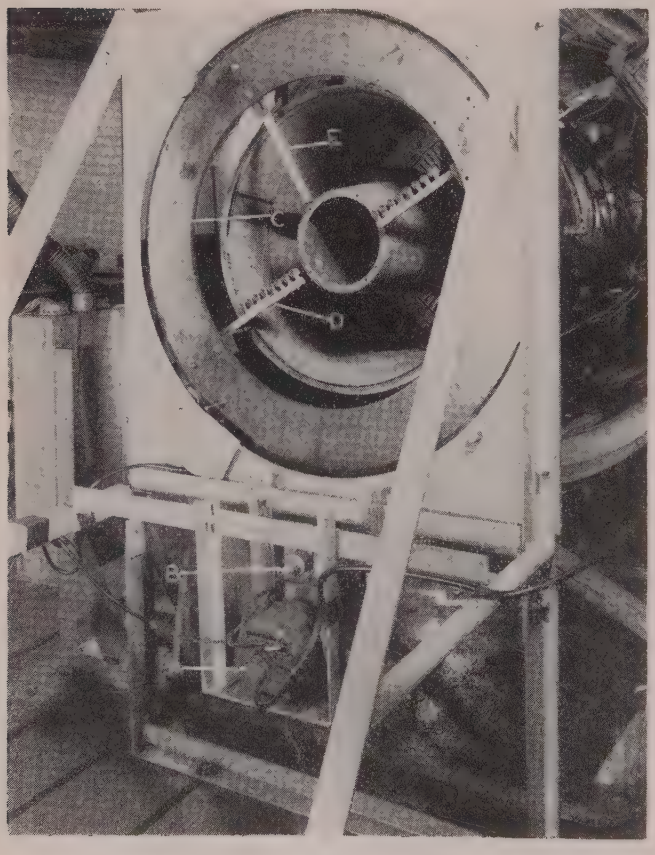


Fig. 17 – Tailpipe rake rotator installation
A – Motor
B – Selsyn transmitter
C – Rotating mount ring
D – Multiple shielded temperature rake
E – Total pressure rake

Temperature traverses in afterburners have been made with readings approaching 4000 F and routine measurements in burner test rigs are being made daily without engineering supervision. A two-coordinate plotter is being adapted for use with the probe to permit indication of the temperature directly on a nomographic chart with no calculation.

We are making extensive use of remote-controlled probe-positioning devices. There is hardly a point in the engine where temperature and pressure measurements have not been taken to determine flow characteristics. The basic positioning unit has been described,[13] but the addition of a self-balancing yaw mechanism has greatly increased its usefulness. Fig. 16 shows the apparatus used for yaw-angle determination with conventional yaw probes. The probe is installed in the positioner, which contains the actuating motors to provide translational and rotational motion. A differential-pressure transducer is connected to the yaw holes of the probe and the electrical output of the transducer fed to an amplifier in the yaw-balance unit. Output from the amplifier actuates the motor to rotate the probe in a direction to reduce the differential pressure to zero. Probe angle is read on a motor-driven counter in the yaw-balance unit, which is driven by a selsyn-servo loop, one selsyn being geared to the rotational motor in the positioning unit. Translational motion is imparted to the probe by the other motor, manually regulated from the traverse control unit. The amount of motion is measured on an impulse-type counter from a counting contact in the positioner. This self-balancing feature has reduced the time to make a traverse by a factor of two from that required when an operator balanced the probe by observing the differential pressure on a U-tube manometer.

Other special actuators have been constructed to permit circumferential traversing in annuli and traversing in three coordinates behind rectangular cascades. Fig. 17 shows a tailpipe rake rotator used to obtain radial and circumferential temperature and pressure distribution measurements in the tailpipe of an engine.

Means of speeding up data recording have been applied to probe traversing on stationary cascade and rotating compressor and turbine rigs for loss measurements. This system eliminates the tedious point-by-point traversing method and reduces the data-reduction time by producing directly on the test stand a visual plot and on counters the integrated values of the measured variables. Basic features of this automatic plotting and integrating

[14] See National Bureau of Standards Circular 513 (1951), "Bibliography on Measurement of Gas Temperature."
[15] See pp. 66-67 of "Powerplant Testing," by J. A. Moyer. Pub. by McGraw-Hill, New York, 1934.

equipment are illustrated in Fig. 18 as applied to an annular cascade.[16]

A probe-positioning device is mounted on a motorized platform, which can be moved circumferentially around the cascade. Installed in the probe-positioner is a combination probe, which gives simultaneous indication of the three variables, yaw angle, total temperature, and total pressure. Three strip-chart plotters are provided, one for each variable, the chart drive mechanism being tied in with the traverse motion by a servo loop with selsyns. The self-balancing yaw equipment previously described is modified so angular position will be indicated by the pen-motion of the yaw angle plotter. Voltages from the thermocouple in the probe and from the pressure transducer connected to the pressure tap on the probe are supplied to amplifiers, which drive the pens on the corresponding plotters. Connected to the pen-motion of each plotter is a potentiometer excited by a voltage proportional to traverse speed from a generator driven by the strip-chart drive motor. A portion of this voltage proportional to the magnitude of the variable in question is taken off the potentiometer and supplied to the amplifier of a feedback generator type of integrator. The motor driving the feedback generator is connected to a counter, which indicates the integrated value of the variable over the traversed distance. Fig. 19 shows a front panel view of this instrument, the plotters on the left and the integrators and operating controls on the right. Other investigators have reported making similar measurements by somewhat different techniques.[17, 18]

Usual methods of indicating steady-state thermocouple measurements are by self-balancing potentiometer or multipoint recorders. Pressure measurements are usually indicated on U-tubes or Bourdon-tube gages. Readings of these instruments are taken by test personnel and recorded on data sheets. Such methods, while satisfactory for a small number of measurements, are becoming inadequate timewise for the large quantities of measurements now required. When a test involves taking 300 or more individual measurements per point, recording for each point may take one-half hour or more. It can be seen that a system capable of recording automatically and at much greater speed would appreciably reduce operating time. Short reading time would permit measurements at essentially constant conditions, particularly important on engines or rigs difficult to hold on a set point, and would greatly increase the possibility of obtaining data on engine or rigs operating at conditions likely to cause quick failure. Greater accuracy

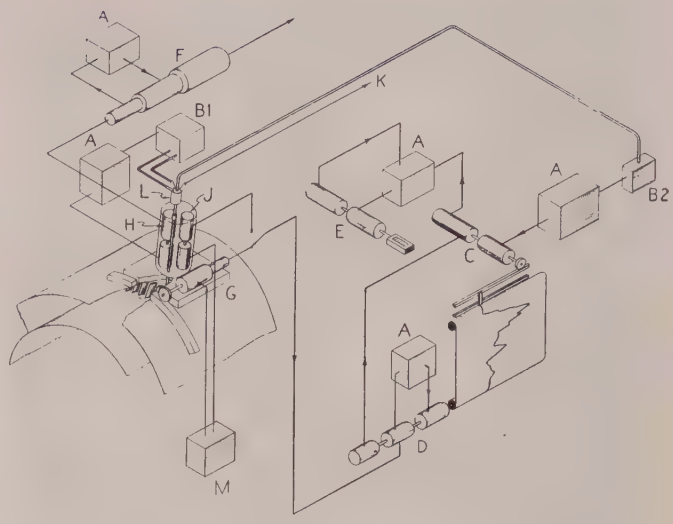


Fig. 18 – Schematic diagram of automatic plotting and integrating instrument

A – Amplifiers
B1 – Pressure transducer. Output voltage proportional to differential pressure from probe yaw holes
B2 – Pressure transducer. Voltage output proportional to probe total pressure
C – Pen motor and potentiometer. Pen deflection and potentiometer position proportional to voltage output from total pressure transducer B2, thermocouple K, or probe rotation follower potentiometer P
D – Strip chart drive motor, selsyn, and generator. Paper speed and generator output synchronized with traverse speed through servo loop with selsyns
E – Integrator motor, counter, and feedback generator. Motor speed proportional to voltage product of generator output D and potentiometer position C
F – Probe rotation follower motor, selsyn, and potentiometer. Potentiometer synchronized with probe rotation position through servo loop with selsyns
G – Circumferential traverse motor and selsyn. Motor operated from traverse control unit M
H – Probe rotation motor and selsyn. Motor controlled by voltage output from transducer B1 to rotate probe to null balance
J – Radial traverse motor and selsyn. Motor operated from traverse control unit M
K – Thermocouple
L – Probe. Combination type for simultaneous determination of yaw angle, total pressure, and total temperature
M – Manual traverse control unit

should be obtained by the elimination of the human error in reading and transcribing data. Various laboratories are developing equipment for such automatic data recording, with the output in the form of typewritten columns of figures, in the form of punched cards, or both.

As mentioned earlier a satisfactory method of obtaining slowly varying pressure measurements is by photographing steady-state indicating instruments. With long distances between the indicating instrument and pressure source or with increased acceleration rates, this method is unsatisfactory and it is necessary to resort to transducers. Such transducers should have small zero-shift, hystere-

[16] Development of this automatic plotting and integrating apparatus has been a cooperative project between P&WA and the United Aircraft Corp. Research Department.

[17] See ASME Transactions, Vol. 74, July, 1952, pp. 685-693: "High-Speed Cascade Testing Techniques," by F. H. Keast.

[18] See ASME Transactions, Vol. 62, August, 1940, pp. 479-488: "Automatic Integating Pressure-Traverse Recorder for Study of Flow Phenomena in Steam Turbine Nozzles and Buckets," by H. Kraft and T. M. Berry.

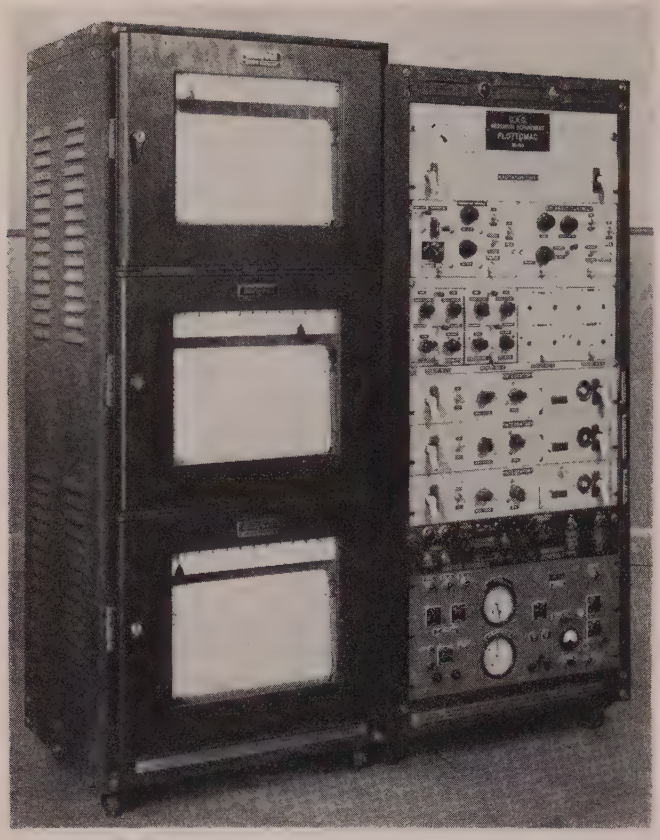


Fig. 19 – Automatic plotting and integrating instrument

sis, and changes in sensitivity with time, temperature, and pressure and, of course, a frequency response adequate for the particular measurement. Vibration insensitivity is not always required, since the transducer can be shock mounted and connected to the pressure source by flexible tubing. Tubing size and length should be carefully selected so that the pressure-time output of the transducer will not be distorted from the actual pressure fluctuation.[19, 20]

High-frequency pressure measurements require that the transducer be so located that its sensing element is exposed directly to the fluctuating pressure, without intervening restricting passages or cavities. This results in the unit being mounted directly on the engine or part, which may be subject to vibratory accelerations. Such transducers are thus required to have a high natural frequency or to have a construction balanced against acceleration in order to minimize the vibration effect. High sensitivity is necessary to detect small pressure fluctuations superimposed upon high steady-state pressures. Zero-drift can usually be neglected in oscillating pressure measurements. Some form of cooling is often necessary to prevent damage to the transducer in hot locations. Pressure transducers of the capacitance types[21] or the strain-gage type[22] have been developed that are adequate for almost all requirements.

For the determination of pressure distribution on blades of low-temperature research turbines a pressure-transfer device is used based on a design suggested by the Langley Aeronautical Laboratory of the NACA.

Fig. 20 illustrates the basic features of the unit. Commercial ball bearings are used having synthetic rubber seals on each bearing face. These seals, reversed from their normal position, are pressed into grooves in the outer race and bear against a formed surface on the inner race of the bearing. A water-cooled housing is provided to contain the bearings. A shaft, having a hole drilled from end to end, is installed in the bearings. Sealing between the bearing outer race and housing and the inner race and shaft is accomplished with O-rings. A cap fitted to one end of the housing locks the outer races of the bearings and contains the stationary pressure tap. Connection of the transfer unit to the turbine shaft is through a remotely actuated pressure selector shaft having drilled passages arranged to permit selection of 24 pressures, one at a time.

A device of this type has been operated continuously for a period of 40-hr on the test bench at speeds between 10,000 and 15,000 rpm with a pressure differential of 20 to 25 in. of mercury across the bearing. Quality of seal, while not perfect, was sufficiently good to permit measurement with insignificant error of pressures supplied through the restriction of the drilled holes in the turbine blades. The ultimate life of the seal is unknown at these operating conditions but no change in the rate of leakage was observed during the 40-hr test. This type of seal construction is readily adapted to multichannel operation by stacking the bearings and introducing and taking off the pressures in the spaces between bearings. Fig. 21 illustrates the change with speed in pressure distribution on a turbine blade, determined with this equipment.

Measurement methods applied to mass airflow determination for turbine engines are no different than those used for piston engines except for the greatly increased rate of flow. Systems capable of handling approximately 600,000 lb per hr and required to have low losses in order to keep power requirements within reasonable values have been designed with orifice diameters up to 38 in. and pipe sizes up to 54-in. diameter. The primary flow elements are commercial venturi meters or smooth approach nozzles or orifices installed according to accepted metering standards.[23, 24] Nonstandard installations, usually necessitated by space considerations, or installations of questionable accuracy, are verified by comparison with a standard setup

[19] See ASME Transactions, Vol. 72, July, 1950, pp. 689-695: "Attenuation of Oscillatory Pressures in Instrument Lines," by A. S. Iberall.

[20] See NACA TN-1819 (1949), "Response of Pressure-Measuring Systems to Oscillating Pressures," by I. Taback.

[21] See SAE Transactions, Vol. 52, November, 1944, pp. 534-556: "Improved Indicator for Measuring Static and Dynamic Pressures," by C. E. Grinstead, R. N. Frawley, F. W. Chapman, and H. F. Schultz.

[22] See *Journal of Aeronautical Sciences*, Vol. 16, October, 1949, pp. 593-610: "New High-Performance Engine Indicator of Strain-Gage Type," by C. S. Draper and Y. T. Li.

installed in series with the system or by pressure and temperature traverses of the primary element.

For the investigation of flow characteristics of wakes behind rotating blades, the usual types of velocity probes depending on pressure measurements are inadequate because of their very slow frequency response. The hot-wire anemometer type of instrument lends itself to this type of measurement, and we have made limited use of it for qualitative analysis behind rotating compressor blades.

Fuel-flow metering is required for a very wide range of flow. Only 20 lb per hr is required for some component testing, while approximately 60,000 lb per hr for future engine and afterburner combinations is now used as a design figure for new test stands. Variable-orifice meters having a float in a taper bore tube through which the fuel flows are used for steady-state measurements, where the meter may be mounted in direct view of the operating personnel. The position of the float relative to a scale on the tube indicates the flow rate. These meters may also be obtained with remote-indicating devices. Positive displacement piston-type meters or impeller-type velocity meters are also employed when remote indication is required. The piston motion of the positive-displacement meter is magnetically coupled to a selsyn transmitter. A drag-cup type of tachometer calibrated in flow-rate is driven by the receiver selsyn. Total-flow indicating meters are also available with an impulse type of counter actuated by contacts driven by the piston. The impeller type of velocity meter generates by magnetic induction a fluctuating voltage whose frequency is proportional to flow velocity. Flow-rate may be indicated on an electronic frequency meter or by counting the voltage pulses for a given time interval on an electronic counter.

For varying flow-rate determinations such as during acceleration, the positive-displacement or velocity type of meter is used. In addition, a moving-vane type of meter having a potentiometer connected to the vane is employed. Position of the vane and therefore the voltage output from the potentiometer is proportional to flow rate. These meters are readily adapted to indication on oscillograph recorders used for other acceleration data.

All these meters have too much inertia to be usable for rapidly fluctuating flow determination, such as occurs when a fuel-metering valve becomes unstable. Such measurements can be made using the direct electrical effect of fluid flowing through a magnetic field.[25, 26] Commercial instruments are available employing this principle.

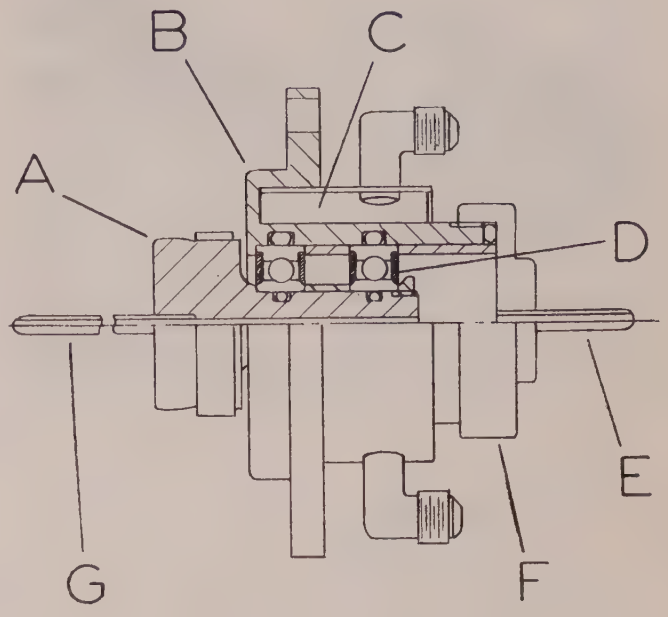


Fig. 20 – Pressure-transfer unit construction
A – Shaft
B – Housing
C – Cooling-water jacket
D – Bearing with synthetic rubber seal
E – Pressure tap to manometer
F – Cap
G – Pressure tube to selector shaft

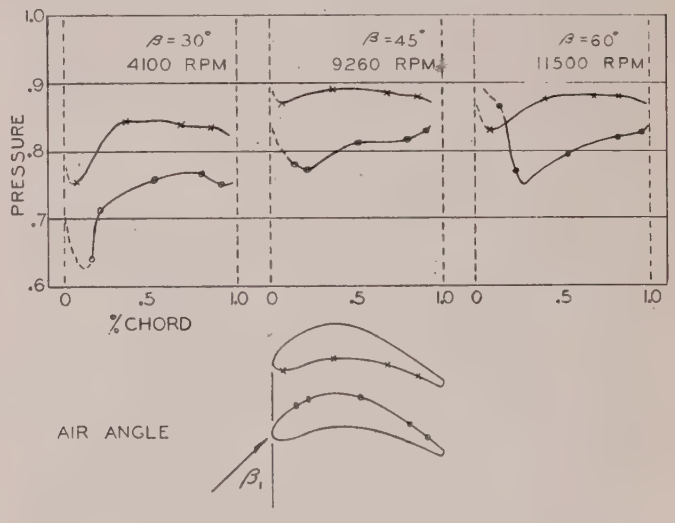


Fig. 21 – Pressure distribution on turbine blade at various rpm's, obtained with pressure-transfer unit

[23] See ASME Power Test Codes PTC-19.5: 4-1949, "Supplement on Instruments and Apparatus – Flow Measurement." (Being revised.)
[24] See NACA TM-952 (1940), "Standards for Discharge Measurement with Standardized Nozzles and Orifices."
[25] See NACA RM E50L12 (1951), "Electromagnetic Flowmeter for Rocket Research," by L. Jaffee, B. A. Coss, and D. R. Daykin.
[26] See *Review of Scientific Instruments*, Vol. 22, January, 1951, pp. 43-47: "Electromagnetic Flowmeter for Transient Flow Studies," by J. S. Arnold.

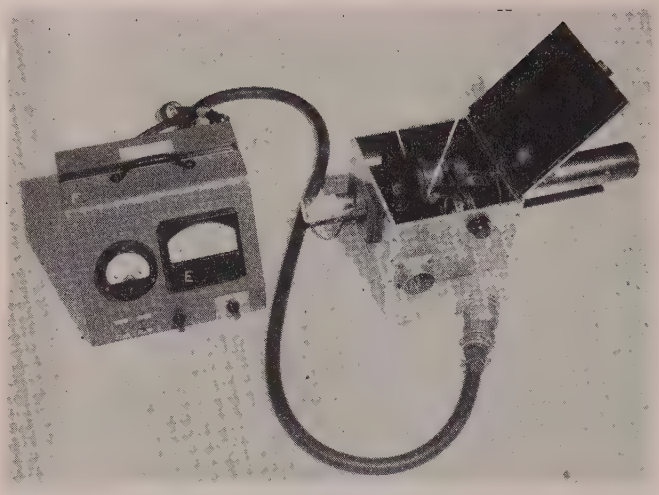


Fig. 22 – Photoelectric radiation detector
A – Collimating tube
B – Toothed chopper disc
C – Water-cooled jacket for photocell
D – Remotely positioned filter
E – Radiation indicating meter
F – Chassis containing high-voltage power supply, amplifier, and controls
G – Telescope eyepiece for aiming instrument

Instruments for Burner Studies

A group of instruments particularly useful for burner development includes apparatus for gas analysis, radiation measurement, and high-speed photography.

Analysis of the gas composition of samples from various points within a burner will give a measure of the local fuel/air ratio, and analysis of the gases discharged from a burner will give a measure of the completeness of combustion, or burner efficiency. While in college most mechanical engineers are exposed to the Orsat type of gas analysis apparatus used for laboratory tests of flue or exhaust gases. As a result of this brief exposure, many of them retain the impression that, while such apparatus may give good exercise to the arm lifting and lowering the mercury bottle, it is unlikely to give an accurate measure of gas composition. With proper maintenance and with a trained operator, absorption-type analyzers can give consistent and accurate gas analyses. However, these absorption-type analyzers in which the change in volume of an isolated sample of gas is measured as various components are absorbed, one after the other, are too slow. From testing just one burner design at several altitude conditions, several airflows, and several fuel/air ratios, with sampling from a dozen or so points, many hundred samples could be collected in a few hours. These precision laboratory-type analyzers are, hence, reserved for calibration purposes, checking the composition of bottles of gas mixtures that are compounded as required. These mixtures are then used to check the calibration of direct-reading, continuous-flow gas analyzers. Various physical phenomena, such as the magnetic permeability of oxygen, the thermal conductivity of carbon dioxide, or the temperature rise of a platinum wire when hydrogen or carbon monoxide suffers catalytic combustion on its surface, are used in various direct-reading commercially available instruments.[27]

More of a problem than the selection of a suitable analyzer is the procuring of a correct sample to analyze. The gas sample is usually extracted from the burner through an upstream entrance into a small tube in a water-cooled probe. The flow rate of the gas sample may be controlled to get a wide range of velocities into the entrance to the sampling tube. The analysis of the sample is usually found to vary with sampling velocity. This results from the fact that the gas mixture being sampled in or just downstream of the region of combustion is not homogeneous.

The curvature of streamlines of flow into and near the sampling entrance is affected by the velocity of flow into the entrance. Lighter density portions of the gas stream follow a different streamline than the portions that may have greater density and are harder to deviate from their original flow direction. The most representative sample is obtained from an upstream-facing sampling tube with the sampling rate adjusted so that the velocity into the sampling entrance is equal to free-stream velocity. Then there is the minimum bending of the streamlines approaching the entrance, and the minimum discrimination between light and heavy portions of the inhomogeneous mixture.

With a sampling probe and technique that will provide a representative sample from one point, traverses of a burner are made to study the variation of fuel/air ratio or completeness of combustion from point to point. If an analysis for the overall combustion efficiency of a burner is desired, care must be used to sample from sufficient points over the outlet area to provide a true average. Steep gradients of velocity and gas composition are likely to be found across the outlet.

A second instrument for burner studies is the photoelectric radiation detector, as shown in Fig. 22.[28]

This contains a quartz lens, which, through a window in the burner wall, may be sighted upon a nonradiating pocket in the far burner wall. The optical system focuses any radiation within a narrow field of view upon a multiplier phototube. The phototube output, a measure of the radiation present in the field of view, is presented on a remote meter. This indication permits the engineer to tell relative intensity of flame at the particular location. The light beam to the photocell is chopped by a toothed disc to obtain an a-c signal that can easily

[27] See "Gas Analysis and Testing," by V. J. Altieri. Pub. by American Gas Association, New York, 1946.
[28] Development of the basic instrument was carried out by the United Aircraft Corp. Research Department.

be amplified. Between the lens and the photocell is also a remotely controlled positioning device carrying two optical filters, each of which removes ultra violet radiation of shorter wavelength than a selected value. By reading the output with each of the two filters interposed, and with no filter, a sort of color analysis of the flame is obtained. The filter cut-off wavelengths are so chosen as to permit detecting presence or absence of some of the more important chemical radicals occurring in hydrocarbon combustion.

The remaining type of instrumentation that finds its principal use in burner studies comes under the general title of high-speed photographic equipment. Most simple are various high-intensity, short-duration light sources used to obtain snapshots of high-velocity objects. Typical uses for such apparatus are the photographing of spray patterns from fuel nozzles and the photographing of schlieren patterns of airflow around the various airfoil sections. Also very useful are moving pictures taken at rates of one thousand to fifteen thousand pictures per second and projected at normal speed to slow down the phenomenon for observation. Such movies permit studying the motion of burner flames while starting the engine or while running with unsteady combustion. A complete and up-to-date bibliography of high-speed photography of many kinds is available, in conjunction with reports upon the latest techniques.[29]

Personnel and Instruments for Instrumentation

It is evident from a study of the instruments that have been applied to turbine engine testing that almost without exception they involve electrical or electronic circuits. Even though many instruments are commercially available, these frequently need modification to fit a wide variety of applications. They also need frequent calibration and servicing, particularly when subjected to the environment of an engine test house. Also, many new instruments must be developed and constructed to fit specific needs. In order to accomplish this calibration, modification, servicing, development, or construction, a large supply of auxiliary instruments is required, together with electronics personnel skilled in their use. All the voltmeters, vacuum tube voltmeters, ohm-meters, oscilloscopes, oscillators, square-wave generators, tube checkers, and other instruments used by the electronics engineer must be available.

Where electrical or electronic instruments are in continuous use, some thought must be given to standards to which all calibrations may be referred. The basic quantities for which standards must be provided are voltage and frequency. Portable voltmeters and portable tuning forks or crystal oscillators may be used to calibrate the instruments used for engine testing, when they are in place on the engine test stand. Sooner or later some unexpected trend of test data will cause the accuracy of the calibration to be questioned. Usually, the answer cannot await the sending of the portable calibrating instruments to the maker or another laboratory for checking. Too often an answer is obtained by collecting several meters (voltmeters, as an example) all of equal precision when new, comparing their results when connected in parallel, and assuming that the average is correct, that the closest three out of four are correct, or that one is more correct because it is newer. Such assumptions should be replaced by reference standards, accurate to one order of magnitude better than any of the portable or engine test instruments, and kept under lock and key to be used only for calibration purposes. Voltage standards to one-tenth of one per cent may be obtained from precision meters. If more precision is required, as in checking the portable standard cells used in potentiometers, a bank of saturated standard cells may be maintained under constant temperature to give an accuracy of one one-thousandth per cent or better. Frequency standards are readily obtained by tuning a radio receiver to one of the frequencies transmitted over WWV by the National Bureau of Standards.

In addition to a staff of skilled electronics personnel, an instrument section must have personnel familiar with sound and vibration theory, the physics and thermodynamics of temperature and pressure measurements, and with the performance of the engines or components to be tested. It is extremely easy to get erroneous or misleading measurements or to draw the wrong conclusions if the basic principles of both the instruments and the test specimen are not completely understood by the men setting up the instrumentation. Once the instrumentation has been set up, calibrated, and checked, it should be possible for less specialized personnel to carry on the testing, if the instruments have been properly designed and maintained.

Personnel capable of constructing and repairing some of the instruments must also have quite specialized skills. The construction of strain gages or hot-wire anemometer probes from wire a fraction of a thousandth of an inch in diameter requires a patience and manual dexterity found in very few people. Welders capable of joining microscopic hypodermic tubing into small air-pressure probes without plugging the holes are equally rare.

Perhaps the most valuable attribute of instrumentation personnel, whether engineer or technician, is a very skeptical attitude toward the accuracy of the data obtained from their instruments. With a good supply of the special instruments described, and with this skeptical attitude leading the personnel continually to check and recheck the performance of the instruments, an instrument section is well equipped to measure practically any quantity that the engine development engineer may desire to know.

[29] See *Journal of Society of Motion Picture and Television Engineers*, "High-Speed Photography," Vol. 3, 1951.

DEVELOPMENT OF HIGH-

IT has been said by many that this is the age of speed. The U. S. Air Force has concentrated more and more emphasis on speed, on altitude, and on performance. We have aircraft operating from sea level to more than 40,000 ft; aircraft operating at more than 600 mph; aircraft operating from torrid climates, arctic climates; big aircraft, little aircraft. Many of these aircraft have landing gears, wing flaps, dive brakes, automatic controls, bomb-bay doors, gun turrets, brakes, and many other actuating and dissipating mechanisms. What transmits or dissipates the energy required to operate these units? It is generally a hydraulic fluid—a necessary and vital component for the satisfactory operation of present-day and future aircraft. Temperature extremes encountered by these aircraft demand a versatile fluid. The development of hydraulic fluids has been—and continues to be—an interesting and challenging research and development problem.

Prior to World War II two fluids were commonly used by the services: (1) Specification 3580, was a petroleum-base fluid; (2) Specification 3586, was a castor-oil-base fluid comparable to the automotive brake fluids. The use of the castor-base fluids was rapidly decreasing at the outset of World War II, due to some of its poor low-temperature properties and the critical shortage of natural rubber seals. The principal, if not the only, objection to the mineral-oil-base fluid (Specification 3580) was

HYDRAULIC fluids for high-speed, high-altitude aircraft need to encompass the temperature range —65 F to 450 F and over. In addition, the other physical properties must be satisfactory.

Since none of the fluids now being used in hydraulic systems are completely satisfactory, the Air Force has instituted a research program designed to develop a fluid that is capable of meeting these severe requirements.

Of the fluids investigated, the silicates and the polysiloxanes are considered to have real possibilities, although both need further development work done on them to take care of certain deficiencies.

The Authors

J. C. MOSTELLER is assistant chief of the Petroleum Products Branch, Materials Laboratory, at Wright Air Development Center. He is responsible for the technical supervision of the United States Air Force research and development program for hydraulic fluids, greases, fuels, lubricants, and related materials. Prior to employment by the USAF in 1946, Mr. Mosteller was with the Atlantic Refining Co.

LT. JOHN A. KING is in charge of the development section, Petroleum Products Branch of the Materials Laboratory, Wright Air Development Center. This section conducts fluids and lubricant research and development for the USAF.

TEMPERATURE HYDRAULIC FLUIDS

J. C. Mosteller and Lt. John A. King,
Wright Air Development Center

This paper was presented at the SAE National Aeronautic Meeting, New York, April 22, 1953.

its low-temperature properties. Development of Specification AN-VV-0-366 (now the MIL-0-5606 fluid) followed. Specification MIL-0-5606 demonstrated satisfactory service in World War II aircraft operating over the temperature range of —65 F to above 180 F, and is still satisfactory for the conventional aircraft of today.

In order to accomplish its military mission the U. S. Air Force must continually develop and procure new and improved equipment. In doing this, great emphasis has been placed on higher speed, longer range, and higher altitudes for its aircraft, either piloted or not. Speed has been achieved by advancements in airframe and propulsion design, whereas longer range is being accomplished through a combination of propulsion unit design and miniaturization of equipment within the airframe. Up to this time the maximum operating temperature of an aircraft was predicated by heat due to solar load, which was established as 160 F plus a small amount of additional heat due to proximity to heat-producing components and the dissipation of energy, which gave a temperature requirement of a maximum of 200 F for a hydraulic system. However, with the increase in speed and also the increase in the operating altitude, temperatures of hydraulic systems will rise abruptly several hundred degrees, due to proximity to heat-producing components; that is, engines, miniaturized electronic equipment, skin of aircraft, and the like, heat produced by dissipation of energy from hydraulic systems of higher pressure, and low heat transfer from such heat-producing bodies in the rarified air. These heat-producing bodies will increase the maximum operating temperature of hydraulic system from its normal 180 F temperature to above 450 F. Combined with this high-temperature requirement are the always present low-temperature requirements, imposed by the necessity to operate aircraft globally. Thus, any improvement in hydraulic fluids must be directed toward encompassing the temperature range of —65 F to about 450 F.

A study of the various applications in which higher operating temperatures are or will be imposed on the hydraulic fluid pointed out several necessary properties. Of primary importance was stability of the fluid to such temperatures, combined with good oxidation resistance and low vapor pressures. In addition to the primary requirement, the fluid should be noncorrosive to such metals as steel, titanium, aluminum, possibly silver and beryllium alloys; be nontoxic; exhibit a viscosity of 3.5 cs at 400 F, preferably at 450 F or higher, combined with 2500 cs at —65 F; be stable to hydrolysis; exhibit stability to shear; have a pour point below —65 F and a flash point above 400 F; be nonfoaming; and—not to be forgotten—it must lubricate steel-on-steel sliding and rolling surfaces. Quite a lot to expect from a molecule.

Examination of the hydraulic fluids now being utilized by the military services or in industry allows one to visualize the complexity of the problem. Those fluids satisfying the low-temperature requirements, in whole or in part, have either a mineral oil base (now used in U. S. Air Force aircraft, MIL-0-5606), a water-glycol base (used by the Navy, MIL-F-7083), a dibasic acid ester base (used in the U. S. Air Force constant-speed alternator drives, MIL-L-6387), or they are the commercial nonflammable hydraulic fluid Skydrol. As can be seen by examination of the viscosity-temperature characteristics of these fluids (shown in Fig. 1) and by comparison of the physical proper-

Table 1 – Physical Properties of High-Temperature Fluid – Desired versus Actual

	Requirement of High-Temperature Fluid	B MIL-L-6387	A Skydrol	D MIL-O-5606	C MIL-F-7083
Viscosity at −65 F, cs	2500	7500	....	....	....
Viscosity at −40 F, cs	....	1500	7000	500	550
Viscosity at 0 F, cs	....	....	....	....	....
Viscosity at 100 F, cs	....	....	15.5	14.3	10.0[a]
Viscosity at 210 F, cs	....	4.5	3.85	5.1	....
Viscosity at 400 F, cs	3.5	....	....	....	....
Vapor Pressure at 400 F, mm Hg	3	....	....	....	....
Hydrolytic Stability	Satisfactory	Satisfactory	Satisfactory	Satisfactory	Satisfactory
Thermal Stability	Satisfactory	Satisfactory	Unsatisfactory	Satisfactory	Unsatisfactory
Oxidation and Corrosion (Inhibited)	Satisfactory	Satisfactory	Marginal	Satisfactory	Unsatisfactory
Flash Point (min), F	400	350	....	200	....
Pour Point, F	−65	−75	−65	−75	−65
Wear – Steel on Steel	Satisfactory	Satisfactory	Satisfactory	Satisfactory	Questionable

[a] Viscosity at 130 F.

ties to those desired of a high-temperature hydraulic fluid (Table 1), no one fluid meets the need.

In view of the above noted deficiencies in present-day hydraulic fluids an inclusive survey was made by the U. S. Air Force research and development laboratories for another breed of molecule that would possess a majority of the properties desired. Particular attention was directed to the dibasic acid esters and the organo-phosphorus compounds, as these materials, represented in present-day fluids as MIL-L-6387 and Skydrol, respectively, appeared to exhibit some promise, as illustrated in Table 2. The studies included organo-phosphorus compounds; esters of mono- and dicarboxylic acids, carbonates, borates, polyglycols, polysiloxanes, silicates, and halogen-containing compounds. Mineral oils were excluded from this program because of their poor viscosity-volatility characteristics. Let us now discuss some of the

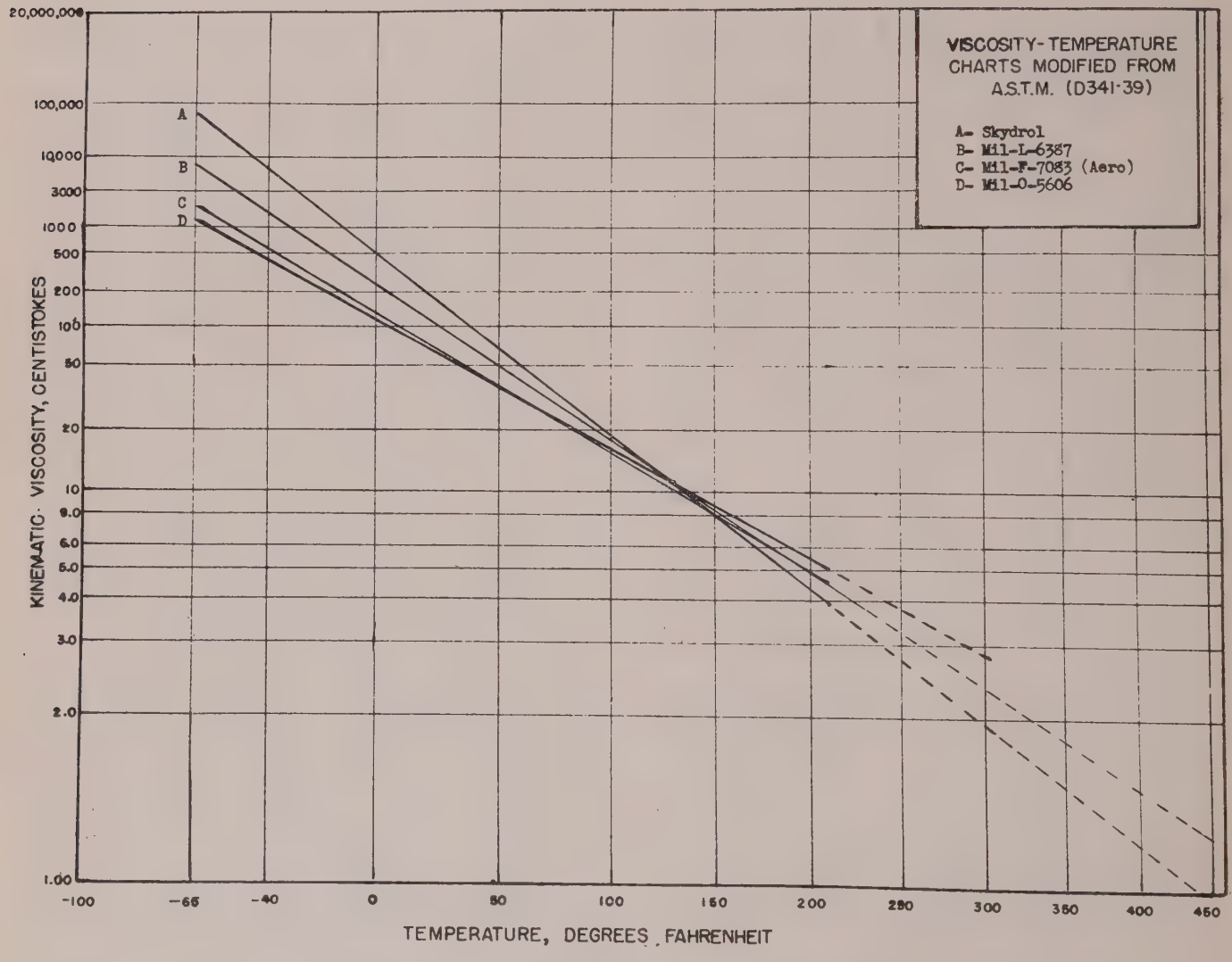


Fig. 1 – Viscosity-temperature characteristics of present-day hydraulic fluids

advantages and disadvantages of the above synthetic materials thus far synthesized and evaluated by the USAF.

Organo-Phosphorus Compounds – The organophosphorus compounds, and in particular the phosphates, have reduced flammability and in general have good lubricating properties. They have viscosity-temperature characteristics equivalent to highly refined petroleum stocks and are superior in volatility properties. The viscosity properties of these materials rival those of the silicates but are far inferior to those of the silicones, as shown in Table 2. Compounds such as the dihexylhexane phosphonate and the various benzene phosphonates have been examined for hydraulic fluid formulations. Materials of this class will require additives to thicken to the required viscosity at high temperature as well as to impart oxidation and hydrolytic stability. However, the phosphates exhibit questionable thermal stability for extended periods of use in the presence of metal catalysts.

Dicarboxylic Acid Esters – The mono- and dicarboxylic acid esters exhibit lubricating properties slightly less or at least equal to the fluids with a mineral oil base but far exceed the mineral oils in viscosity-temperature and viscosity-volatility characteristics. They possess high flash points combined with low pour points, good thermal stability, and, when properly inhibited, are noncorrosive to metals. They would require the use of a suitable viscosity index improver to meet the viscosity requirements of a high-temperature hydraulic fluid. Materials of the type di-2-ethylhexyl sebacate, isooctyl azelate, or isooctyl adipate offer considerable hope in the high-temperature hydraulic fluid development program.

Carbonates – The viscosity-volatility and viscosity-temperature characteristics of the carbonates are somewhat superior to the petroleum-base stocks. However, the dialkyl and cyclic carbonates prepared from low-molecular-weight alcohols or glycols are not sufficiently stable in the presence of water, with partial decomposition taking place, producing the alcohols, glycols, and carbonic acid. Attempts to reduce this complete or partial hydrolysis by preparing carbonates from alcohols having from 10 to 16 carbon atoms have resulted in compounds having too great a viscosity and questionable thermal stability.

Borates – These materials exhibit viscosity-temperature characteristics equal to or better than the petroleum-base stocks; however, their poor hydrolytic stability prohibits their consideration as high-temperature hydraulic fluids.

Polyglycols – Compounds of this group, known as polyalkylene glycols, and derivatives, present a possible source for high-temperature hydraulic fluid, with the exception of questionable thermal stability over prolonged usage in the presence of metal catalysts. These materials do exhibit good lubricating qualities, viscosity-volatility characteristics, and viscosity-temperature relationships, and are equal to or better than petroleum stocks but not sufficient to meet the requirements of a high-temperature hydraulic fluid except by the use of suitable thickening agents.

Silicates – Materials of this class, represented by compounds such as tetra-2-ethylhexoxy silane and tetra-2-ethylbutyoxy silane, dimers and higher polymers thereof, exhibit viscosity-temperature, viscosity-volatility characteristics far exceeding a petroleum stock and in some cases approaching those of the polysiloxanes. Compounds of this type are stable to shear and, when properly inhibited, are stable to oxidation and hydrolysis. Studies have shown that either polysiloxanes or the "acryloids" are effective viscosity index improvers and will produce blends meeting the viscosity requirements of the high-temperature hydraulic fluid. The flash point and high-temperature stability of these materials are slightly questionable and require further research and development studies.

Polysiloxanes – Materials of this class of compounds, as represented by the polymethylsiloxanes, polymethylphenylsiloxanes, and possibly polyethylsiloxanes, have unquestionably excellent viscosity-

Table 2 – Properties of Various Synthetic Materials

	Requirements of High-Temperature Fluid	Organo-Phosphorus Compounds	Diesters	Carbonates	Borates	Polyglycols	Silicones	Silicates
Viscosity at −65 F, cs	2,500	1,440	7,600	180 (ex.)	17,350	40,000 (ex.)	1,100 (ex.)	2,500
Viscosity at −40 F, cs	.....	264	1,420	59.1	1,781	3,500	600 (ex.)	781
Viscosity at 0 F, cs	.....	.....	213.0	.....	.....	270	290.0	.....
Viscosity at 100 F, cs	.....	4.84	17.5	2.93	6.6	11.5	74.0	29.0
Viscosity at 210 F, cs	.....	1.58	4.50	1.17	1.84	2.78	30.0	8.02
Viscosity at 400 F, cs	3.5	.....	.....	.....	.....	.....	11.0 (ex.)	.....
Vapor Pressure at 400 F, mm Hg	3	.....	.....	.....	.....	.....	.....	3.0
Hydrolytic Stability	Satisfactory	Satisfactory	Satisfactory	Unsatisfactory	Unsatisfactory	Satisfactory	Satisfactory	Satisfactory
Thermal Stability at 400 F	Satisfactory	Unsatisfactory	Satisfactory	Unsatisfactory	Unsatisfactory	Satisfactory	Satisfactory	Satisfactory
Oxidation and Corrosion (Inhibited)	Satisfactory	Unsatisfactory	Satisfactory	Unsatisfactory	Unsatisfactory	Satisfactory	Satisfactory	Satisfactory
Flash Point (min.), F	400	300	460	275	345	275	575	350
Pour Point, F	−65	−65	−65	−65	−70	−70	−65	−100
Wear – Steel on Steel	Satisfactory	Satisfactory	Satisfactory	Satisfactory	Satisfactory	Satisfactory	Unsatisfactory	Satisfactory

Table 3 – Effect of a Chlorofluoroheptene upon the Wear Properties of a Silicone as Shown by Shell Four-Ball Tests

Composition of Fluid Parts, by Weight		Average Wear Scar Diameter (Steel-on-Steel Bearing Surfaces), mm	
Silicone	Chlorofluoro-heptene	4.0-Kg Load	40.0-Kg Load
10	0	0.376	2.415
9	1	0.244	0.758
8	2	0.252	0.854
7	3	0.300	1.006
6	4	0.346	1.035
0	10	0.318	1.102

Table 4 – Viscosity Properties of Experimental Fluid

Temperature, F	Viscosity, cs
−65	2333
−40	544
100	13.8
210	4.55
400	1.75 (extrapolated)

temperature and viscosity-volatility characteristics entirely suitable as such for high-temperature hydraulic fluids. Several polymers of this class exhibit good thermal stability as well as resistance to shear and oxidation. Flash and pour points are well within the limits desirable. However, as stated before, our requirement is that the high-temperature hydraulic fluid must afford lubrication to steel-on-steel sliding and rolling surfaces under considerable load. The property is not present in the polysiloxane, unfortunately. Blending of polysiloxanes with other superior lubricants may result in a satisfactory material. Also, studies pioneered and conducted by the U. S. Air Force in high-temperature lubricating additives for polysiloxanes may precipitate a satisfactory high-temperature fluid. Table 3 exhibits the results of the addition of certain halogen-containing organics to polysiloxanes to enhance lubrication. These present halogenated materials are not satisfactory at high temperatures, due to poor volatility characteristics; however, further research in this field may result in additives for polysiloxanes suitable for use at high temperatures.

Halogen-Containing Organic Compounds – Materials included in this class include the chlorine-, bromine-, and fluorine-containing organic compounds. Mixtures of bromine, chlorine, and fluorine within the same organic structure are included. Generally, the chlorine-containing organics are toxic, exhibit poor stability, and are limited relative to high-temperature stability. Many of these deficiencies are true for the bromine-containing organics also. On the other hand, the fluorocarbons and, in many cases, organic compounds containing chlorine and fluorine, bromine and fluorine, or a combination of all three exhibit excellent resistance to chemical attack, oxidation, hydrolysis, and heat. Unfortunately, these fluorine-containing organic compounds possess extremely poor viscosity-volatility and viscosity-temperature characteristics, which would prohibit their satisfactory use in high-temperature hydraulic fluids, where a −65 F requirement also exists.

Generally, then, one would conclude that the organo-phosphorus compounds, silicates, polysiloxanes, and esters of carboxylic acids offer the most promising immediate solution to the high-temperature hydraulic fluid problem, that is, if the low-temperature requirement of satisfactory operation of hydraulic mechanisms at −65 F is to be considered. Naturally, if one were permitted to overlook this low-temperature requirement and concentrate solely on providing a satisfactory high-temperature and lubricating hydraulic oil, the problem could be solved quite simply by selecting the proper cut of crude oil. Also, forget the lubricating requirement and the polysiloxanes will provide the viscosity and thermal properties required of a −65 F to 500 F fluid.

Table 5 – Viscosity-Volatility Properties of Diesters

Compound Diabasic Acid Esters	Viscosity at 210 F, cs	100 F, cs	0 F, cs	−40 F, cs	−65 F, cs	ASTM Slope	Flash Point, F	Pour Point, F	Vapor Pressure at 400 F
Adipates									
Bis (3-methylbutyl)	1.70	4.87	.....	.....	.....	0.78	355	−80	.....
Bis (2 ethylhexyl)	2.354	8.166	108.3	818	4810	0.769	360	−75	3.3
Bis (C_8Oxo)	2.700	9.44	121.1	913	5450	0.725	380	−75	2.9
Azelates									
Bis (2 ethylbutyl)	2.412	7.764	79.2	467	2160	0.726	365	−75	4.3
Bis (2 ethylhexyl)	3.000	11.09	154.3	1202	6668	0.721	405	−75	1.2
Sebacates									
Bis (sec. Amyl)	2.271	7.483	81.5	528	2970	0.756	375	−75	6.4
Bis (ethylhexyl)	3.285	12.46	183.4	1386	7690	0.704	420	−75	0.78
Glycol center diester	2.367	8.075	.....	647	3405	.....	400	−75	3.8

Let us now discuss some of the development studies conducted thus far in the field of organo-phosphorus compounds, silicates, and diesters.

A comprehensive program was initiated by the U. S. Air Force research and development laboratories to investigate synthetics as high-temperature hydraulic fluids. These classes of organic materials were investigated extensively and may be divided into organo-phosphorus compounds, the carboxylates or as they are commonly called the esters, and the organo-silicon compounds. The base materials for high-temperature hydraulic fluids were selected on stability to oxidation and corrosion, viscosity, low volatility at elevated temperatures, and antiwear properties.

The organo-phosphorus compounds have been investigated as potential high-temperature hydraulic fluids chiefly because they have reduced flammability. Desirable properties such as lubricity and low-temperature coefficients of viscosity can be found in selected compounds. Base materials were prepared from the alkyl and aryl phosphates and phosphonates.

The alkyl phosphonates were the only members of this class of compounds that had the desirable viscosity properties and extended liquid range necessary for a high-temperature hydraulic fluid. The phosphonates were evaluated in the high-temperature oxidation and corrosion test at 347 F, and they were found to be corrosive. All known oxidation and hydrolysis inhibitors were tried in these materials and none functioned satisfactorily. A fluid was formulated for comparative purposes using bis (butyl) isooctane phosphate and a conventional polymethacrylate viscosity index improver. The viscosity properties of the fluid are given in Table 4.

Although the present studies in organo-phosphorus compounds have failed to produce a desirable high-temperature hydraulic fluid, further consideration should be given to these compounds when more effective inhibitors or more stable materials become available.

The carboxylates, or esters, have been extensively investigated under USAF contract as fluids and lubricants. For the purpose of this discussion only the diesters will be considered, since any monoester that had suitable viscosity properties for a high-temperature hydraulic fluid had low flash points and exceptionally high vapor pressures at 400 F.

Carboxylates of the diester class are considered to be promising for high-temperature hydraulic fluid applications, since these materials possess good stability and lubricating properties, have a wide liquid range, and low vapor pressures at 400 F. The viscosities at —65 F of the more conventional diesters range from 1000 for bis (3-methylbutyl) adipate to 7600 cs for bis (2-ethylhexyl) sebacate. The physical properties of a selected group of diesters used in the program is included in Table 5.

Table 6 – Properties of Diester-Base High-Temperature Fluids

	Diester Fluid A	Diester Fluid B	Diester Fluid C
Viscosity at			
210 F, cs	4.51	4.95	5.3
130 F, cs	10.1	11.1	....
100 F, cs	15.4	16.9	20.8
0 F, cs	179	193	374
—40 F, cs	1150	1220	2700
—65 F, cs	6300	6070	16,000
ASTM Slope			
210 F to —65 F	0.62	0.60	0.60 (210/100 F)
Flash Point, F	415	360	450
Pour Point, F	—75	—75	—75
Oxidation Corrosion	Satisfactory	Satisfactory	Satisfactory
Hydrolytic Stability	Satisfactory	Satisfactory	Satisfactory
Wear Characteristics	....	....	....
Shell Four-Ball Wear Tester			
800 Rpm, 1 Hr			
1 Kg	0.16	0.21	0.13
10 Kg	0.28	0.62	....
40 Kg	0.78	0.91	0.40

In the formulations of diesters for high-temperature use, polymethacrylates have been used as viscosity index improvers. Recent investigation has shown that polysiloxanes may also be used as thickeners, and fluids prepared from these compounds will have lower —65 F viscosities than fluids prepared with polymethacrylate thickeners. Test data are incomplete at the present time on diester polysiloxanes fluids.

Three diester fluids suitable for high-temperature hydraulic fluid applications have been formulated using polymethacrylate viscosity index improvers and organo-phosphate antiwear agents. The physical properties of these fluids are listed in Table 6. Fluids A and C have been extensively service tested in lubricant applications. Fluid A contains an ester of a secondary alcohol and is, therefore, less thermally stable than fluids B and C. The service use of fluid A will be discussed later in this paper. Fluid C was originally formulated as a gear and engine lubricant. Because of its excellent properties, this material should be given consideration as a potential high-temperature hydraulic fluid where —65 F viscosity requirements permit a material of this high viscosity to be used.

The orthosilicates are considered to be the most promising materials for the formulation of a high-temperature hydraulic fluid that will operate from —65 F and also low vapor pressures at 400 F. These compounds can be made hydrolytically stable by fabricating a steric-hindered molecule and adding a stabilizer. In this program a great variety of these compounds was synthesized and evaluated for oxidation corrosion resistance, hydrolytic stability, and lubricity. Silicate compounds of the proper configuration when stabilized were evaluated in the high-temperature oxidation corrosion test at 347 F. These materials displayed stability equal to that of the diesters.

Table 7 – Properties of Organo-Silicon Compounds and Blends of Viscosity Index Improvers

Base Fluid	Additive	Viscosity at −65 F, cs	−40 F, cs	100 F, cs	210 F, cs
C_6 silicate	.....	204	67	3.94	1.55
C_6 silicate	Polysiloxane A 6%	1466	...	25.4	9.38
C_6 silicate	Polysiloxane A 8%	2351	...	42.2	14.7
C_6 silicate	Polysiloxane B 14%	1682	...	32.7	11.8
C_8 silicate	.....	1320	316	6.95	2.26
C_8 silicate	Polysiloxane A 6%	6315	...	34.5	10.9
Mixed silicate	.....	472	136	5.02	1.82
Mixed silicate	Polymethacrylate	6845	...	27.9	9.52
Mixed silicate	Polysiloxane C 10%	2500	781	27.0	9.45

Extensive formulation studies were conducted utilizing stabilized alkyl silicates and adding polymeric thickening agents to obtain the proper viscosities. The polysiloxanes and the polymethacrylates both functioned satisfactorily as viscosity index improvers but the polysiloxane-thickened fluid had superior viscosities. Table 7 lists the viscosities of these alkyl silicates and blends of these materials with viscosity index improvers.

Although the silicates had good viscosity properties and stability, as hydraulic fluids they must adequately lubricate the pump and components. When the wear properties of the orthosilicates were measured on the Shell Four-Ball Wear Tester, it was found that these materials had lubricity comparable to that of a light mineral oil.

Three finished high-temperature hydraulic fluids have been formulated utilizing alkyl silicates and adding polysiloxanes as viscosity index improvers. The properties of these fluids are presented in Table 8.

The silicate-base fluid D was considered most nearly to meet the requirements for a high-temperature hydraulic fluid that will operate from −65 F to 400 F. This fluid is being evaluated by the U. S. Air Force and hydraulic system component manufacturers. A fluid improvement program will be guided by these evaluations to insure that this material will be a good high-temperature hydraulic fluid.

Table 8 – Properties of Silicate-Base High-Temperature Hydraulic Fluids

	Fluid D	Fluid E	Fluid F
Gravity, deg API	24.2	24.2	17.7
Density, −65 F/4 C	0.9572	0.958	1.1013
Density, 400 F/4 C	0.7624	0.759	0.7963
Flash, F	350	325	320
Fire, F	395	360	355
Viscosity at −65 F, cs	2504	1122	2197
Viscosity at −40 F, cs	780	400	695
Viscosity at −100 F, cs	27.0	29.8	34.5
Viscosity at −130 F, cs	19.5	22.6	25.0
Viscosity at 210 F, cs	9.45	12.47	12.66
Viscosity at 400 F, cs	3.3	4.82	4.63
Viscosity Index	185	1.81	177
ASTM Slope, 100 to 210 F	0.42	0.31	0.35
Pour Point, F	−100	−100	−80
Acid No.	0	0	0
Precipitation No.	0	0	0
Appearance	Clear	Clear	Clear
Color ASTM	4	4	4½
Vapor Pressure, mm Hg	3.0	11	3.1
Spontaneous Ignition Temperature, F	745	660	900
Preformance Tests			
Low-Temperature Stability	Slight haze	Haze	Haze
Hydrolytic Stability	Passes	Passes	Passes
Oxidation Corrosion Stability			
168 Hr at 250 F	Satisfactory	Satisfactory	Satisfactory
72 Hr at 347 F	Satisfactory	Satisfactory	Questionable
Shear Stability	Excellent	Good	Good

Evaluation

The actual operating conditions of high-temperature hydraulic systems, where the pressure is maintained at 3000 psi and temperature may reach 400 F or above, are severe tests for hydraulic fluids. Not only is a fluid required to operate in this environment but it must also remain stable and not form lacquers on precision parts or corrode the metal in the system. Besides this, the fluid is required to lubricate the high-speed hydraulic pumps. To ensure that a high-temperature hydraulic fluid meets these requirements, extensive chemical and physical evaluations are conducted by the U. S. Air Force

Chemical evaluation of a fluid consists of such tests as a high-temperature oxidation corrosion test at 347 F or hydrolytic stability tests to assure a stable fluid in storage as well as in use.

The physical evaluation of a fluid begins with laboratory wear tests. Pump wear and shear stability tests follow. A fluid that is satisfactory under these conditions is then tested in an aircraft mock-up to evaluate the effect of the hydraulic fluid on the overall hydraulic system.

Lubricity

The Shell four-ball wear tester has been employed in this program to measure the lubricity of various base materials and finished fluids. In this test higher wear values have been noted for the silicate fluid D than for diester fluids A, B, and C; however, the diester fluids have adequate lubricity to be used in lightly loaded gear and engine applications. Specification MIL-0-5606 petroleum-base aircraft hydraulic fluid was used as a reference fluid for wear data. In actual service usage a specification MIL-0-5606 type of fluid, without antiwear additive, was found to be a satisfactory lubricant for hydraulic systems. This type of material is similar to a light-petroleum-base lubricant. An antiwear additive was added to extend the use of these materials to propeller mechanisms. When measured on the Shell four-ball wear tester the silicate fluid D was found to have antiwear properties comparable to those of a light petroleum oil. At elevated temperatures the excellent viscosity properties of the fluid will also be an aid to lubrication. In this test the lubricity of the diester fluids A, B, and C were equal to that of a good petroleum lubricant. Table 9 lists the wear data of the diester fluids and the silicate fluids at temperatures from 167 F to 399 F. As a guide line, a high flash point

Table 9 – Wear Characteristics of Hydraulic Fluids at Elevated Temperatures

(Values given are average wear scar diameter (min) for steel-on-steel bearing surfaces)

Fluid No.	Description	1-Kg Load Test Temperature, F					10-Kg Load Test Temperature, F					40-Kg Load Test Temperature, F				
		167	266	300	351	399	167	266	300	351	399	167	266	300	351	399
1	Specification 51-F-21 Ships hydraulic fluid	0.15	0.16	0.15	0.15	0.14	0.22	0.22	0.25	0.23	0.23	0.44	0.54	0.70	0.97	0.84
4	Diester with antiwear additive	0.14	0.17	0.15	0.16	0.18	0.25	0.25	0.26	0.26	0.30	0.38	0.80	0.91	0.97	0.97
3	Silicate-base Fluid D	0.37	0.24	0.34	0.34	0.34	0.61	0.61	0.67	0.58	0.53	0.65	1.17	1.04	1.17	1.04
5	Diester fluid A	0.18	0.16	0.16	0.18	0.20	0.28	0.44	0.59	0.50	0.58	0.91	0.91	0.97	0.84	0.74
2	Petroleum fluid without antiwear additive	0.35	0.44	0.38	0.41	0.33	0.56	0.65	0.56	0.57	0.51	0.66	0.76	0.84	0.93	0.91

Navy Ordnance hydraulic fluid, Specification 51-F-21 Ships, was used in place of the more volatile Specification MIL-0-5606 aircraft hydraulic fluid.

In pump wear testing, the high-wear parts of the pump are weighed before and after each test to determine weight loss. Reference fluids are used to establish the maximum wear.

Both the silicate fluid and the diester fluids A, B, and C have exhibited excellent lubricity in gear-type hydraulic pumps. Silicate fluid D was evaluated in a similar bench test using piston pumps. Satisfactory lubrication was obtained at temperatures up to 160 F. The test temperature was then increased to 250 F, 300 F, 350 F, and 400 F. In each case, pump failure occurred before the maximum 500 hr. The time for failure decreased as the temperature was increased. At 400 F the time for failure was as low as 1.1 hr. Detailed examination of these pumps revealed searing of the valve plate, fatigue or excessive wear of the piston knuckles and other parts of the pump. In each case scoring and excessive wear were noted on the thrust shoulder of the main shaft, due to bearing slippage at elevated temperatures. This caused fine metal particles to be distributed throughout the pump, causing excessive wear and fatigue. It soon became evident that it was impossible to evaluate pump wear at elevated temperatures until a suitable high-temperature pump became available. To prove that it was mechanical failure and not a lack of lubricity that caused this condition, Specification MIL-0-6081, Grade 1120 engine oil was tested in a piston pump at 400 F. This oil has approximately the same viscosity at 400 F as silicate fluid D. Pump failure occurred in the same pattern as was previously noted. The total running time of the test was less than 30 hr.

The diester fluids have not been tested in bench wear tests using piston pumps. However, diester fluid A was formulated to be used in alternator-drive assemblies, and in this application the fluid acts as a hydraulic fluid as well as a lubricant. In service use this fluid has operated for 500 hr at an average temperature of 250 F without change. Diester fluid B is considered interchangeable with fluid A; however, fluid A contains an ester of a secondary alcohol and therefore will be less thermally stable than diester fluids B and C.

Shear Stability

When a hydraulic fluid maintained at high pressures is unloaded through a small orifice, shearing or cutting of the high-molecular-weight molecules occurs. This results in a decrease in viscosity. In service if a hydraulic fluid shears excessively, the efficiency of the hydraulic system will be affected. It is necessary, therefore, to ensure that a hydraulic fluid is resistant to shear. The silicate fluid and the diester fluids have both exhibited excellent shear stability.

Viscosity and Pump Efficiency

The minimum design value for the viscosity of a high-temperature hydraulic fluid is considered to be 3.0 cs at 400 F. A test was conducted to evaluate the pump efficiency at various low viscosities. Specification MIL-0-5606 hydraulic fluid was used and two other fluids of lower viscosities were formulated. The viscosities of the test fluids ranged from 14.4 cs at 100 F for Specification MIL-0-5606 hydraulic fluid to 1.06 cs at 200 F for fluid J. The test data are summarized in Table 10. This test indicates that a viscosity as low as 1.5 could be tolerated without affecting pump efficiency. By lowering the minimum viscosity to 1.5 cs, diester-base fluids could be used at temperature as low as —40 F, with a resulting viscosity of 2500 cs.

Mockup Testing

An aircraft hydraulic system mockup unit utilizing the important parts of an aircraft hydraulic system was fabricated in order to evaluate the effect of a hydraulic fluid on the overall hydraulic system. A schematic diagram of this mockup is presented in Fig. 2.

The silicate-base fluid D has been tested in this

Table 10 – Pump Performance Studies with Low Viscosity Hydraulic Fluids

	Specification MIL-D-5606 Fluid	Fluid G	Fluid H	Fluid I	Fluid J
Viscosity at 100 F, cs	14.4	3.27	....	....	....
Viscosity at Test Temperature, cs	14.4	3.27	2.18	1.40	1.06
Pump Speed, rpm	3600	3600	3600	3600	3600
Pump Inlet Temperature, F	100	100	100	150	200
System Pressure, psi	2900	3000	3000	2950	2750
Test Time, hr	52	24	24	11	1
Average Flow Rate, gpm	3.10	3.04	3.00	2.89	2.76

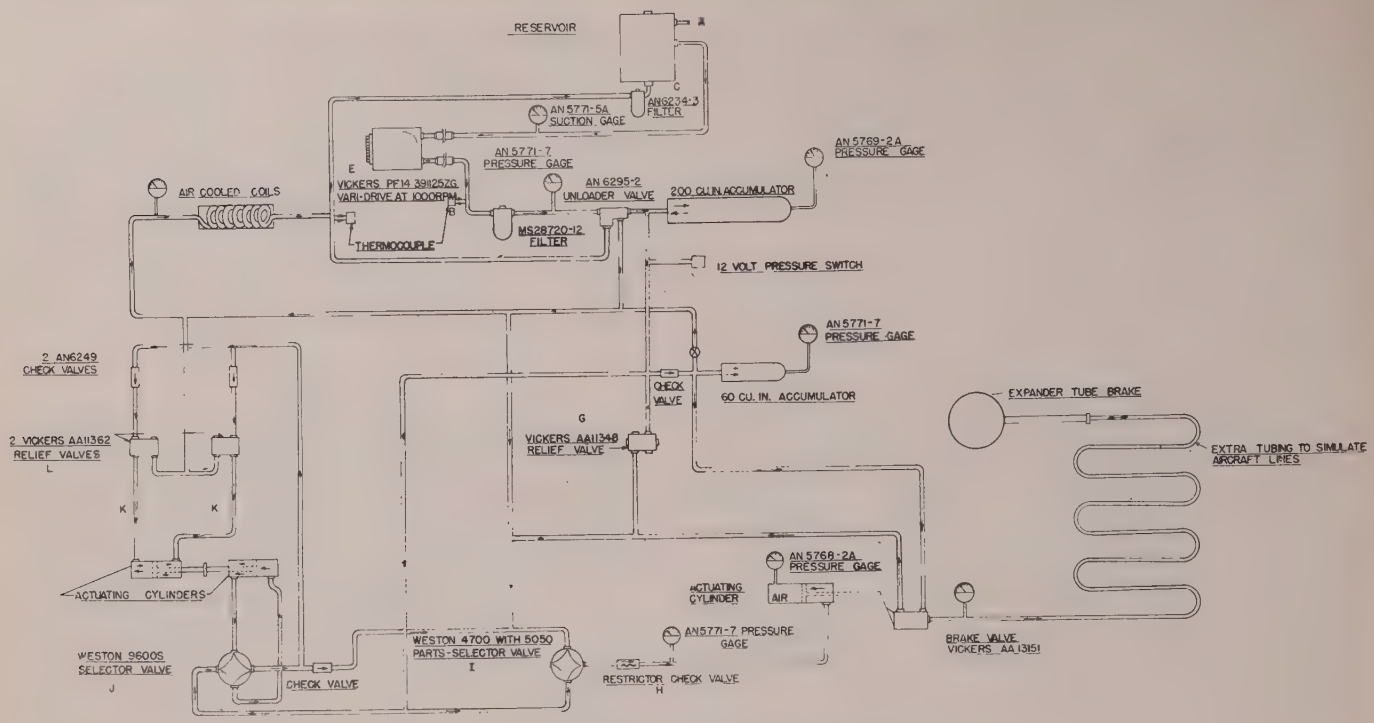


Fig. 2 – Schematic diagram of hydraulic fluid test stand mockup

mockup unit at temperatures of —65 F to 300 F. A pressure of 3000 psi was maintained. When this fluid was cold soaked at —65 F for 96 hr, thickening of the fluid was noted. This was traced to an inferior viscosity index improver, and has been corrected. At room temperatures and at 160 F normal operation was experienced. Above this temperature pump breakdowns occurred and, upon examination, the pumps showed the same type of failure previously noted in pump wear tests. The 300 F run was continued for 89 hr. No lacquering or corrosion of the metal parts was noted and the lap-fit valves used in the assembly operated with ease.

The evaluation studies have indicated that the flash point of silicate-base fluid D should be raised to a temperature above 400 F. At the present time a fluid improvement program is being conducted on the silicate-base high-temperature fluid to ensure that a hydraulic fluid used by the USAF for high-temperature applications will function satisfactorily and give adequate component life. Mockup testing of the diester fluids will be conducted up to temperatures of 300 F, the limit of the present equipment. Of the four fluids considered in the evaluation program, only silicate fluid D will meet the viscosity requirements of 2500 cs at —65 F. The viscosity-temperature relationships of the four finish fluids are compared in Fig. 3. Although the diester fluids have higher viscosity values at —65 F than silicate fluid D, it is safe to say that diester fluids, such as fluids A, B, and C, will play an important part in the high-temperature hydraulic fluid applications.

In view of the information presented, encouraging or discouraging as the case may be, one may ask, "What about the future?" Many times the questions, "What are the limits of temperature for hydraulic systems?" and "What is the possibility of a fluid to meet such requirements?" are presented. These two questions cannot be separated, since any fluid development of an organic nature will be limited to the bond strength of the molecules contained in the hydraulic fluid, thereby establishing a high-temperature limit for an organic hydraulic fluid at about 850 F, or possibly 1000 F as an absolute maximum. Therefore, if hydraulic systems can be developed for temperatures exceeding 850 F, the possibilities of providing a fluid for them are highly improbable. However, since we are concerned, at this time, with developing a satisfactory fluid for the 400 F to 600 F range, one may predict, on the basis of recent research studies, that silicates can be produced that have higher flash points and greater stability than those discussed in this paper; hence, these will be of increasing importance for high-temperature fluids of the future. In like respect, the continued efforts by the USAF and industry to improve the lubricity and thermal stability of the

polysiloxanes will undoubtedly produce materials useful in the high-temperature program. In addition, the more unique synthetics, such as polynuclear aromatics, organometalloidal, and organometallic compounds now in the test tubes of the Air Force scientists will also play a part in future high-temperature hydraulic fluid development.

To summarize, one may safely say that the U. S. Air Force, aided by industry, will provide a fluid for high-temperature hydraulic systems that will be a satisfactory lubricant, thermally stable from 400 F to 600 F, and produced from readily available materials.

Bibliography

1. Standard Oil Development Co., Fourth Quarterly Report, July, 1951, Contract AF 33(038)-14593.

2. "Dibasic Acid Esters, an Ester Type for Fluids and Lubricants," Pennsylvania State College Report, PRL 3.49 Jan 51, Contract Nord 7958 (Task B).

3. "Fluids, Lubricants, and Related Materials," Pennsylvania State College Report, PRL 5.3 1953, Contract AF 33(038)18193.

4. "Fluids, Lubricants, and Related Materials," Pennsylvania State College Report, PRL 5.3 June 52, Contract AF 33(038)18193.

5. "Fluids, Lubricants, and Related Materials," Pennsylvania State College PRL Final Report, September, 1953, Contract AF 33(038)18193.

6. "Development of High-Temperature Hydraulic Fluids," California Research Corp., Contract AF 33(038)9831, Final Report dated January, 1951, Quarterly Reports: January, 1952, April, 1952, June, 1952, and September, 1952.

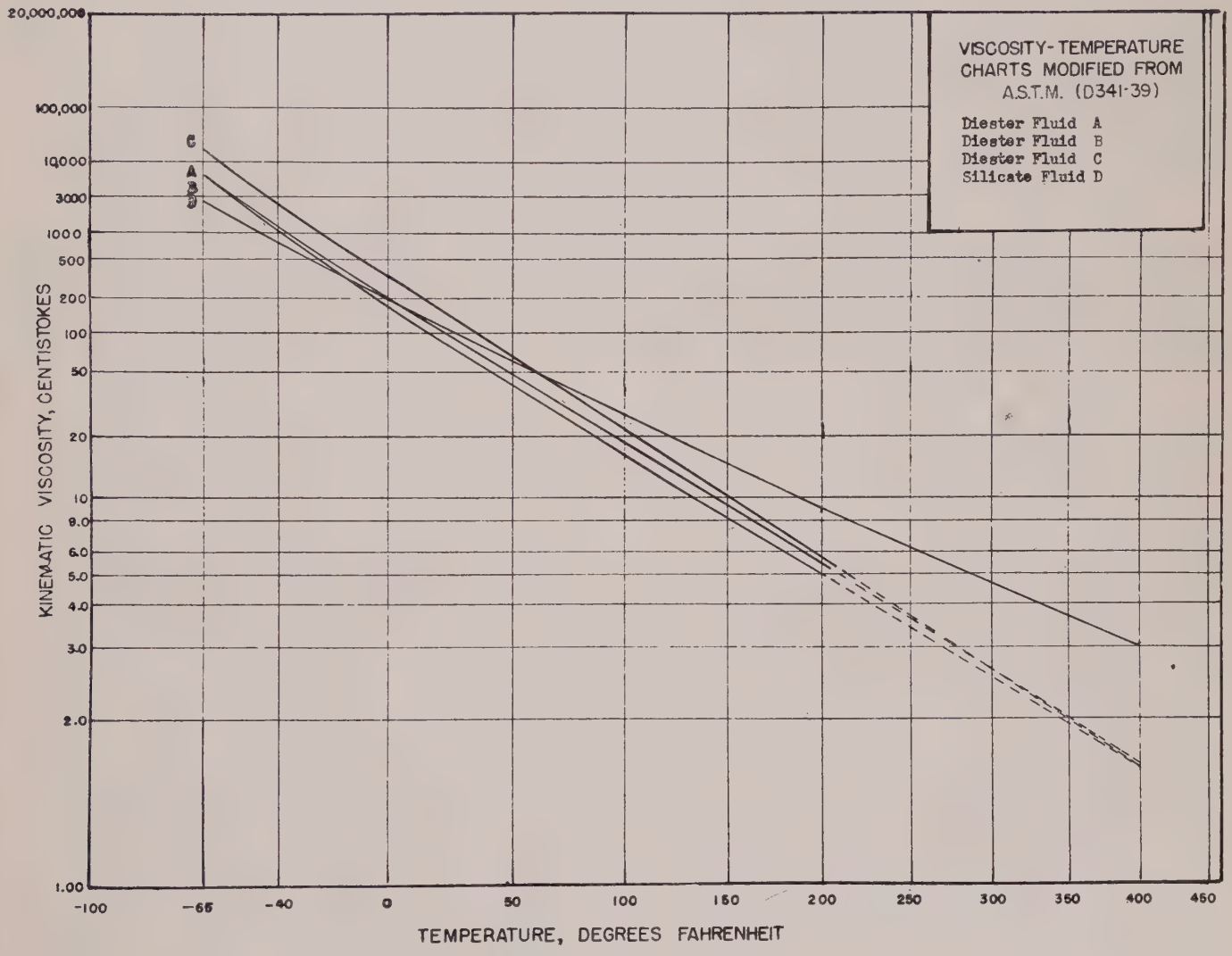


Fig. 3 – Viscosity-temperature characteristics of high-temperature hydraulic fluids

OVERHEAD VALVE

THE inherent performance advantages of overhead-valve engines have accentuated trends towards higher speeds. The valve gear, on the other hand, is much more sensitive to speed than it is in L-head engines, mainly due to the greater flexibility and weight of the parts. The endurance characteristics of the tappet faces are also more critical. A discussion of this problem and its relationship to cam and valve-gear design is the subject of this paper.

Valve Dynamics

Since many of the problems encountered are a function of the dynamics of the system at high speed a brief study from this point of view is in order. Let us consider the valve train as a vibratory system with a single degree of freedom as shown in Fig. 1. The flexibility in the push-rod and rocker arm can be represented by a spring having a constant K, and the entire mass of the gear is grouped together at M. Although this is not identical to the actual system it is close enough for comparative purposes since most of the mass is concentrated at the ends of the system – the valve and tappet and the major portion of the flexibility exists in the push-rod and rocker arm.

The lift and accelerating curves of a typical cam are also plotted in Fig. 1. An analysis of the shape of the cross-hatched portion of the acceleration curve and its effect upon the vibratory conditions will now be made. It would be desirable to consider the entire acceleration curve since we are

EFFECT of high speeds on valve gear, and the relationship of wear problems to design, is the subject of this paper. As automobile speeds increase, sensitivity of overhead valve parts increases, due to their flexibility and weight. Stiffness requirements and cam design are considered in relation to the dynamics of the system at high speed.

Another critical factor is encountered in the wear of tappet faces. The author reviews the large number of variables that play a part in tappet wear. He cites definite steps which have been taken to improve the situation, particularly in regard to the alignment of tappets. He states that the self-aligning tappet offers a new and encouraging approach to this problem.

Discussion of this and other papers on "Valve-Gear Problems in Modern Overhead-Valve Engines" starts on page 714.

The Author

RALPH P. HORAN (J '46) is chief project engineer at Eaton Mfg. Co. He took his mechanical engineering degree at the College of the City of New York, and his automotive engineering degree at Chrysler Institute. He then worked two years at Chrysler on engine development from the performance standpoint. Following a brief period with the Carbide and Carbon Chemicals Co. at Oak Ridge, he joined Hudson Motor Car Co., leaving there in 1948 for his present position.

GEAR PROBLEMS

Ralph P. Horan, Eaton Mfg. Co.

Paper presented at SAE Nat'l Passenger-Car, Body, and Materials Mtg., Detroit, March 5, 1953.

most interested in the vibratory conditions near valve closing. However, this calculation would be exceedingly complex and laborious, and would yield the same conclusions.

In general, transient vibrations excited in the linkage are excited during the accelerating portions of the lift, these being extinguished by the seating of the valve, so that while the cycle is recurrent, it cannot be considered a steady-state vibration but must be analyzed as a transient system. The vibrations excited by the first accelerating period will largely endure to the second since the system is not highly damped, and may be reduced or reinforced by the second, depending upon its phase as it reaches the second period. But since the phase changes with engine speed, reinforcement is unavoidable at intervals throughout the speed range, which doubles the vibration excited by the first, so that in estimating the deflection at the end of the lift cycle, which is responsible for most of our problems, we may be justified in analyzing the deflection at the end of the first accelerating period.

Let us proceed to the analysis of the deflection at the end of the first period of acceleration for a group of typical acceleration curves. These are all of equal area and would all produce equal velocities at the end of the period, thus having little effect upon the area under the lift curve. They were calculated from a rigorous solution of the differential equations of motion of the system. A sample derivation of the formulas used is given in the appendix.

Fig. 2 shows the vibratory response to a gradually increasing acceleration with an abrupt drop. The cam speed is plotted in terms of critical cam speed which is defined as the speed at which one cycle of gear vibration takes place during the accelerating cycle. It will be noted that the amplitude remains fairly high even at lower speeds.

A relative amplitude of vibration of 1, as shown on these curves, is fairly high and is more than enough to cause valve float. When calculated for a typical example this amplitude was three times greater than that required to obtain gear separation.

Fig. 3 indicates that the sharp triangular acceleration shown results in excessively high vibrations. These first two comparisons clearly indicate that very sharp changes in acceleration should be avoided.

In Figs. 4 through 7 curve shapes are indicated which are much closer to actual design practice. Comparison of the response of the abruptly changing accelerations of Figs. 4 and 6, with the smooth parabolic acceleration of Fig. 7, shows the distinct advantage of a smooth curve. The result in Fig. 5 which employs one cycle of a sine curve is a little surprising since this most certainly would be classified as a smooth curve. The least amount of vibration is obtained with the parabolic contour of Fig. 7. It is conceivable that other shapes might give even better characteristics but the main value of this comparison is in the clarification of the dynamic cam requirements.

The critical cam speed as defined in this analysis depends upon two factors. These are: length of the acceleration cycle and the natural vibratory frequency of the valve gear. The increase in vibratory amplitude is so rapid beyond the critical

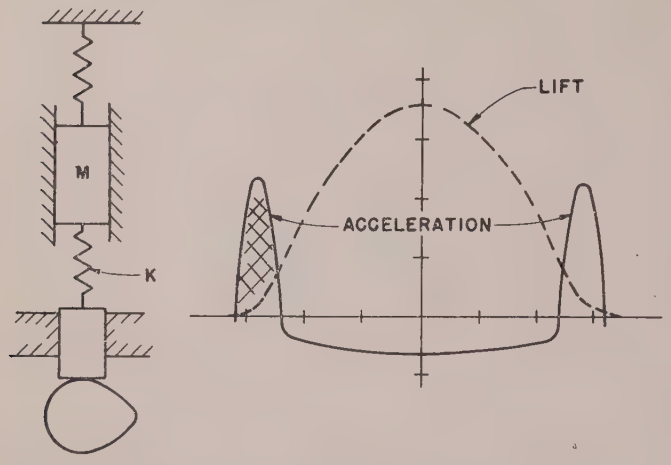


Fig. 1 – Valve train shown as vibratory system with single degree of freedom, together with lift and accelerating curves of typical cam

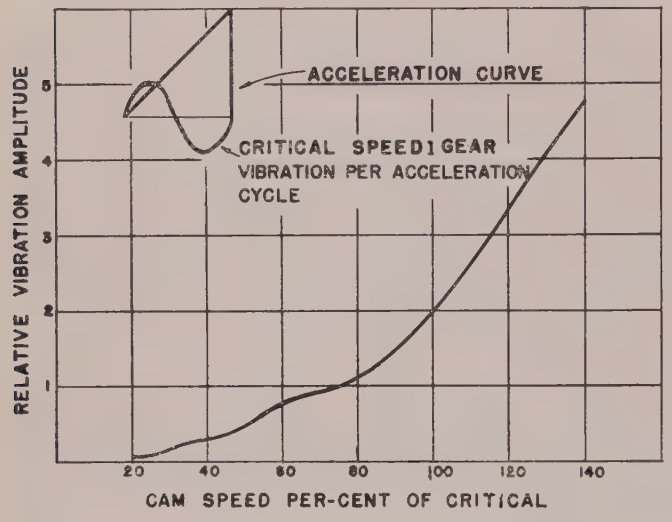


Fig. 2 – Vibratory response of valve gear to gradually increasing acceleration with abrupt drop

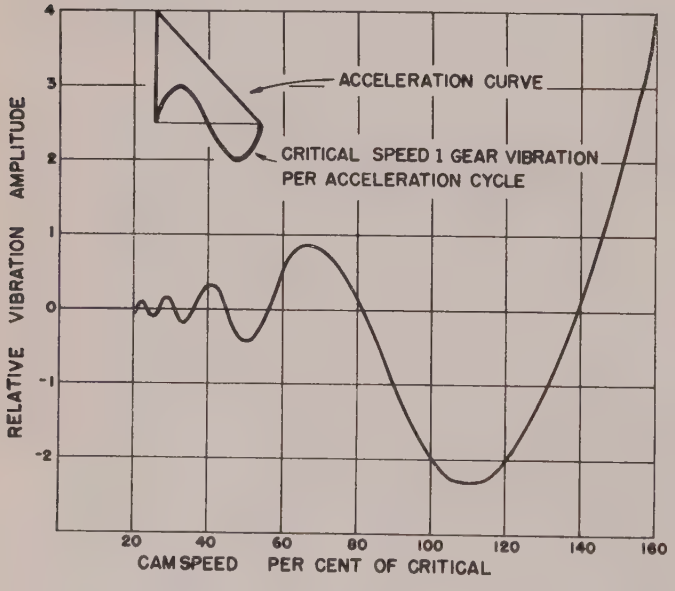


Fig. 3 – Sharp triangular acceleration results in excessively high vibrations in valve gear

speed that it is obviously a limiting condition. The longer the acceleration cycle and the higher the natural frequency, the higher will be the critical speed of the cam. The design of the cam then determines how close the gear can be operated to the critical speed and the design of the remainder of the valve gear determines how high the critical speed will be. The natural vibratory frequency of the valve gear, as in other vibratory systems, is directly proportional to the square root of the stiffness, and inversely proportional to the square root of the mass. It is, therefore, apparent that in order to keep the vibration-frequency high the mass of the gear should be kept low and the stiffness high.

Design of valve gear components having a high stiffness to mass ratio should consider several factors: beefing up the rocker brackets increases stiffness without increasing mass; a greater camshaft diameter increases stiffness faster than mass, as does a greater rocker-arm section.

Shorter push-rods obviously increase stiffness while reducing mass and therefore are highly desirable, other conditions remaining equal.

The cross-section of the push-rod should be selected judiciously since the stiffness and mass increase in direct proportion. An optimum value exists for the stiffness/mass ratio of the gear when:

$$K_p M_p = K_1 M_1$$

where K_1 and M_1 represent the stiffness and mass of the entire system exclusive of the push-rod, and K_p and M_p represent the stiffness and mass of the push-rod.

In general then, the stiffness/mass ratio and thus the natural frequency of the valve gear can be raised almost indefinitely.

There are a number of manifestations of false valve-gear motion at high engine speeds. When mechanical tappets are being used, and as the severity of vibration increases at higher speeds, a point is reached where high-velocity seating on the flank of the cam can occur, resulting in possible valve failure. If proper attention has been paid to the various dynamic considerations, this condition will not take place within the operating range of the engine. When hydraulic lifters are being used any momentary gear separations at the lifter socket can result in pump-up (a gain in lifter length each cycle), since it is the normal function of a hydraulic lifter to compensate for lash. A hydraulic lifter can also be designed to lose lift during the separation cycle. The methods of doing this are too detailed to be included within the scope of this paper. If the loss in lift is sufficient it would cause premature high-velocity valve seating. This condition has been called valve clatter, and if serious enough could cause valve failure. Very small changes in hydraulic-lifter design will make appreciable changes in pump-up and clatter tendencies.

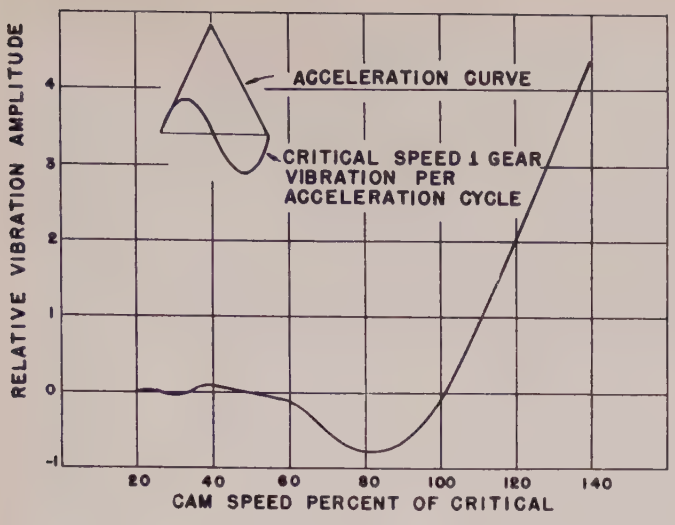


Fig. 4 – Vibratory response results in curve shape which is close to actual design practice

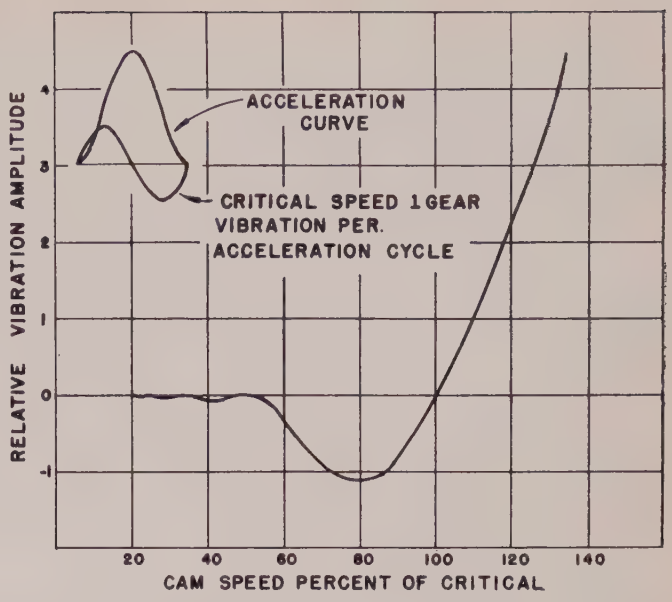


Fig. 5 – Vibratory response gives result employing one cycle of sine curve

It has been found that an increase in the spring load with the valve closed, and the use of spring dampers of various types, permits satisfactory engine operation at higher speeds. The spring load, however, cannot be increased without regard to other considerations, since tappet-face wear is influenced, as will be shown later.

Other Cam Considerations

The dynamic considerations are by no means the only factors that require attention in a valve-gear design. The design of a cam must also be balanced carefully against the performance requirements of the engine. It is desirable to obtain as high a valve lift as possible, with a minimum restriction to flow. The methods of achieving these conditions are not new, but a review of them will show that they must be considered together with the dynamic problems if a good design is to be expected.

There are two ways of achieving lift: the first is by spreading the opening and closing events, and the second by high acceleration of the valve. The desirability of keeping the acceleration down in order to avoid various high-speed problems is obvious from the previous discussion. While it is advantageous to spread the events as far as pos-

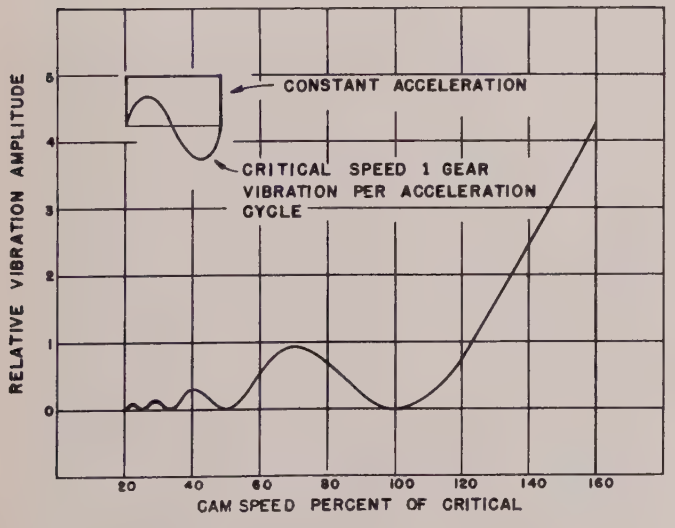


Fig. 6 – Response resulting from abruptly changing accelerations

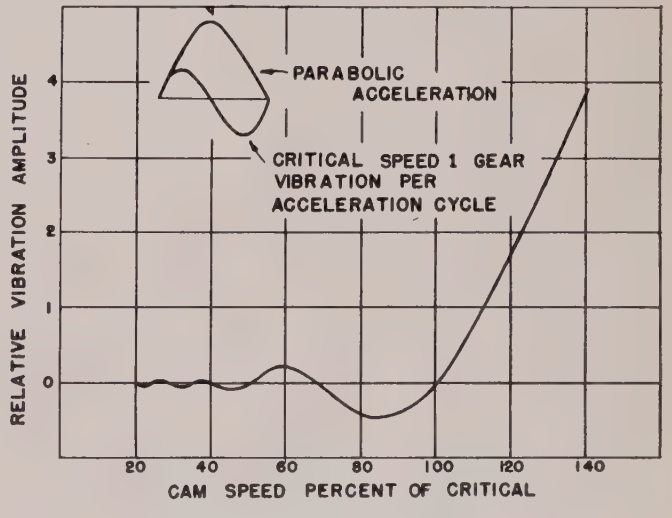


Fig. 7 – Distinct advantage of smooth curve obtained as result of least amount of vibration

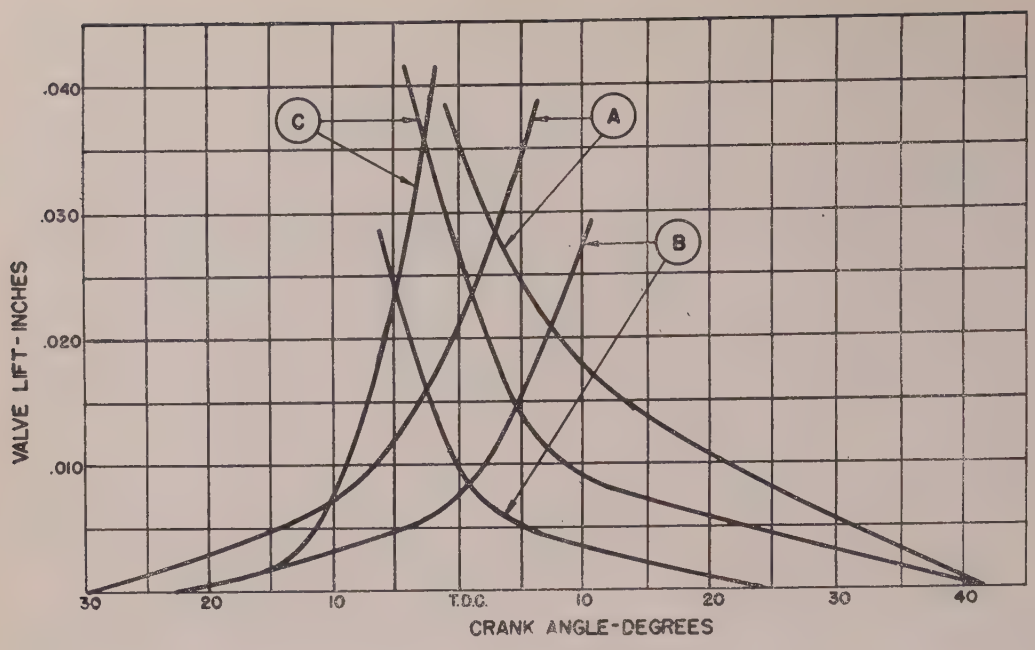


Fig. 8 – Valve lift plotted against crank angle for three different conditions

sible because of mechanical considerations, the factor which usually dictates how far the event can be spread is the requirement for smooth, low-speed idle performance. The events which have the greatest influence on this condition are the intake opening and exhaust closing. The overlap of these events should not be thought of in terms of degrees. Valve lift against crank angle is plotted in Fig. 8 for three different conditions. The lifts marked A have an overlap of 72 deg, whereas C is 64 deg, but both give high manifold vacuum which, incidentally, is a good criterion for idle performance. The curves B give smooth operation but offer the opportunity of spreading the timing events without adverse results. Fig. 9 more clearly illustrates this possibility. Here both curves A and B give excellent idle characteristics, although there is a considerable difference in the duration of the events. The event has to be spread to that shown in C before the manifold vacuum drops appreciably and results in unsteady operation.

Before leaving the question of overlap one other factor which has a considerable influence on overlap tolerance, but is not directly connected with valve events, should be mentioned. If adjacent cylinders of an engine fire in such a way as to cause one cylinder to be on overlap while the other is exhausting, there is a possibility of flow into, rather than out of, the first cylinder. This will result in poorer performance, but by providing a longer path for the gases by using individual exhaust manifold branches for these cylinders, the interference can be improved without sacrificing desirable characteristics in cam design. After all the various factors have been considered in the design of the shaft, a careful check of the actual shaft is advisable. A good method of doing this is given in the references listed at the end of this paper.

A valuable index as to whether the performance of an engine is in keeping with the potentialities offered by the valve sizes is shown in Fig. 10. This alignment chart is used to determine the intake and exhaust throat diameters (cylinder-seat small diameter) by drawing a straight line between the displacement per cylinder and the desired peak power speed. The chart is based on considerable engine data and has been proved to be fairly accurate. If the engine does not develop the power-peaking speed indicated by the throat diameters, the desired output can usually be achieved by bringing the other factors influencing performance to their optimum values.

Tappet Face Problems

A large number of variables have a considerable influence on whether or not tappet-face failures will occur. The factors which have demonstrated their effect upon the life of the mating parts include such items as tappet-face stresses, alignment, material combinations, finish, manufacturing techniques, surface coatings, and lubricants. Since all of these variables are effective to a greater or lesser degree in each design the problem of isolating each factor and determining its influence on tappet life is exceedingly difficult. Under the best conditions certain trends are indicated which aid in our understanding of the problem. Let us consider each variable separately and determine if possible its effect.

In Fig. 11 the cross-hatched area represents the range of tappet-face stresses encountered. The range is rather large for L-head and overhead-

valve engines as compared to a test fixture in which every effort has been made to control all other variables, especially those of alignment and finish. Difficulties have been experienced in a number of overhead-valve engine tappets operating in the lower portion of the stress range. The fact that the stress potential indicated by the fixture tests is not achieved for a given material combination is a sign that other variables are playing an important part.

Fig. 12 shows that stresses in an engine installation approaching those in the fixture are possible without failure, and that stress plays a definite role. No failures occurred at the lower stress values as is shown in the bottom row of photographs, and an increasing number of failures occurred at the higher stresses, examples of which are pictured in the top row.

The tappet alignment conditions in the engine have a considerable influence on tappet-face stresses. In Fig. 13 we see that when a flat face is used on the tappet and the alignment with the cam is perfect under all operating conditions, the contact is across the entire width of the cam, resulting in low stresses. If the tappet bore or the camshaft are not perpendicular, due to manufacturing tolerances or the deflection of the parts, edge bearing is possible, causing a very high stress. One method of overcoming this problem is to use a spherical-face tappet. This results in a stress which is higher than in case A, but less than with edge bearing. When a spherical face is used it has been found advantageous to taper and offset the centerlines to insure tappet rotation. This permits the tappet to wear in smoothly and better adjust itself to the operating conditions. The engine, however, is the final judge as to whether rotation so induced is desirable. If it is possible to exercise sufficient care in the manufacture of the cam, its bearings, and the tappet bore, a flat face can be used. In one case field failures were a considerable problem, and the stresses for a spherical face were found to be too high for a satisfactory life. After exercising very careful production control over the factors affecting alignment, the problem was eliminated.

One entirely new method of approaching this problem of alignment has recently been conceived. This is the so-called hyperbolic tappet, as shown in Fig. 14. In this case, the side of the tappet is contoured so as to permit the tappet face to accommodate itself to any misalignment which may occur. This design combines the advantages of the flat face with its lower stress and alignment. The laboratory results with this tappet have been very encouraging with a number of different engine designs. In several cases where failures occurred with the standard tappet, the hyperbolic tappets did not fail at spring loads 50% higher than standard. One manufacturer who had a very critical tappet failure problem obtained excellent results with the self-aligning tappet. Where 100% failures

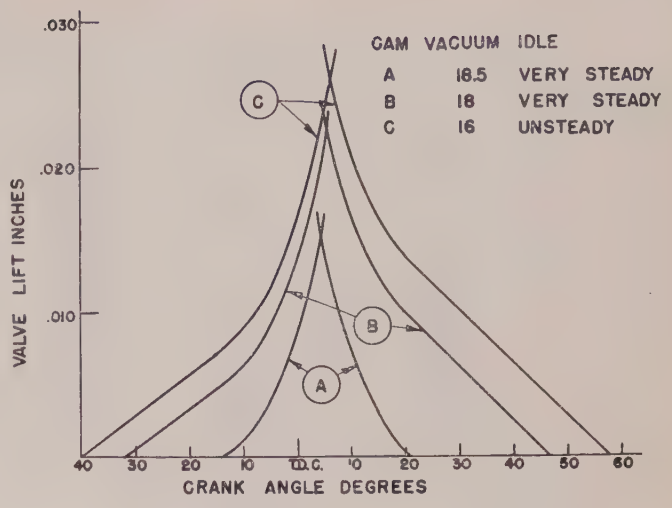


Fig. 9 – Curves A and B, showing timing of intake opening and exhaust closing, give excellent idle characteristics; in C, duration of event leads to manifold vacuum drop which results in unsteady operation

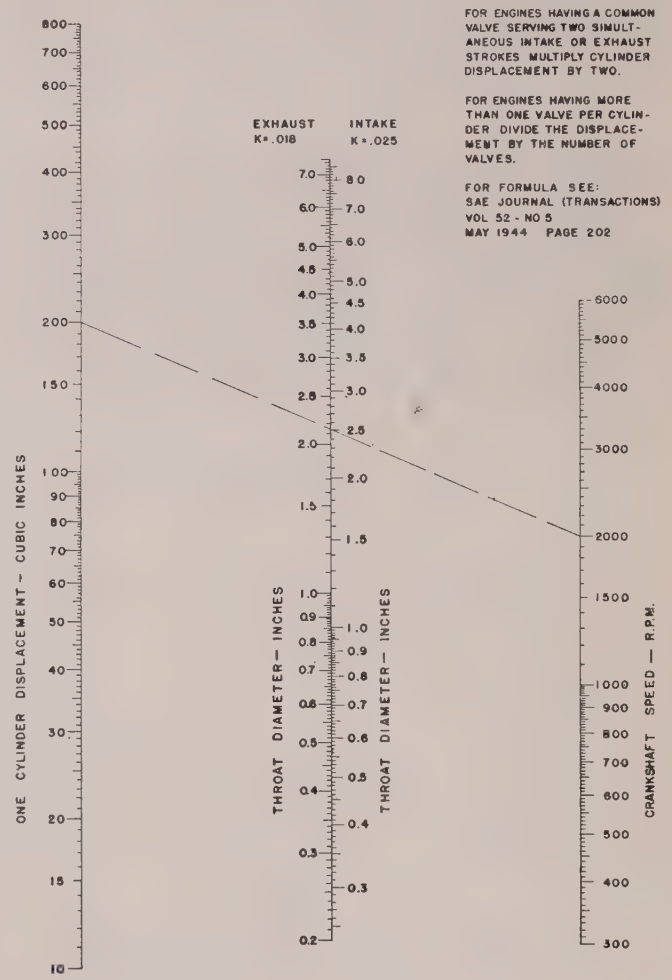


Fig. 10 – Index shows whether engine performance is in keeping with potentialities offered by valve sizes

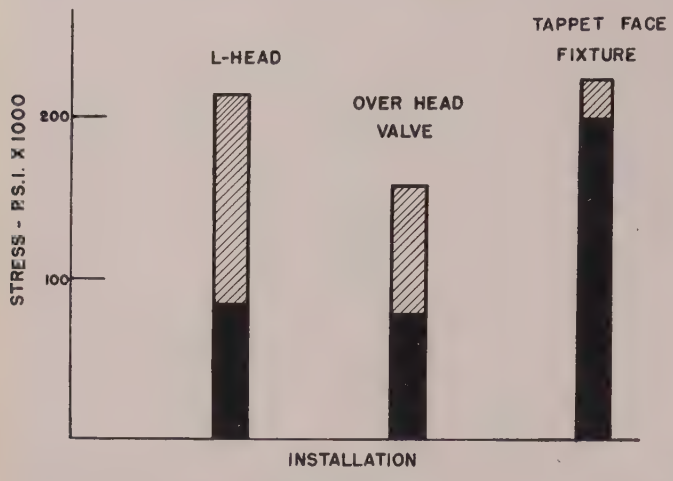


Fig. 11 – Cross-hatched areas represent range of tappet-face stresses found in comparing engine installations with test fixture

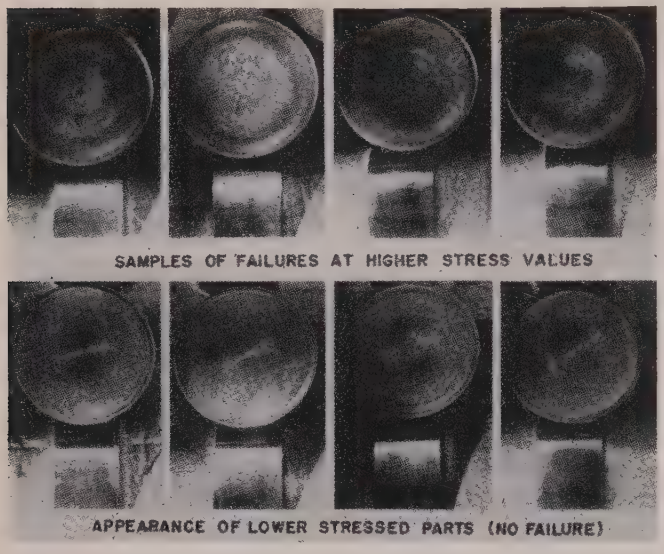


Fig. 12 – Effect of tappet-face stresses at higher and lower values. Theoretical face stresses at 187,000 psi showed no failures in 8 tests; at 195,000 psi there were 4 failures out of 8 tests; at 216,000 psi there were 6 failures out of 8 tests

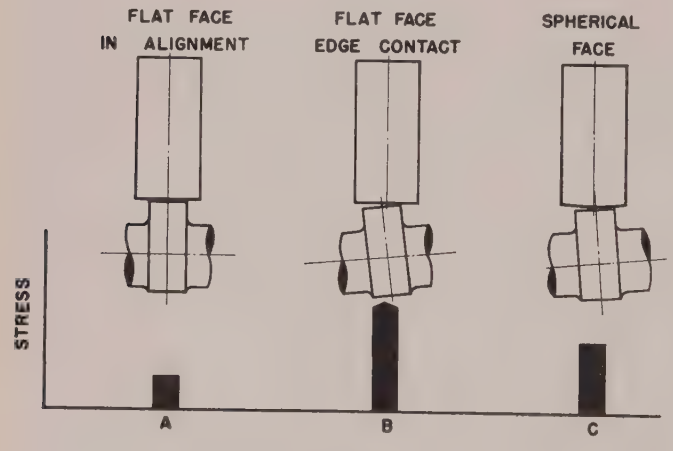


Fig. 13 – Tappet alignment conditions in engine effect tappet-face stresses

were previously experienced, the hyperbolic tappet ran indefinitely with exactly the same material combinations. Hyperbolic tappets have not given the expected improvement in all cases where they are tried. There is good reason to suspect that in these cases alignment was not the critical variable.

Camshafts are usually made from either hardenable grey iron or carburized steel with alloy additions to suit a particular installation. Tappet face materials of six types have been used:

1. Hardened high-carbon steel.
2. Carburized low-alloy steel.
3. Chilled cast iron.
4. Puddled chilled cast iron.
5. Mottled hardenable iron.
6. Grey hardenable iron.

Tungsten carbide has also been used for a tappet face where a very severe face-wear problem existed. It has excellent wearing properties, but it should be avoided if possible because of its high cost.

Experience has indicated that the best results are obtained with a chilled cast-iron tappet face when run against a steel camshaft. When a cast-iron shaft is used, equally successful applications occur with cast-iron or steel tappet faces. In practice many installations have operated successfully with various combinations of the other materials listed.

In the manufacture of tappets many variations can occur. Every effort has been extended in the past to control the techniques employed, while keeping the costs within reason. As more and more field experience is gained, the influence of any other important variables should be determined.

The coatings commonly used on the tappet faces are ferrox and parco-lubrite. The ferrox treatment cannot be applied to hardenable irons and steels because of the temperature of application. In general, the ferrox treatment has proved superior to lubrited surfaces for the chilled tappet face. Other surface treatments have been tried with some promise, but results to date are not extensive enough to permit any definite conclusions to be drawn. Some differences in the life of the tappet have been experienced by variations in the lubricants used. One factor which has demonstrated some influence is the "film strength" of the oil.

This discussion of tappet-face problems is intended to convey some idea as to the large number of factors which play a part in tappet wear, and to indicate that definite steps are being taken to improve the situation. As the more important variables are isolated and their influence determined, the tappet load-carrying capacity should increase.

It is hoped that this paper has shed some light on overhead valve gear problems, and has shown that the interrelationship of dynamics, performance, and endurance factors, must be considered to achieve a well-balanced design.

APPENDIX

Typical Derivation

Assumptions:

1. Opening deflection neglected.
2. Negative acceleration neglected.
3. Damping neglected.
4. Single degree of freedom.
5. Equal angles of acceleration for all cam designs.
6. Equal velocity for all cams at end of accelerating period.

Symbols

A_c = Acceleration at cam
A_p = Peak acceleration of parabolic acceleration curve
V = Velocity
S_c = Displacement at cam
S_m = Displacement of mass
$S_c - S_m = \delta$ = Deflection of valve gear
t = Time
P_a = Period for acceleration = $2t_a$
t_a = Time for ½ acceleration period
W = Frequency of vibration
K = Spring constant of gear
R = Cycles of gear vibration per accelerating period

$$W^2 = \frac{K}{M}$$

M = Mass
C = Cam

Subscripts:

m = Of mass
c = At cam

Case 1 – Parabolic Acceleration Curve (See Fig. 15)

$$(A_c - A_p) = -C\,(t - t_a)^2 \tag{1}$$

when:

$$t = 0,\ A_c = 0$$
$$-A_p = -Ct_a^2$$
$$C = \frac{A_p}{t_a^2}$$
$$A_c = A_p\left[1 - \frac{(t - t_a)^2}{t_a^2}\right]$$

or:

$$A_c = A_p\left[-\frac{t^2 + 2t_a t}{t_a^2}\right] \tag{2}$$

$$V_c = \int_0^t \frac{A_p}{t_a^2}\left[-t^2 + 2t_a t\right]dt = \frac{A_p}{t_a^2}\left[-\frac{t^3}{3} + t^2 t_a\right] \tag{3}$$

$$S_c = \int_0^t \frac{A_p}{t_a^2}\left[-\frac{t^3}{3} + t^2 t_a\right]dt = \frac{A_p}{t_a^2}\left[-\frac{t^4}{12} + \frac{t^3 t_a}{3}\right] \tag{4}$$

$$A_m = \frac{F}{M} = \frac{K}{M}(S_c - S_m) = \omega^2\left[\frac{A_p}{t_a^2}\left(-\frac{t^4}{12} + \frac{t^3 t_a}{3}\right) - S_m\right]$$

$$\frac{A_m}{\omega^2} + S_m = -\frac{A_p t^4}{12 t_a^2} + \frac{A_p t^3}{3t_a} \tag{5}$$

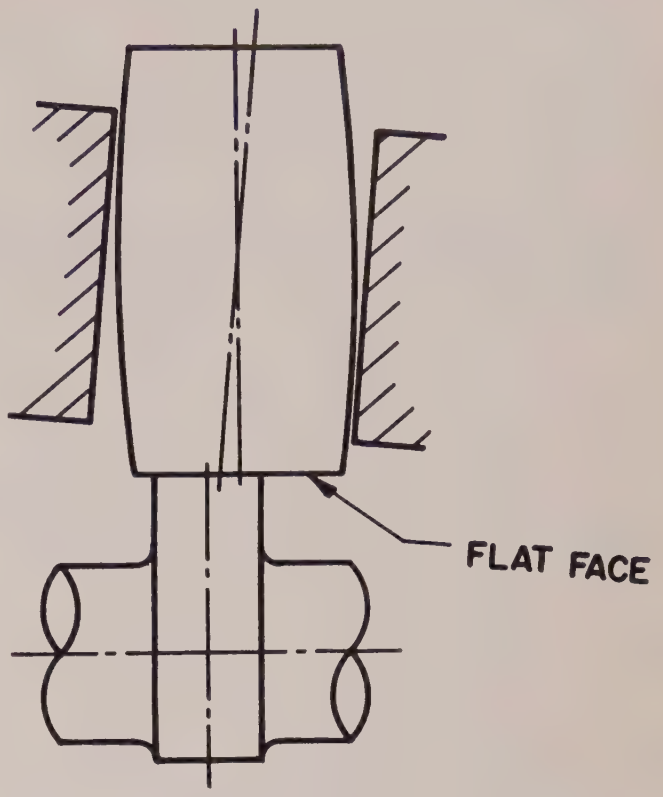


Fig. 14 – Hyperbolic tappet offers new approach to alignment problem

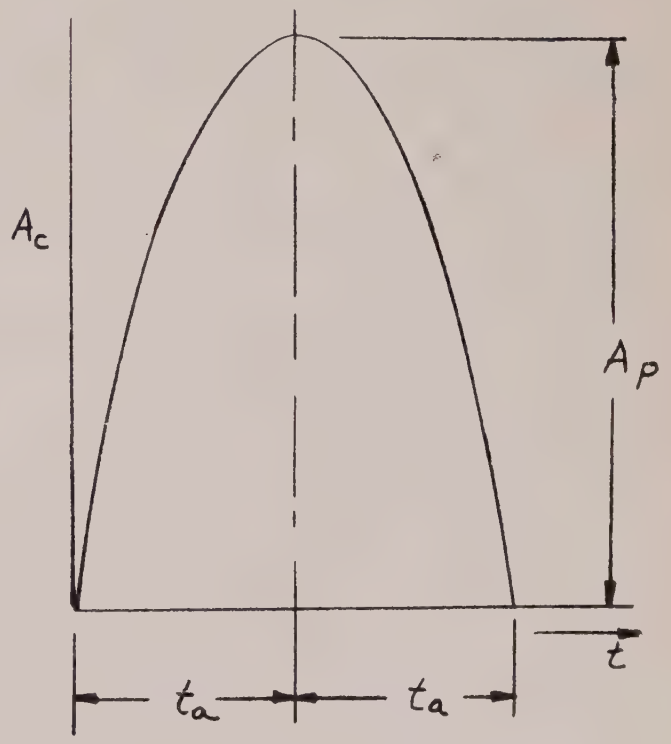


Fig. 15 – Parabolic acceleration curve

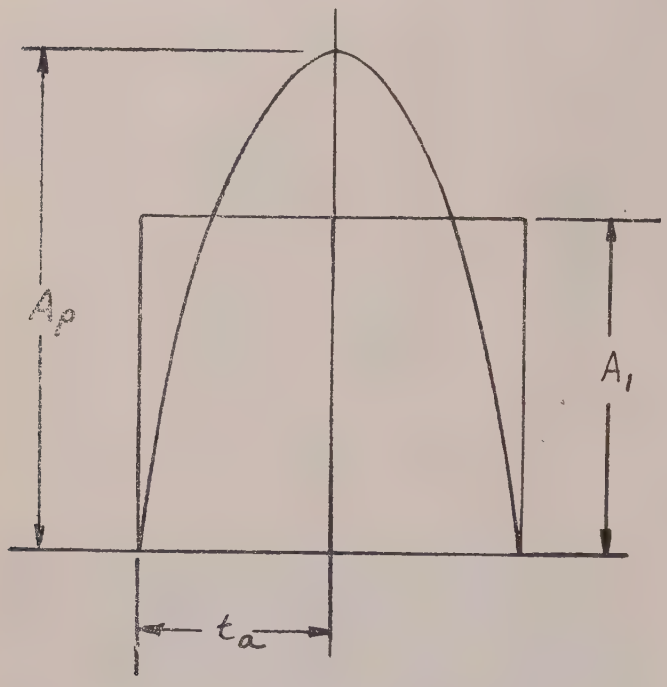


Fig. 16 – Comparison of constant and parabolic acceleration cams

$$S_m = C_1 \cos \omega t + C_2 \sin \omega t - \frac{A_p t^4}{12 t_a^2} + \frac{A_p t^3}{3 t_a} + \frac{A_p t^2}{\omega^2 t_a^2} - \frac{2A_p t}{\omega^2 t_a} - \frac{2A_p}{\omega^4 t_a^2} \quad (6)$$

$$V_m = -C_1 \omega \sin \omega t + C_2 \omega \cos \omega t - \frac{A_p t^3}{3 t_a^2} + \frac{A_p t^2}{t_a} + \frac{2A_p t}{\omega^2 t_a^2} - \frac{2A_p}{\omega^2 t_a} \quad (7)$$

$$A_m = -C_1 \omega^2 \cos \omega t - C_2 \omega^2 \sin \omega t - \frac{A_p t^2}{t_a^2} + \frac{2A_p t}{t_a} + \frac{2A_p}{\omega^2 t_a^2} \quad (8)$$

$$\frac{A_m}{\omega^2} = -C_1 \cos \omega t - C_2 \sin \omega t - \frac{A_p t^2}{\omega^2 t_a^2} + \frac{2A_p t}{\omega^2 t_a} + \frac{2A_p}{\omega^4 t_a^2} \quad (9)$$

Equation 9 was substituted in equation 5 to give the solution for S_m in (6)

when:

$$t = 0,\ s_m = 0$$

$$C_1 - \frac{2A_p}{\omega^4 t_a^2} = 0$$

$$\therefore C_1 = \frac{2A_p}{\omega^4 t_a^2}$$

when:

$$t = 0,\ V_m = 0$$

$$C_2 \omega = \frac{2A_p}{\omega^2 t_a} = 0$$

$$\therefore C_2 = \frac{2A_p}{\omega^3 t_a}$$

$$S_m = \frac{2A_p}{\omega^4 t_a^2} \cos \omega t + \frac{2A_p}{\omega^3 t_a} \sin \omega t - \frac{A_p t^4}{12 t_a^2} + \frac{A_p t^3}{3 t_a} + \frac{A_p t^2}{\omega^2 t_a^2} - \frac{2A_p t}{\omega^2 t_a} - \frac{2A_p}{\omega^4 t_a^2} \quad (10)$$

$$\delta = S_c - S_m = -\frac{2A_p}{\omega^4 t_a^2} \cos \omega t - \frac{2A_p}{\omega^3 t_a} \sin \omega t - \frac{A_p t^2}{\omega^2 t_a^2} + \frac{2A_p t}{\omega^2 t_a} + \frac{2A_p}{\omega^4 t_a^2}$$

$$\delta = -\frac{2A_p}{\omega^4 t_a^2} \cos \omega t - \frac{2A_p}{\omega^3 t_a} \sin \omega t - \frac{A_p}{\omega^2} \left[\frac{t^2}{t_a^2} - \frac{2t}{t_a} - \frac{2}{\omega^2 t_a^2} \right] \quad (11)$$

when:

$$t = 2t_a$$

$$\frac{\delta}{A_p/\omega^2} = -\frac{2}{\omega^2 t_a^2} \cos 2\omega t_a - \frac{2}{\omega t_a} \sin 2\omega t_a - 4 + 4 + \frac{2}{\omega^2 t_a^2} \quad (12)$$

$$= \frac{2}{\omega^2 t_a^2} (1 - \cos 2\omega t_a) - \frac{2}{\omega t_a} \sin 2\omega t_a$$

$$= 2 \left[\frac{1 - \cos 2\omega t_a}{\omega^2 t_a^2} - \frac{\sin 2\omega t_a}{\omega t_a} \right] \quad (13)$$

Comparison of constant and parabolic acceleration cams to give equal area under curves $V_p = V_1$ (see Fig. 16):

For equality of V at $t = 2t_a$:

$$A_1 \times 2t_a = \frac{A_p}{t_a^2} \left[t_a t^2 - \frac{t^3}{3} \right] \quad (14)$$

Solving A_p gives:

$$A_p = 1.5A_1$$

Substituting in (14):

$$\frac{\delta \omega^2}{A_1} = 3 \left[\frac{1 - \cos 2\omega t_a}{\omega^2 t_a^2} - \frac{\sin 2\omega t_a}{\omega t_a} \right] \quad (15)$$

$$R = \frac{\omega t_a}{II}$$

speed is proportional to $\frac{1}{R}$

Relative magnitudes of acceleration would be given by $\frac{\delta \omega^2}{A_1 R^2}$ since acceleration is proportional to the square of the speed.

The values plotted in the curves in Figs. 2 to 7, inclusive, are $\frac{\delta \omega^2}{A_1 R^2}$ (relative amplitude) versus $\frac{1}{R}$ (cam speed).

Note: Derivation of the formulas for the other acceleration curves are available upon request.

References

1. "Considerations in Valve Gear Design," by V. C. Young. SAE *Quarterly* Transactions, Vol. 1, July, 1947, pp. 359-365.

2. "Inspection of Cam Contours by the Electronic Method," by M. C. Turkish. *Eaton Forum*, March, 1950.

3. "Valve Lash, Automatic Tappets, and Instrumentation," by V. Ayres. Paper presented at SAE National Passenger-Car, Body, and Materials Meeting, Detroit, March 8, 1951.

4. "Aircraft-Engine Inlet and Exhaust Porting," by V. C. Young. *SAE Journal* (Transactions), Vol. 52, May, 1944, pp. 202-206.

5. "Valve Gear Design," by M. C. Turkish. Pub. by Eaton Mfg. Co., Detroit, 1946.

Calculation of High-Speed Valve Motion with a Flexible Overhead Linkage

Philip Barkan

Paper presented at SAE Nat'l Passenger-Car, Body, and Materials Meeting, Detroit, March 5, 1953.

THE cam designer today must rely mainly on empirical knowledge to anticipate difficulties arising from the use of flexible overhead linkage. The result is that development of a high-speed overhead-valve mechanism is frequently an expensive trial and error process. Even where elaborate theoretical analyses have been employed, the element of unpredictability, although lessened, still persists.

The object of this paper is to describe a method of calculating the true motion of a valve which is actuated by a cam, through a flexible linkage. It is intended to assist the designer in understanding the valve-motion phenomenon and to enable him to predict it quantitatively with sufficient accuracy at a given speed and a given valve lash.

Nomenclature

Note: The Newtonian convention for designating derivatives is employed only to indicate total derivatives with respect to time, for example, $\dot{x} = dx/dt$, and so forth.

b = Velocity proportional friction factor (internal)

$\mathrm{F}(t)$ = Force function as defined in equation 6

F_n = Value of $\mathrm{F}(t)$ at beginning of nth increment

h = Spring precompression = difference between spring free length and its length installed with valve closed

$K = M(\nu_0')^2$ = linkage rigidity of single mass system

K' = Rigidity of a component in linkage of Fig. 3

K'' = Rigidity of a component in linkage of Fig. 2

$K_1 = [1 + k/K - \varphi] \cdot K$

$K_3 = [1 + k/K + \varphi] \cdot K$

k = Valve spring rigidity, or spring rate

M = Equivalent mass of single mass system

M' = Mass of component of linkage of Fig. 3

M'' = Mass of component of linkage of Fig. 2

m = An integer, representing number of term in an infinite series

N = Camshaft rotational speed

$N_1 \cdots N_8$ = Constant multipliers of the variable quantities of equations 20 and 21

n = An integer, representing number of increment

$P_{\lambda\mu}$ = Amplitude of resonant spring vibration force in $(\lambda\mu)$th mode

THE author presents a method for calculating high-speed motion of a cam-actuated engine valve, operated with a flexible linkage.

Correlation between calculated and experimental valve motion is shown to be reasonably good. The designer is enabled to understand the valve-motion phenomenon, and to predict it quantitatively, thereby eliminating many of the usual expensive trial processes.

Discussion of this and other papers on "Valve-Gear Problems in Modern Overhead-Valve Engines" starts on page 714.

PHILIP BARKAN (J '49) is with the Advance Engineering Development Section of the Switchgear and Control Division, General Electric Co. He received his B.S. in M.E. from Tufts College, his M.S. in M.E. from the University of Michigan, and his Ph.D. in M.E. from Pennsylvania State College in 1945, 1948, and 1953, respectively.

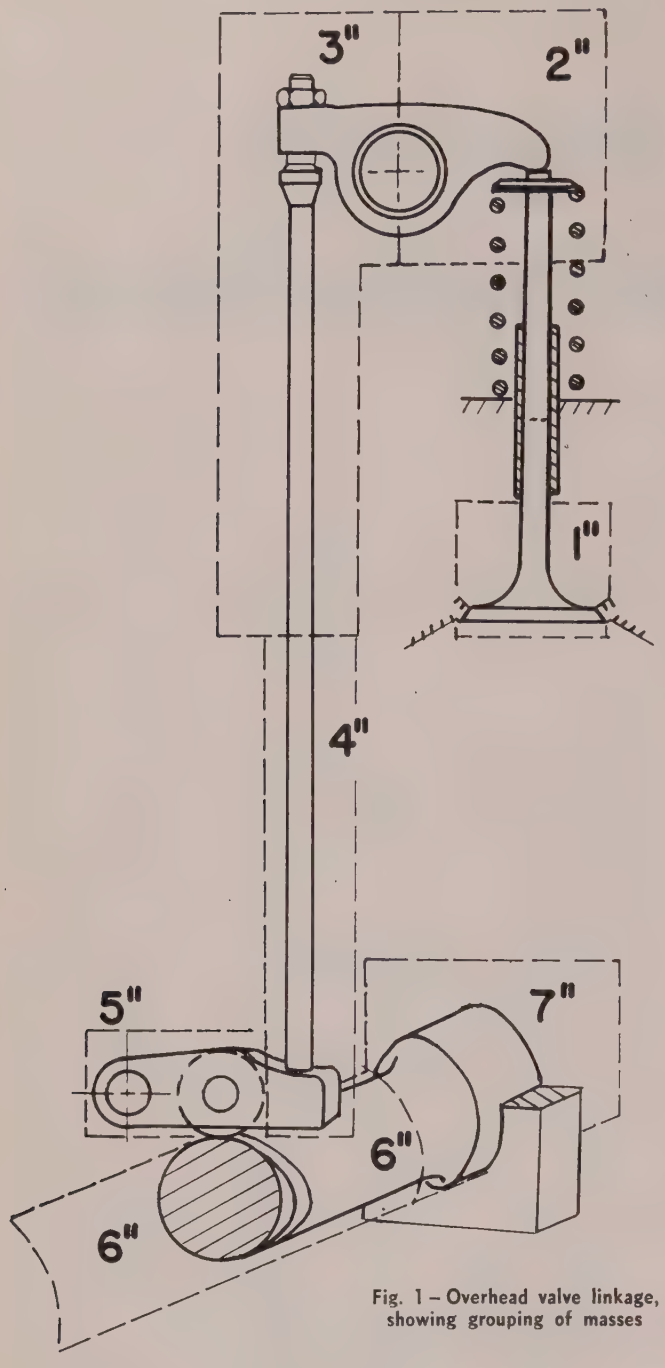


Fig. 1 – Overhead valve linkage, showing grouping of masses

Table 1 – Mass and Rigidity of Components

Masses		Rigidity	
M'_1	valve head + ½ valve stem	K'_{12}	valve stem
M'_2	½ valve stem + spring cap and keys + ⅓ valve spring + upper rocker arm (valve side)	K'_{23}	upper rocker arm
M'_3	upper rocker arm (push-rod side) + ½ push-rod	K'_{34}	push-rod
M'_4	½ push-rod + lower rocker lever (push-rod section)	K'_{45}	lower rocker lever
M'_5	lower rocker lever (cam section)	K''_{56}	assumed infinite
M'_6	⅔ of camshaft between journals	K''_{67}	camshaft in bending
M'_7	⅓ of camshaft between journals + camshaft journals	K''_{78}	oil film on camshaft bearings

R = Lever ratio, from geometry of linkage,

$$= \frac{\text{lift at valve}}{\text{lift at cam}} \text{ (with infinite rigidity of linkage)}$$

r_{ij} = Ratio: $\dfrac{\text{lift at datum } i}{\text{lift at datum } j}$

r = Lever ratio: $\dfrac{\text{valve lift}}{\text{push rod lift}}$

t = Time

$x = Y - y$ = relative displacement of M to equivalent cam follower

x_n = Value of x at the beginning of the nth increment

$(x + \epsilon)$ = Deviation = difference between equivalent cam lift and actual valve lift

$Y = Z \cdot R - \epsilon$

y = Dynamic lift of mass M

Z = Cam lift

$Z \cdot R$ = Equivalent cam lift

$\alpha = \nu \cdot \Delta t$

β = Velocity proportional damping factor (external)

$\gamma_{\lambda\mu}$ = Phase angle of λth mode valve-spring vibration forces in resonance with $(\lambda\mu)$th harmonic

Δt = A fixed increment of time

$\Delta\Theta = 6N\Delta t$ = Angular equivalent of time increment

ϵ = Valve lash

ζ_1 = Critical viscous damping factor $= \dfrac{b + \beta}{2M\nu_1}$

$\zeta_2 = \dfrac{\beta}{2M\nu_2}$

$\zeta_3 = \zeta_1 \cdot \dfrac{\nu_1}{\nu_3}$

Θ = Cam angle of rotation

λ = Mode of vibration

μ = Number of harmonic of valve lift curve in resonance with fundamental spring frequency

$\lambda\mu$ = Number of harmonic in resonance with λth mode of spring vibration

$\nu_1 = \nu'_0\left[1 + \dfrac{k}{K} - \varphi\right]^{1/2}$

$\nu_2 = [k/M]^{1/2}$

$\nu_3 = \nu'_0\left[1 + \dfrac{k}{K} + \varphi\right]^{1/2}$

ν'_0 = Fundamental natural frequency of linkage

$\sigma = [1 - \zeta^2]^{1/2}\,\nu\Delta t$

τ = Time relative to the start of an increment

Φ = Static load proportional friction factor

φ = Dynamic load proportional friction factor

ω = Angular velocity of camshaft

Linkage Analysis – To make mathematical treatment of the valve linkage feasible, a simplified linkage system must be developed. Figs. 1, 2, 3, and 4 depict the evolution of the simplified system.

Considerations in Development of Equivalent System B (Fig. 2) – The actual linkage of Fig. 1 is divided into several sections. The masses included within each of these sections are lumped into a concentrated mass connected to its neighbors by massless springs of appropriate rigidity

to form the system of Fig. 2.

From Fig. 1 it is seen that the exhaust valve, spring, spring cap, and push-rod undergo simple reciprocating motion, while the upper rocker arm and the lower rocker lever undergo oscillating rotary motion. In addition, because of the flexibility in the camshaft and the compressibility of the oil film between the camshaft journal and bearings, the camshaft and its journals can be considered to undergo a reciprocating type of motion.

Reciprocating Parts – The reciprocating parts may be directly weighed and, with the exception of the camshaft and valve spring, may be directly added to form the appropriate masses, designated as 1″ to 7″ in Fig. 2. It can be shown that one-third of the valve-spring mass should be used in mass 2″, and about two-thirds of the mass of the center section of the camshaft should be used in mass 6″; the remainder of the camshaft mass should be placed in mass 7″.

Rotating Parts – The general approach to the problem of substituting the point masses shown in Fig. 2 for rotating parts in Fig. 1 is briefly:

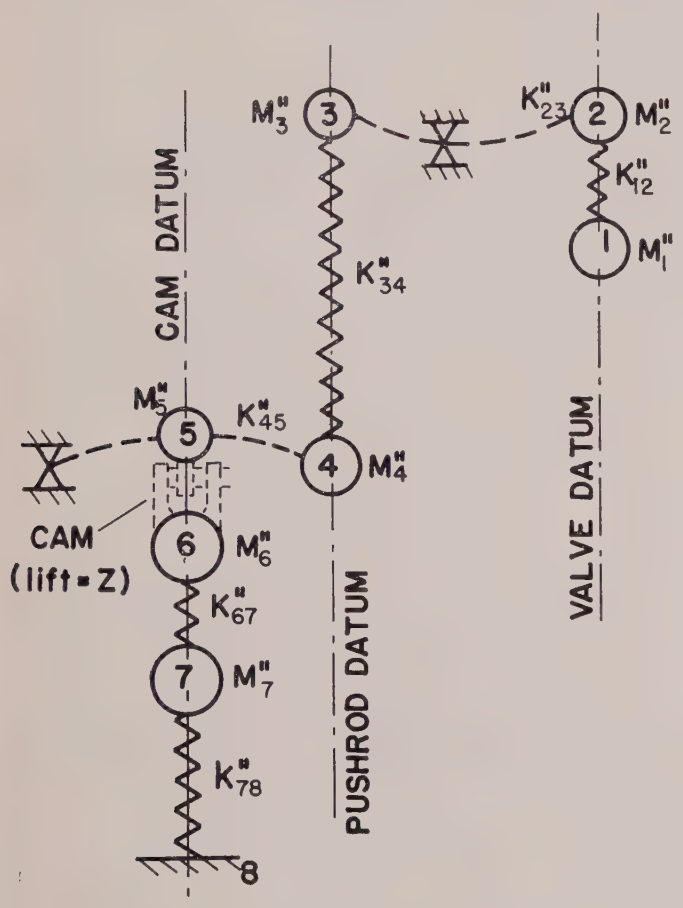
(a) Determine the moment of inertia, I, of the piece about its center of rotation.

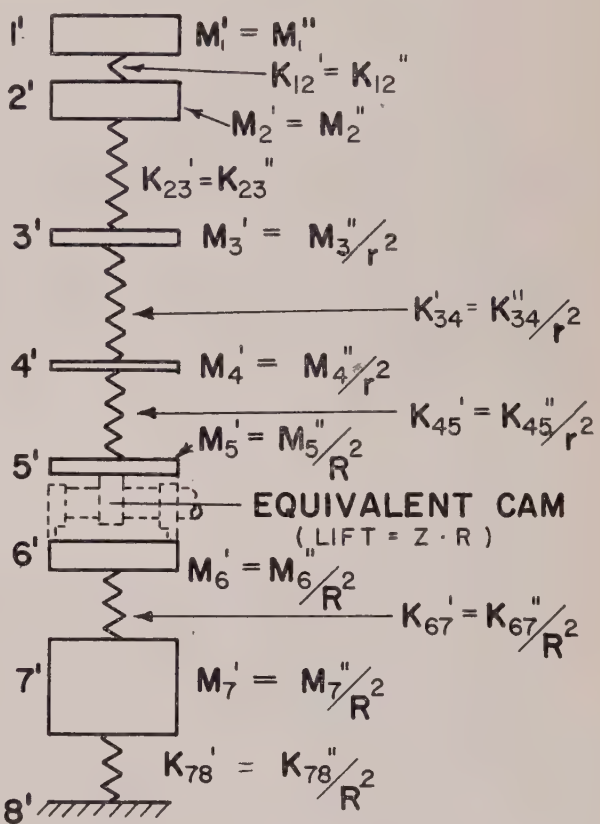
(b) Substitute a simplified member, with point masses at convenient radii from the center of rotation, so that the simplified member has the same moment of inertia, same center of gravity, and same total mass, as the actual component.

(c) Since the angular displacements of the rotary parts are small, the displacements of these point masses can be considered to be linear and treated as reciprocating motion.

The mass and rigidity of components of Fig. 2 are composed as shown in Table 1.

Development of Equivalent System C, Fig. 3 – It is next necessary to construct a system which is dynamically equivalent to Fig. 2, but in which the parts all act along the same datum. The valve axis is selected as this datum since we are most interested in the motion of the valve. Note that in this new system of Fig. 3, the cam has been replaced by an equivalent cam whose lift is R times





FUNDAMENTAL NATURAL FREQUENCY = λ_0'

r = LEVER RATIO $\frac{\text{VALVE LIFT}}{\text{PUSHROD LIFT}}$

R = LEVER RATIO $\frac{\text{VALVE LIFT}}{\text{CAM LIFT}}$

Fig. 2 (above) – Intermediate linkage, system B

Fig. 3 (right) – Intermediate linkage, equivalent system C

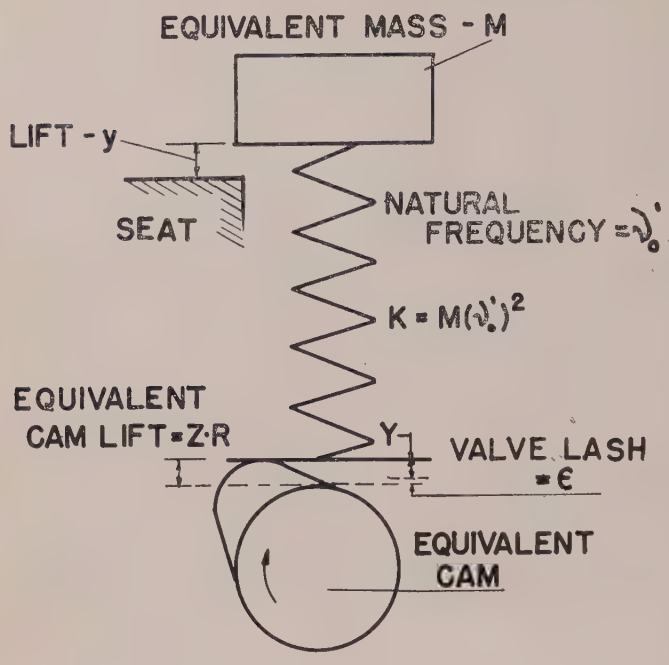


Fig. 4 – Single mass linkage, system D

the actual cam lift Z, where R is the factor representing the amplifying effect of the levers. Thus, equivalent cam lift $= Z \cdot R$.

The transition from Fig. 2 to the dynamically equivalent system of Fig. 3 is based upon the requirement that both potential and kinetic energy of corresponding components be equal. It is easily shown that this requirement is satisfied by the following relationships:

$$M' = \frac{M''}{(r_{ij})^2} \tag{1}$$

$$K' = \frac{K''}{(r_{ij})^2} \tag{2}$$

The relationships between corresponding components of systems B and C are given in Fig. 3.

Development of Simplified System D (Fig. 4) from System C – Since it requires some 13 terms to define the multimass system C, and only two to define single-mass system D, it is evident that many properties of the multimass system must be sacrificed. Thus, this simplification must be made with discretion.

The parameters selected to define the single-mass system are:

(1) The mass of system D is equal to the total mass of the cam-actuated parts of system C, or:

$$M = M_1' + M_2' + M_3' + M_4' + M_5'$$

(2) The natural frequency of system D is equal to the lowest or fundamental natural frequency of system C $= \nu_o'$. (A Holzer tabulation may be used for calculating ν_o'.)

The first condition makes it possible to employ in the calculations a spring force of the same magnitude as is exerted on the original system, Fig. 1. The second insures that the system's response to the external excitation will be of the same phase as the original system, since experiment shows that the fundamental frequency is predominant in the actual system.

This second requirement also provides a means for introducing into the calculations the effects of the distribution of rigidity and mass throughout the original system. As an illustration, it is apparent in Fig. 3 that if originally $M'_1 >> M'_4$, and if the magnitude of M'_1 and M'_4 are interchanged, a significant change in the system has occurred. This change is directly reflected in the magnitude of ν_o', the fundamental natural frequency of the system. Thus, a means is provided for realistically studying effects of modifications in the valve linkage.

By specifying M and ν_o' of the 1-mass system of Fig. 4, the linkage rigidity K is automatically defined by the relationship:

$$K = M \cdot (\nu_0')^2 \tag{3}$$

and therefore cannot be specified according to experimental measurement. Obviously, some error is necessarily introduced at this point.

Derivation of Differential Equations of Valve Motion

When the linkage is compressed, mass M, as shown in Fig. 5, is subjected to the following forces:

A. Spring force.
B. Inertia force.
C. A linkage compression force.
D. Friction forces.
E. A gas force.

A. *Spring Force* – The spring, here considered as an external member of the linkage, exerts a net force consisting of three components:

1. A normal spring compression force $= k \cdot y$.
2. A preload force $= k \cdot h$.
3. A force produced by resonant vibrations of the spring in all its modes

$$= \sum_{\lambda} P_{\lambda\mu} \sin (\lambda\mu\omega t + \gamma_{\lambda\mu}).$$

B. *Inertia Force* – The inertia force is $M \cdot \ddot{y}$.

C. *Linkage Compression Force* – The compression of the linkage is $(Y - y)$. The force on mass M produced by this compression is then $K(Y - y)$.

D. *Friction Force* – There are several points in the linkage at which relative motion and hence friction can develop. Mathematically, the sum of all these frictional forces will be approximated by three terms:

1. *Coulomb Friction (Proportional to Load)* – The net load acting on the members of the linkage is $K(Y - y)$. The coulomb type of frictional force is proportional to this load and acts in the direction opposite to the motion of mass M. The friction force is then:

$$\left[\frac{\dot{y}}{|\dot{y}|}\right]\varphi K(Y - y)$$

where:

$$\left[\frac{\dot{y}}{|\dot{y}|}\right] = \begin{matrix} +1 \text{ when } \dot{y} \text{ is positive} \\ -1 \text{ when } \dot{y} \text{ is negative} \end{matrix}$$

2. *Viscous Friction (Proportional to Relative Velocity)* – This type of friction arises from two sources:

(a) The large number of joints in the actual system contain either partial or full oil films. As the load on the parts varies, the thickness of these oil films also changes, that is, oil at a considerable local pressure is being pumped into or out of the joints. This pumping work absorbs energy from the system.

(b) An internal friction is developed in the parts as they alternately elongate and compress. To a first approximation this friction may be considered as proportional to the rate of change of strain.

The simplest means of approximating internal viscous friction is to assume a dashpot, as in Fig. 5, parallel with the linkage. Employing a friction proportionality factor, b, the damping terms is $b(\dot{Y} - \dot{y})$. This force will always act in the direction opposite to the relative motion $(\dot{Y} - \dot{y})$.

3. *Viscous Friction (Proportional to Absolute Velocity)* – As components move over stationary guides and pivots, there develops a friction force, assumed proportional to the velocity $\dot{y}$ of mass M. This friction, always acting opposite the direction of motion, is then $\beta \cdot \dot{y}$.

E. *Gas Force* – The difference in pressure between the cylinder and the manifold, acting upon the valve-head projected area, gives rise to a gas force term F_g. If $P(t)$ represents the cylinder pressure as a function of time and $p(t)$ the manifold pressure, then the gas force term is approximately:

$$F_g = [P(t) - p(t)] \cdot A$$

where A is the projected area of the valve head.

Summation of Terms

D'Alembert's principle states that the net force acting on any dynamic system must be zero. Thus, the sum of terms A, B, C, D, and E is zero, or:

Linkage Compression	Inertia	Coulomb Friction	External Viscous Friction	Internal Viscous Friction

$$K(Y - y) - M\ddot{y} - \frac{\dot{y}}{|\dot{y}|}\varphi K(Y - y) - \beta\dot{y} - b(\dot{y} - \dot{Y})$$

Spring Force — Gas Force

$$- k(y + h) - \sum_{\lambda} P_{\lambda\mu} \sin(\lambda\mu\omega t + \gamma_{\lambda\mu}) - F_g = 0 \qquad (4)$$

Now make a substitution in the variable, and let:

$$\left.\begin{aligned} x &= Y - y = \text{Displacement of } M \text{ relative to equivalent cam follower} \\ \dot{x} &= \dot{Y} - \dot{y} = \text{Velocity of } M \text{ relative to equivalent cam follower} \\ \ddot{x} &= \ddot{Y} - \ddot{y} = \text{Acceleration of } M \text{ relative to equivalent cam follower} \end{aligned}\right\} \qquad (5)$$

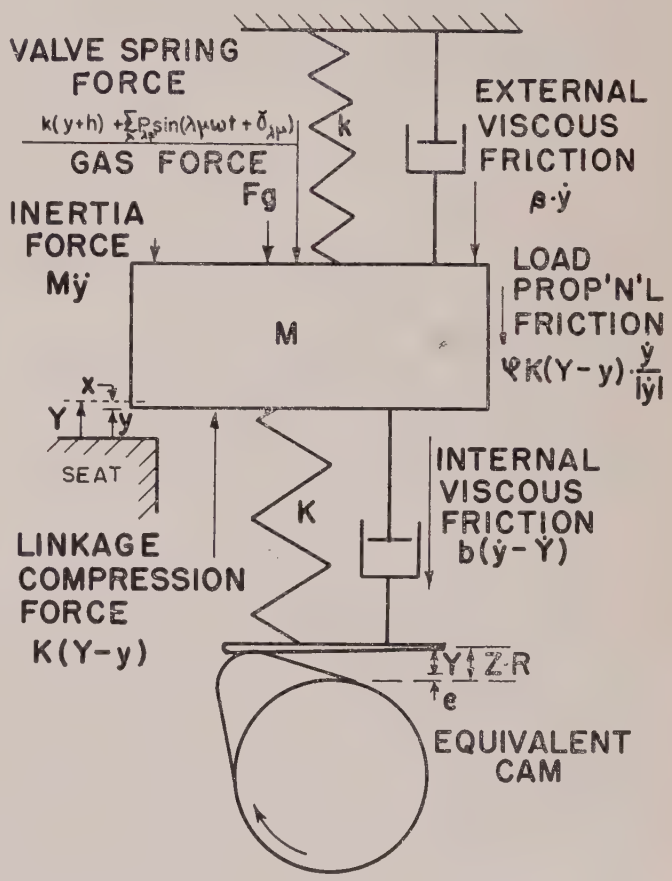


Fig. 5 – Forces acting upon system when linkage is compressed

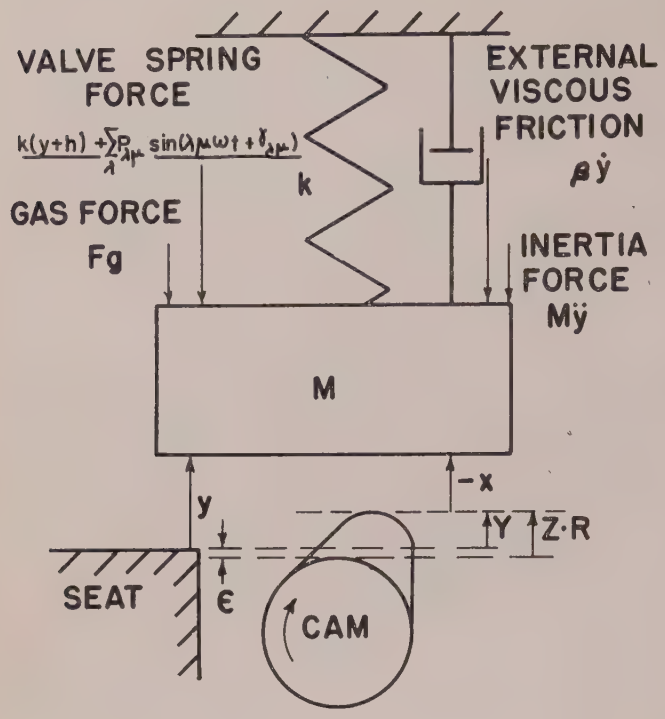


Fig. 6 – Forces acting upon system when linkage jumps off cam

Substituting equation 5 in equation 4, and regrouping terms, the differential equation becomes:

$$M\ddot{x} + (b + \beta)\,\dot{x} + \left[1 + \frac{k}{K} - \frac{\dot{y}}{|\dot{y}|}\varphi\right] Kx = M\ddot{Y} + k\,(Y + h) + \sum_\lambda P_{\lambda\mu}\sin\,(\lambda\mu\omega t + \gamma_{\lambda\mu}) + F_g + \beta\dot{Y} = F\,(t) \qquad (6)$$

Equation 6 is actually two equations, for when $\dot{y}$ is positive, the term:

$$\left[1 + \frac{k}{K} - \frac{\dot{y}}{|\dot{y}|}\varphi\right] K = \left[1 + \frac{k}{K} - \varphi\right] K = K_1 \qquad (7)$$

and when $\dot{y}$ is negative:

$$\left[1 + \frac{k}{K} - \frac{\dot{y}}{|\dot{y}|}\varphi\right] K = \left[1 + \frac{k}{K} + \varphi\right] K = K_3 \qquad (8)$$

Two substitutions are made for convenience:

A. Let:

$$\left.\begin{aligned} \nu_1 &= \sqrt{\frac{K_1}{M}} \\ \nu_3 &= \sqrt{\frac{K_3}{M}} \end{aligned}\right\} \qquad (9)$$

B. Introduce critical damping factors ζ_1, and ζ_3:

$$\left.\begin{aligned} \zeta_1 &= \frac{b + \beta}{2\,M\nu_1} \\ \zeta_3 &= \frac{b + \beta}{2\,M\nu_3}\left(\therefore\ \zeta_3 = \zeta_1\frac{\nu_1}{\nu_3}\right) \end{aligned}\right\} \qquad (10)$$

Substituting equations 7, 8, 9, and 10 in equation 6 results in:

$$\ddot{x} + 2\,\zeta_1\nu_1\,\dot{x} + \nu_1^2\,x = \frac{1}{M}\,F\,(t) \qquad (11)$$

Equation 11 is valid for $\dot{y} > 0$, that is, valve opening.

and
$$\ddot{x} + 2\,\zeta_3\nu_3\,\dot{x} + \nu_3^2\,x = \frac{1}{M}\,F\,(t) \qquad (12)$$

Equation 12 is valid for $\dot{y} < 0$, that is, valve closing.

Note that the right-hand sides of equation 11 and 12 are identical, therefore $F(t)$ is independent of the direction of valve motion. These two equations define the valve motion for all positive x, that is, whenever the linkage is in compression.

A third equation is necessary to describe the system when the linkage jumps off the cam. This occurs whenever x is negative. The entire linkage mass can then be considered to be riding on the valve spring, which results in a single-mass oscillator, such as Fig. 6. Friction in the jumping system is represented by the velocity proportional friction term $\beta\dot{y}$. The mass of this system is the same as that for the previous case, but the system rigidity is now that of the valve spring. The differential equation is then:

$$-M\ddot{y} - \beta\dot{y} - k\,(y + h) - \sum_\lambda P_{\lambda\mu}\sin\,(\lambda\mu\omega t + \gamma_{\lambda\mu}) - F_g = 0$$

Again substituting:

$$\left.\begin{aligned} y &= Y - x \\ \dot{y} &= \dot{Y} - \dot{x} \\ \ddot{y} &= \ddot{Y} - \ddot{x} \end{aligned}\right\}$$

$$\nu_2 = \sqrt{\frac{k}{M}}$$

$$\text{and } \zeta_2 = \frac{\beta}{2\,M\nu_2}$$

results in:

$$\ddot{x} + 2\,\zeta_2\nu_2\,\dot{x} + \nu_2^2\,x = \frac{1}{M}\,F\,(t) \qquad (13)$$

Equation 13 is valid for $x < 0$.

Equation 13 is in the same form as equations 11 and 12, and the same general solution will apply to all three equations. Also, note that the right-hand side of equation 13 is identical to $F(t)$ of equations 11 and 12. This makes it possible to use the same force function, $F(t)$, in a given problem, regardless of whether x is positive or negative and regardless of the direction of valve motion.

The equations which have been developed are in terms of x which is the displacement of mass M relative to the follower of the equivalent cam. The

dynamic valve motion y is easily determined from x by the relationship:

$$y = Z \cdot R - \epsilon - x, \text{ or by definition, } y = Y - x \qquad (14)$$

The term $x + \varepsilon$ will be referred to as "deviation." Deviation is defined as the amount that the actual lift deviates from the equivalent cam lift, as shown in Fig. 7. Positive values of x represent linkage compression, indicating that the linkage is in contact with the cam. Since the linkage is composed of many separate components, it cannot sustain tension forces, therefore negative x must indicate that the linkage is separated from the cam, or that the valve "jumps."

Boundary Conditions

As long as the valve remains on its seat, the cam equivalent lift $Z \cdot R$ is taken up by valve lash ϵ, and by the compression of the linkage. When the linkage compressive force, so produced, exceeds the opposing forces, the valve, or M of the equivalent system (initially at zero lift, zero velocity, and zero acceleration), will begin to accelerate. The beginning of the problem occurs at θ_o, the angle at which the net force on the mass M becomes positive.

Up to and including the instant θ_o, the linkage compressive force is $K(ZR - \varepsilon)$ or KY. The opposing forces acting on M are the gas force F_g, the spring preload $k \cdot h$, spring-vibration force $\sum_{\lambda} P_{\lambda\mu} \sin(\lambda\mu\omega t + \gamma_{\lambda\mu})$, and a friction force φKY. There is no inertia force since the valve starts from rest with zero acceleration. Thus θ_o is the instant at which:

$$KY = kh + \sum_{\lambda} P_{\lambda\mu} \sin(\lambda\mu\omega t + \gamma_{\lambda\mu}) + F_g + \varphi KY$$

or:

$$[1 - \varphi]\, KY = kh + \sum_{\lambda} P_{\lambda\mu} \sin(\lambda\mu\omega t + \gamma_{\lambda\mu}) + F_g \qquad (15)$$

If both sides of equation 15 are plotted versus θ, θ_o is determined by their intersection. The initial values, x_o and $\dot{x}_o$, to be used in the solution of the differential equation are easily found once θ_o is known.

Since:

$$y_o = 0$$

$$x_o = Y_o - y_o = Y_o \qquad (16)$$

Similarly since:

$$\dot{y}_o = 0$$

$$\dot{x}_o = \dot{Y}_o - \dot{y}_o = \dot{Y}_o \qquad (17)$$

Condition for Valve Seating – The differential equations 11, 12, and 13 lose their validity when the valve strikes its seat. This occurs when:

$$y = Y - x = 0.$$
$$x = Y \qquad (18)$$

is the condition for valve seating, and the end of the calculation. The valve seating velocity is $\dot{y} = \dot{Y} - \dot{x}$.

Numerical Solution of Differential Equations of Valve Motion

The three differential equations for x (11, 12, 13) have been put into the form of a second-order linear differential equation:

$$\ddot{x} + 2\zeta\nu\dot{x} + \nu^2 x = \frac{1}{M} F(t)$$

where M, ζ, and ν are constant within a specified region of the problem. $F(t)$ in the most general case is known only numerically, hence a numerical solution of the equation is required.

A completely general numerical solution can be obtained by making the assumption that over a small increment of time Δt, the function $F(t)$ may be represented by a linear equation of the form:

$$F(t)\Big|_{t = t_n}^{t = t_n + \Delta t} = F_n + \frac{F_{n+1} - F_n}{\Delta t} \cdot \tau$$

Fig. 8 shows a graphical representation of this approximation.

The problem can now be resolved into a series of differential equations, each valid only within a small incremental time Δt. The equation to be solved is then:

$$\ddot{x} + 2\zeta\nu\dot{x} + \nu^2 x = F_n + \frac{F_{n+1} - F_n}{\Delta t} \cdot \tau \qquad (19)$$

This is valid for: $t_n \leq t \leq t_{n+1}$

or: $t_n \leq t \leq t_n + \Delta t$

or: $0 \leq \tau \leq \Delta t$

The solution to equation 19 can be put into the following form:

$$x_{n+1} = N_1 \cdot F_n + N_2 \cdot F_{n+1} + N_3 \cdot \dot{x}_n + N_4 \cdot x_n \qquad (20)$$

$$\dot{x}_{n+1} = N_5 \cdot F_n + N_6 \cdot F_{n+1} + N_7 \cdot \dot{x}_n + N_8 \cdot x_n \qquad (21)$$

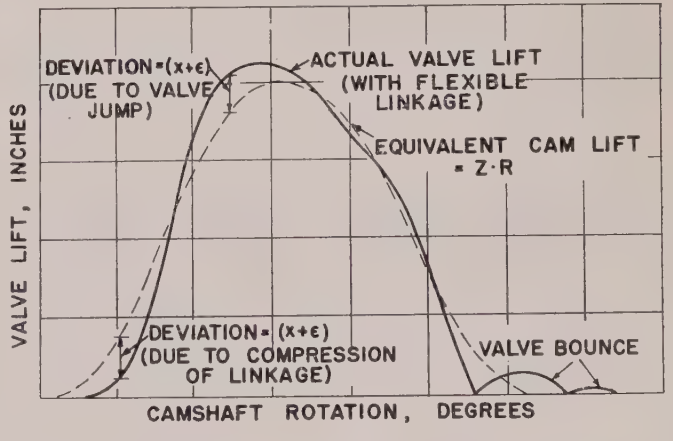


Fig. 7 – Relationship between deviation and valve motion

The expressions for the coefficients $N_1 \ldots N_8$, as given in Appendix I, are composed of parameters which are characteristics of the linkage, all of which are constant in a specified region of the problem. Thus, once evaluated, the coefficients $N_1 \ldots N_8$ may be applied to all increments at all speeds and with all cams, provided only that M, ζ, ν and Δt are held constant.

This incremental solution for the differential equations is only valid for an increment of time, Δt. At N rpm this corresponds to an angular increment of camshaft rotation of:

$$\Delta\Theta = 6N\Delta t$$

It is therefore necessary to determine the values, F_n, of the force function at these increments. The values of F_n correspond to the angles:

$$\Theta_n = \Theta_o + n \cdot \Delta\Theta$$

where $n = 0, 1, 2, 3$, and so forth.

By plotting $F(t)$ versus Θ to a large scale the values of F_n are conveniently read off at the proper angles.

To apply equations 20 and 21 to a solution, the values of F_n must be given for the beginning of each increment, Δt. Initial values, x_o and $\dot{x}_o$, are specified as boundary conditions. Values of F_o, F_1, x_o and $\dot{x}_o$ are used in equations 20 and 21 to evaluate x_1 and $\dot{x}_1$. These two values, together with F_1 and F_2 are then used to calculate x_2 and $\dot{x}_2$, and so forth. The solution continues in this manner, taking care, however, that the coefficients $N_1 \ldots N_8$ employed satisfy the existing conditions. That is, when both x and $\dot{y}$ are positive, coefficients corresponding to solution of equation 11 are employed, when x is negative, coefficients corresponding to the solution of equation 13 are employed, and so forth. The calculation ends when $x = Y$, indicating that the valve has seated.

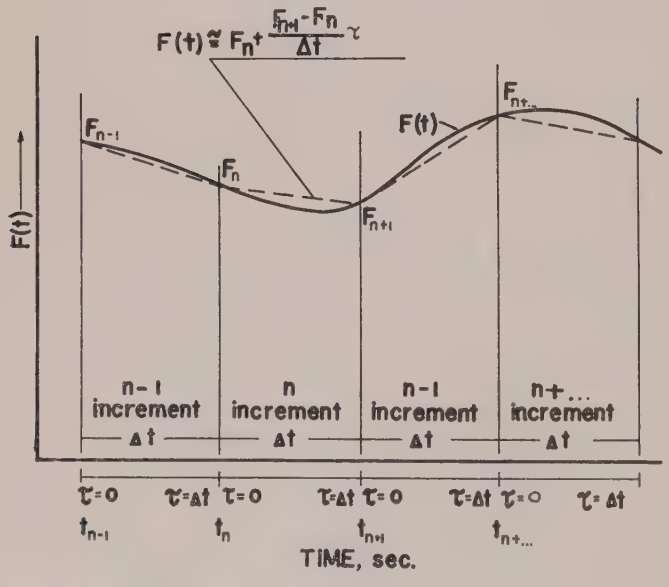


Fig. 8 – Linear approximation of $F(t)$

Evaluation of Terms

Before the method which has been developed here can be employed to predict valve motion, it is necessary to evaluate component terms of the force function, $F(t)$, and the friction terms ζ_1, ζ_2 and φ, ascertaining which are negligible, and the method of evaluating those terms which are important.

Force Function Terms

Inertia Term $M\ddot{Y}$ – The inertia term requires degree by degree tabulated data of the cam acceleration, obtained from cam design data. Cam acceleration data $\frac{d^2Z}{d\Theta^2}$ (in. per deg^2) is transformed to equivalent cam acceleration, $\ddot{Y}$ (in. per sec^2) at the valve, to form the inertia force term:

$$F_I = M\ddot{Y} = 36N^2MR\frac{d^2Z}{d\Theta^2} \qquad (22)$$

Spring Force Term – This term is

$$k(Y + h) + \sum_{\lambda} P_{\lambda\mu} \sin (\lambda\mu\omega t + \gamma_{\lambda\mu}).$$

a. The first part of this expression, $k(Y + h)$, which is the total static spring force term, involves the following:

1. Tabulated data, degree by degree, of the cam lift Z whence $Y = Z \cdot R - \varepsilon$.

2. The difference between the spring free length and its length installed, with the valve closed equals h.

3. The spring rigidity, k. If the spring coils are wound with constant pitch, k will be independent of the valve lift y. However, if a variable pitch spring is employed, or if the coils are closely wound, k will be a function of y and an experimental plot of spring force versus y must be obtained. A variable spring rate can be introduced into the force function with no error. Since, on the left side of equation 6 the spring rate appears as a second-order term, $(k/K << 1)$, negligible error results from using an average, constant value of k in evaluating the coefficients $N_1 \ldots N_8$.

b. The term $\sum_{\lambda} P_{\lambda\mu} \sin (\lambda\mu\omega t + \gamma_{\lambda\mu})$, is a representation according to the harmonic theory of the resonant vibration forces produced by the spring in all its modes λ, when the fundamental mode is in resonance with the μth harmonic.

In 1951, one of the leading cam designers stated: "To date we have not found valve-spring vibrations

or surge to be of signficant amplitude or influence, below design speed, to warrant any study." However, during the period 1920 to 1940 many investigations were devoted to valve-spring vibration phenomena. An excellent review of this work has been published.[1]

The change in emphasis on the role of the spring which has taken place during the past decade is, in part, due to improved cam designs and to recent advances in spring fabrication. Because of metallurgical improvements and shot-blasting techniques, springs today may be operated through a much wider stress range, making it possible to minimize the importance of spring vibrations by employing proportionately higher spring preloads and higher natural frequencies of the springs.

An investigation was carried out to establish to what extent the spring-vibration term influences the valve-motion calculations. It is concluded that with current spring-design practice little significant change occurs in the calculated motion of the overhead valve if the spring-vibration term is neglected.

Gas Force F_g – Gas load forces were not considered in these calculations mainly because of the great complication which would have been involved in the experimental phase of the work. They can be easily introduced in the mathematical analysis of valve motion, provided that sufficient knowledge of the nature of this gas force is available.

External Viscous Friction Term $\beta\dot{Y}$. $\dot{Y}$ is easily obtained from tabulated data of the cam velocity by the relationship

$$\dot{Y} = 6RN \frac{dZ}{d\Theta}$$

The friction factor β must be established empirically as is discussed later. In the valve motion calculations it was found that the term $\beta \dot{Y}$ is negligible compared with the other components of the force function, that is, $F(t) >> \beta \dot{Y}$. However, this term presents no complication to the calculations and may easily be included when required.

Summarization

Summarizing, for the purposes of predicting valve motion, force function $F(t)$ is

$$F(t) = k(Y + h) + M\ddot{Y} \qquad (23)$$

$F(t)$ was calculated to three significant figures in computing valve motion.

Friction Analysis – The friction terms introduced in the calculations are the load proportional friction factor φ, and the critical viscous damping factors ζ_1 and ζ_2. Since these factors can only be evaluated under the load and speed conditions prevailing in the operating linkage, the evaluation of these factors is necessarily empirical.

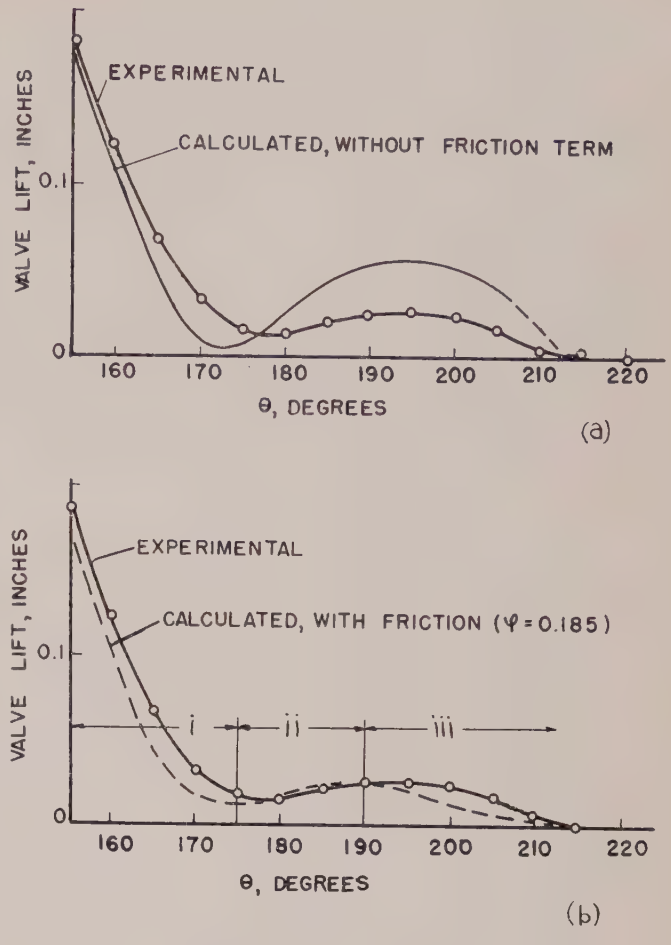


Fig. 9 – On closing ramp, calculations predicted high jump which did not occur in experimental work, as shown in (a); use of friction term φ results in closer correlation shown in (b)

It was observed that without any friction terms, the calculations failed to produce consistent correlation. For example, consider these results with cam 1:

At 1800 rpm the most serious error in the calculation occurred on the closing ramp, as shown in Fig. 9a. The calculations predicted a very high jump off the ramp which in fact did not occur in the experimental work.

At 2120 rpm the difficulty was seemingly just the opposite, the calculations predicted only a very small jump near peak lift on the opening side, while the experimental measurements showed a very large jump.

A load proportional friction term results in precisely these effects. It is seen from equation 6 that

[1] See Report No. 1945/R/8, "Surging of Valve Springs," by G. R. Oliver and H. R. Mills. Published by the Institution of Automobile Engineers, Great Britain.

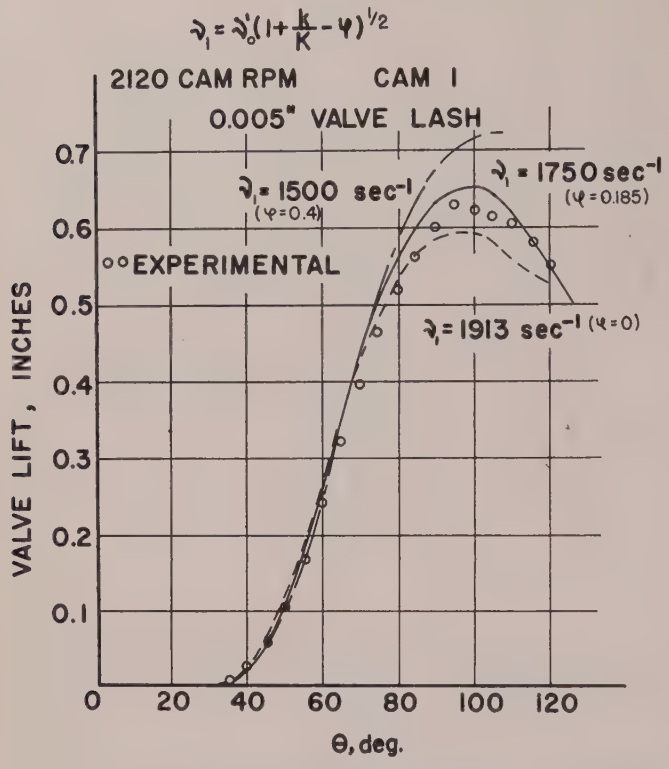


Fig. 10 – Influence of load proportional friction factor φ on calculated motion

the sign of φ is a function of the sign of $\dot{y}$. The effect of φ on the calculations, when $\dot{y}$ is positive, is shown in Fig. 10. The height of valve jump at 2120 rpm is seen to vary directly with φ. A value of $\varphi = 0.185$, corresponding to $\nu_1 = 1750$ radians per sec, resulted in the closest correlations with experimental results.

At 1800 rpm, $\dot{y}$ is negative in regions *i* and *iii* of Fig. 9b, but is positive in region *ii*. By employing equation 12 for negative $\dot{y}$, in regions *i* and *iii*, and equation 11 for positive $\dot{y}$ in region *ii*, the resulting valve motion closely correlates with experiment. The value $\varphi = 0.185$ was adopted for all calculations.

The critical viscous damping factor for internal friction ζ_1 was found to be not as critical as φ, but distinct improvements result from introducing it. In calculations with $\zeta_1 = 0$ there appeared a tendency for the oscillations to grow in amplitude in a manner unlike that observed in any of the experimental work. The presence of a viscous damping term of order of magnitude $\zeta_1 = 0.05$ to 0.15 restricts this instability and results in more realistic calculations. A value of ζ_1 equal to 0.15 was used in all calculations.

The critical viscous damping factor for external friction ζ_2, is attributable to the frictional resistance when the parts are not under load, such as occurs during valve jump. By removing the valve spring from the linkage and moving the linkage by hand it is found that this friction is very small. To a first approximation $\zeta_2 = 0$.

For any other linkage the problem arises as to how to establish the magnitudes of the friction. The order of magnitude of the coefficient φ may be estimated by the static load test described in Appendix II. There does not appear to be any simple means of establishing the order of magnitude of ζ_1. Fortunately this factor does not profoundly affect the calculations and an estimate in the range $\zeta_1 = 0.05$ to $\zeta_1 = 0.15$ probably will result in satisfactory correlation.

Comparison of Calculations and Experimental Data

This method of calculating valve motion was checked by carrying out a number of calculations and comparing the results to experimentally measured valve-motion curves.

Calculations Based on Cam Design Data – Figs. 11, 12, and 13 show the results obtained by computing the valve motion solely on the basis of cam design information. The calculations successfully reproduce significant features of the experimentally measured valve-motion curves including valve-opening and valve-closing times, the approximate valve-closing velocity, and the existence of any significant valve "jumps" (as in Fig. 12).

Calculations Based upon Electronically Measured Acceleration Data of an Actual Cam – Employing an electronic acceleration analyzer developed by Turkish,[2] the true acceleration data on a given cam was measured. The variation between the actual acceleration so measured and the design acceleration values is illustrated in Fig. 14. Considering that a discrepancy of 0.0001 in. in the cam lift will cause differences of the order of magnitude shown, these differences are not too surprising. By employing both design and actual acceleration data in the calculations a measure of the effect of these variations is obtained. It was found in a study of three different cams that variations of the order of magnitude shown do not have a serious influence upon the valve motion. Because the variations are slight, the effect is best seen by basing the comparison upon the deviation curve rather than the valve-motion curve. Figs. 15 and 16 are representative of the results obtained. Important characteristics of the deviation curve are unchanged, although a slight shift in phase does exist.

Calculations Employing an Experimentally Measured Valve-Spring Vibration Term and Electronically Measured Acceleration Data – Based on experimental measurements under the specified operating conditions, a spring-vibration force term was introduced into the calculations. Fig. 17 is a graphical representation of the dynamic spring force as used in calculation of the deviation curve of Fig.

[2] See *Eaton Engineering Forum*, Vol. 11, March, 1950: "Inspection of Cam Contours by Electronic Method," by M. C. Turkish.

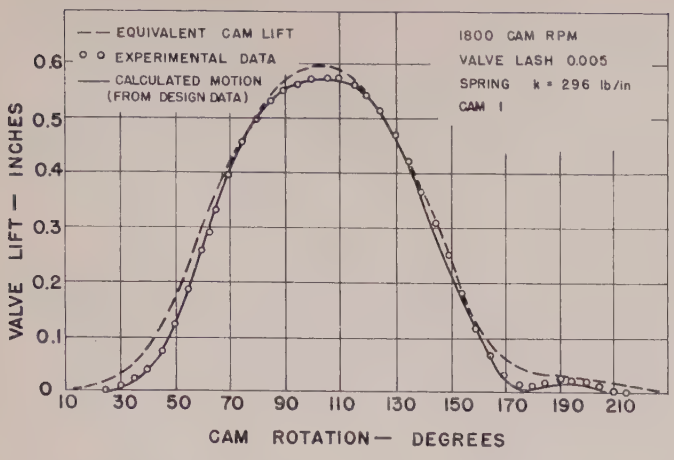


Fig. 11 – Comparison of calculated and experimental valve motion at 1800 cam rpm

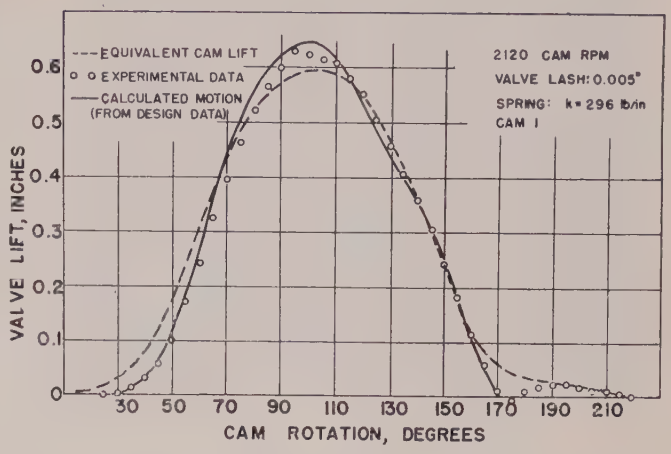


Fig. 12 – Calculated and experimental valve motion compared at 2120 cam rpm

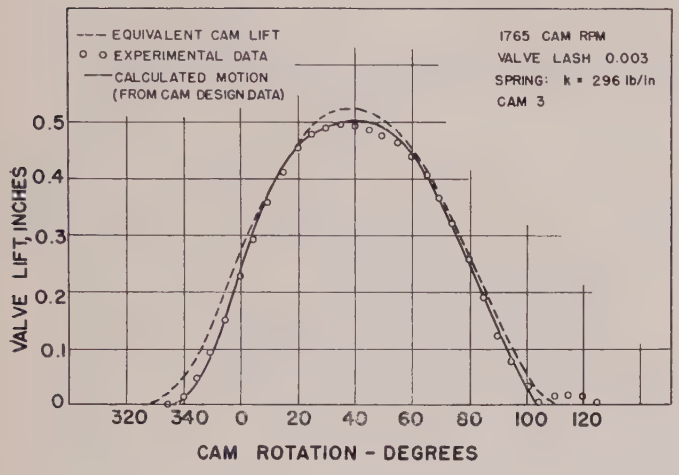


Fig. 13 – At 1765 cam rpm calculated and experimental valve motion correlate as shown here

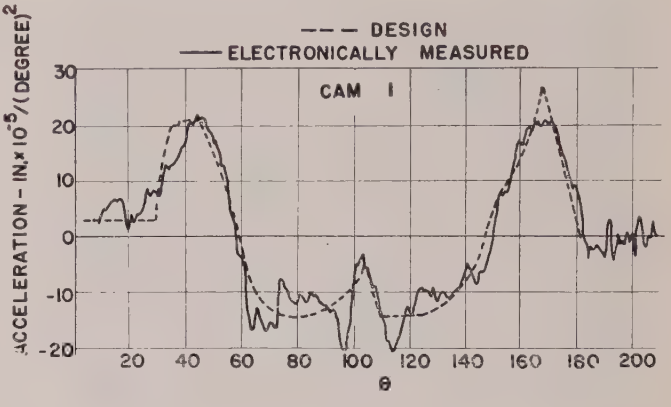


Fig. 14 – Variations between measured acceleration and design acceleration values were determined by use of electronic analyzer

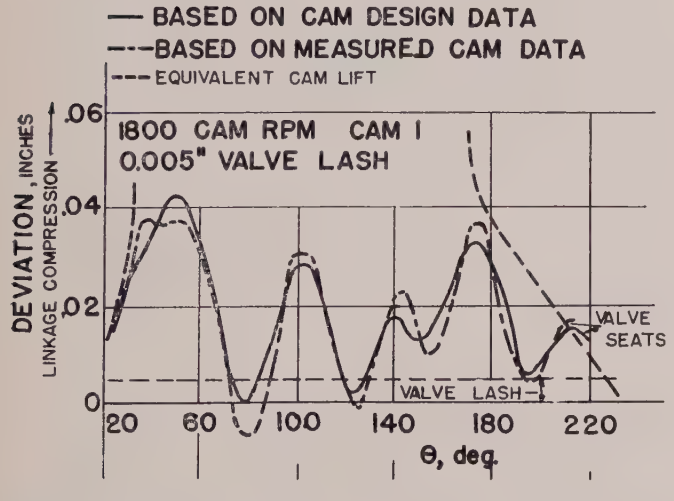


Fig. 15 – Results of one study based on cam design data and measured cam data

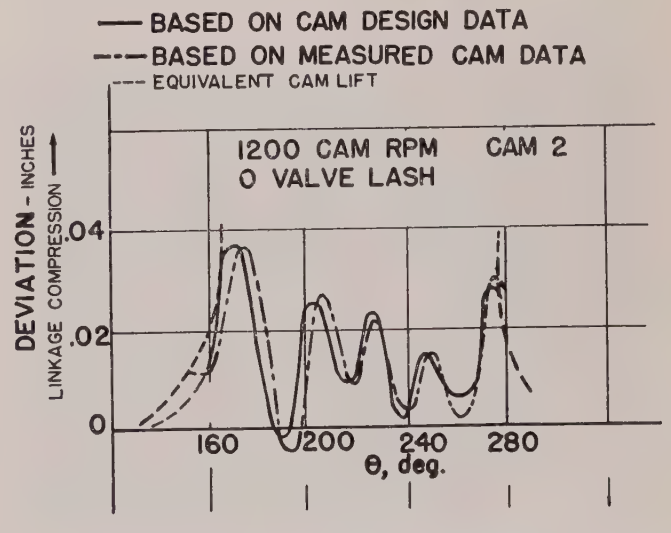


Fig. 16 – Variations of order of magnitude shown in Fig. 14 do not have serious influence on valve motion calculations

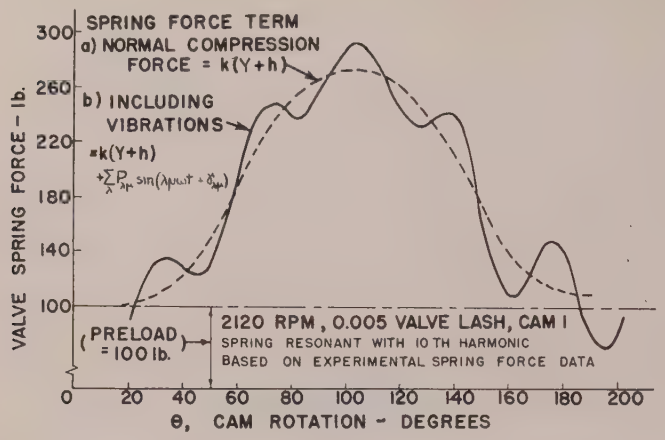


Fig. 17 – Spring force component of $F(t)$

18. The calculation is at a resonant speed where the spring-vibration effect should be maximum. Except for a shift in phase and some variation in amplitude, the spring-vibration term apparently does not have significant effect.

Oscilloscope Records of Linkage Strain – A qualitative correlation exists between the calculated deviation curves and dynamic strain records obtained by means of an SR-4 resistance-type strain gage mounted upon the upper rocker arm. During periods of valve jump the linkage compressive force is practically zero, hence these regions appear as lines of zero force upon the oscilloscope strain records. Taking this factor into account it is seen that good correlation is obtained between Figs. 15 and 19, 16 and 20, 18 and 21.

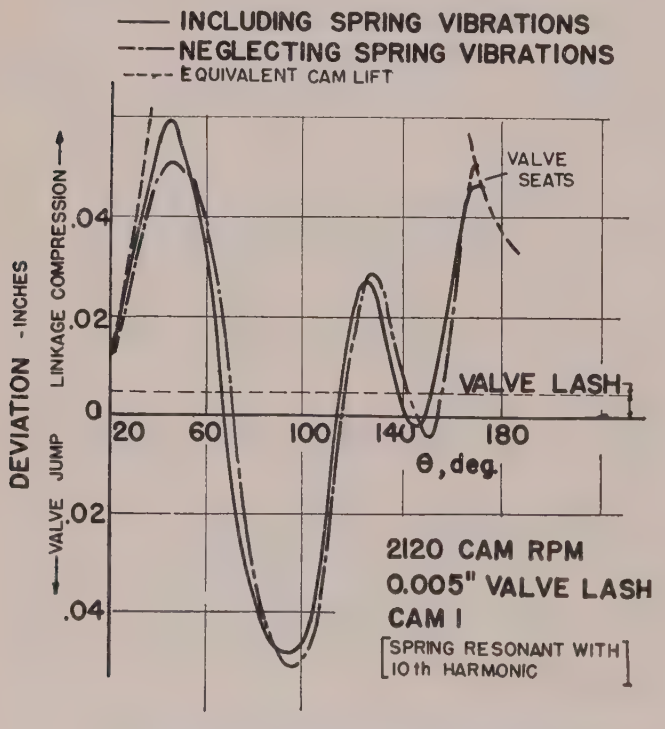


Fig. 18 – Effect of spring vibrations on calculations

Summary

A theory for calculating high-speed valve motion has been developed and confirmed by experimental measurements. It is concluded that valve-motion prediction is feasible from cam design data. If an automatic computer is available the work involved is not excessive.

For those interested in employing this method, an earlier paper[3] covers further details of this work plus a complete sample problem.

In applying this method, these steps are followed:

1. A single-mass system is developed from weight and rigidity measurements on a given linkage.
2. Friction factors φ and ζ_1 are empirically estimated.
3. A small time increment is selected and with (1) and (2) is used to calculate three sets of coefficients $N_1 \ldots N_8$, according to the relationships in Appendix I.
4. The force function $F(t)$ is calculated.
5. Boundary conditions are determined.
6. The solution is carried out for x, employing coefficients calculated in (3).
7. The valve motion is obtained from (6) by the relationship $y = Z \cdot R - (x + \epsilon)$.

APPENDIX I

Evaluations of Coefficients $N_1 \ldots N_8$

The solution employed in these calculations requires evaluation of three sets of coefficients, each corresponding to a specified operating condition of the linkage. One set of coefficients is calculated for each of the ν values listed below:

When linkage is compressed (x positive) and $\dot{y}$ is positive:

$$\nu_1 = \nu_0' \left[1 + \frac{k}{K} - \varphi\right]^{1/2}$$

When linkage is compressed (x positive) and $\dot{y}$ is negative:

$$\nu_3 = \nu_0' \left[1 + \frac{k}{K} + \varphi\right]^{1/2}$$

When valve jumps (x negative):

$$\nu_2 = \left[\frac{k}{M}\right]^{1/2}$$

[3] "Calculation of High-Speed Motion of Flexible, Cam-Actuated System," by P. Barkan. Doctor of Philosophy dissertation, the Pennsylvania State College, January, 1953.

For each of these values of ν, with a given value of ζ, and an arbitrarily selected small time increment Δt, the coefficients $N_1 \ldots N_8$ are calculated by evaluating these terms:

$$\alpha = \nu \cdot \Delta t$$

$$\sigma = (1 - \zeta^2)^{1/2} \cdot \alpha$$

$$e^{-\zeta\alpha} = 1 - \zeta\alpha + \frac{(\zeta\alpha)^2}{2} \cdots \frac{(-1)^m(\zeta\alpha)^m}{m!} + \cdots$$

$$(1 - \zeta^2)^{1/2} = 1 - \frac{\zeta^2}{2} - \frac{\zeta^4}{8} - \frac{\zeta^6}{16} - \frac{5}{128}\zeta^8 \cdots$$

$$a = e^{-\zeta\alpha}\left[\frac{\sigma^2}{3} - \frac{\sigma^4}{30} \cdots \frac{(-1)^{m+1}2m \cdot \sigma^{2m}}{(2m+1)!} + \cdots\right]$$

$$b = (1 - e^{-\zeta\alpha}) + e^{-\zeta\alpha}\left[\frac{\sigma^2}{6} - \frac{\sigma^4}{120} \cdots \frac{(-1)^{m+1}\sigma^{2m}}{(2m+1)!} + \cdots\right]$$

$$c = a + b$$

$$d = 1 - b$$

$$f = e^{-\zeta\alpha}\left[\sigma - \frac{\sigma^3}{6} \cdots (-1)^{m+1}\frac{\sigma^{2m-1}}{(2m-1)!} + \cdots\right]$$

$$N_1 = \frac{1}{M\nu^2}\left[a + \frac{2\zeta}{\alpha}c - (2\zeta^2 + \zeta\alpha)d\right]$$

$$N_2 = \frac{1}{M\nu^2}\left[b - \frac{2\zeta}{\alpha}c + 2\zeta^2 d\right]$$

$$N_3 = \Delta t \cdot d$$

$$N_4 = (\zeta\alpha) \cdot d + 1 - c$$

$$N_5 = \frac{1}{M\nu^2\Delta t}\left[-c + \frac{\zeta + \alpha}{\sqrt{1 - \zeta^2}} \cdot f\right]$$

$$N_6 = \frac{1}{M\nu^2\Delta t}\left[c - \frac{\zeta}{\sqrt{1 - \zeta^2}} \cdot f\right]$$

$$N_7 = 1 - c - \frac{\zeta}{\sqrt{1 - \zeta^2}} \cdot f$$

$$N_8 = -\frac{\nu}{\sqrt{1 - \zeta^2}} \cdot$$

For Δt, an increment of 1/6000 sec was employed. This corresponds to $\Delta\theta = 1$ cam deg at 1000 rpm, or 2 cam deg at 2000 rpm, and so forth.

Each term was evaluated to eight significant figures. This is the capacity of the automatic digital computer in which coefficients $N_1 \ldots N_8$ were used to solve equations 11, 12, and 13 for x. Since x is needed to only two significant-figure accuracy, the use of an 8-figure capacity automatic computer insures that only negligible cumulative error will result from applying the solution to 50 or 100 increments.

APPENDIX II

A Simple Means of Estimating Friction Factor φ

A simple static test may be made in order to establish the approximate magnitude of the load proportional friction factor φ. When gradually increasing *static* force is applied, mass M undergoes a displacement x towards its seat. The applied

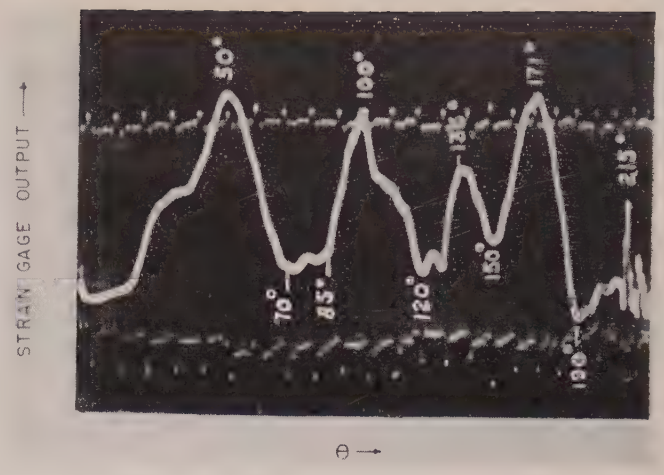


Fig. 19 – Oscilloscope record, with SR-4 strain gage on upper rocker arm (operating conditions same as in Fig. 15)

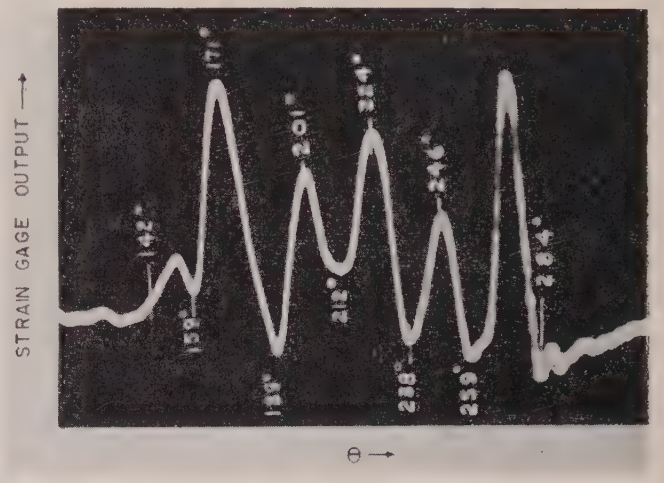


Fig. 20 – Same operating conditions as shown in Fig. 16 resulted in this oscilloscope record – SR-4 strain gage on upper rocker arm

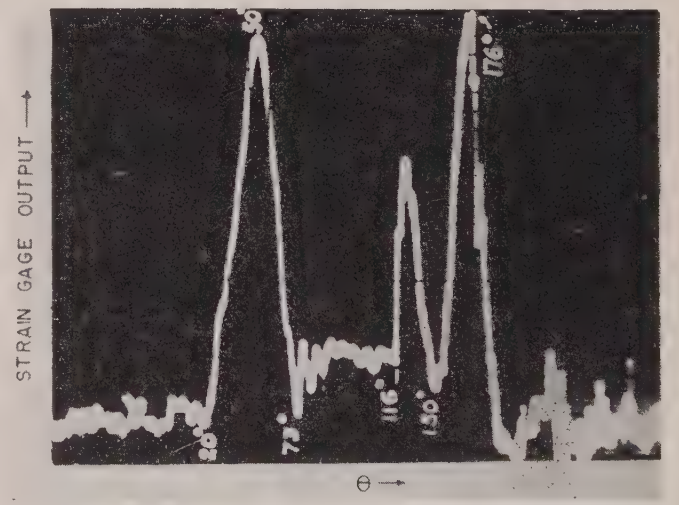


Fig. 21 – Another oscilloscope record with same operating conditions shown in Fig. 18 – SR-4 strain gage on upper rocker arm

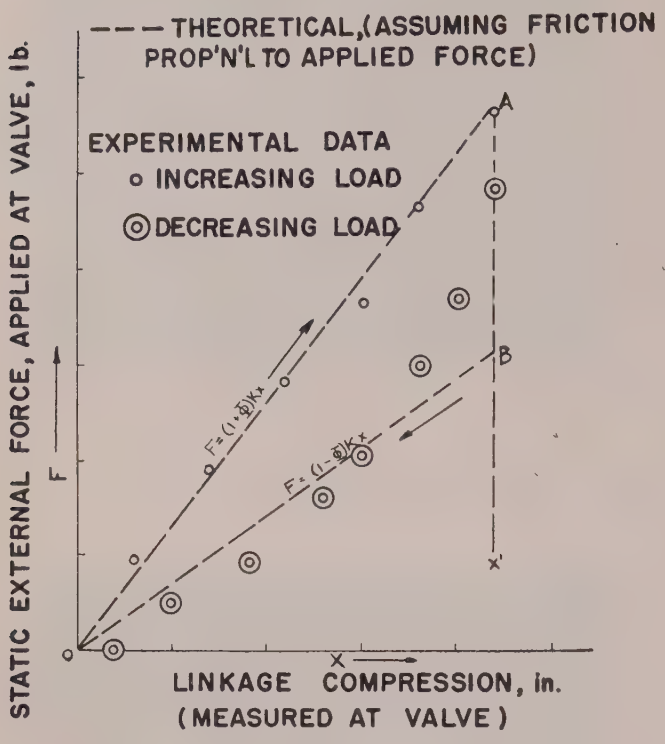


Fig. 22 – An idealized analysis is used in estimating friction factor φ

force, F, is opposed by a friction force, f, plus the linkage compressive force. Therefore:

$$F - f - Kx = 0 \tag{24}$$

With the linkage compressed, now reduce the applied force. Mass M will now move in the opposite direction and the friction force will also change direction. Therefore:

$$F + f - Kx = 0 \tag{25}$$

Adopting the assumption made previously that the frictional force is directly proportional to linkage load, let:

$$f = \Phi Kx \tag{26}$$

where Φ is now the *static* load proportional friction factor. Then substituting equation 26 in 24 gives:

$$F - (1 + \Phi)Kx = 0$$

and equation 25 becomes:

$$F - (1 - \Phi)Kx = 0$$

Plotting these two equations produces Fig. 22. According to this idealized analysis, the line OA is developed as load is applied. At A the load is gradually removed and curve ABO is followed as load is reduced. Line AB represents the region in which load is removed, but because of friction no displacement occurs.

The friction work W_f is the area OAB or:

$$W_f = \int_0^{x^1} \Big[(1 + \Phi)Kx - (1 - \Phi)Kx\Big]dx$$

$$= \Phi K(x^1)^2 \tag{27}$$

By integrating an experimental plot such as that formed by the points in Fig. 22, an experimental value of W_f is determined. Knowing W_f, Φ is determined from equation 27. Assuming a ratio $\varphi/\Phi = 0.5$ to 0.75 will probably result in a good approximation of φ. The values of empirical factors ζ and φ selected are best checked by calculating a known valve motion.

Acknowledgments

Grateful acknowledgment is made to Thompson Products, Inc. for financial support of this investigation, and to P. H. Schweitzer and A. W. Hussmann of the Pennsylvania State College for their advice and encouragement.

This paper is based on a thesis submitted to the Pennsylvania State College, department of mechanical engineering, in partial fulfillment of the requirements for the degree of doctor of philosophy.

Additional References

1. "Poppet Valve Dynamics," by E. H. Olmstead and E. S. Taylor. *Journal of the Aeronautical Sciences,* Vol. 6, July, 1939. pp. 370-375.
2. "Investigation of Actual Valve Motion of a Poppet Valve Engine," by R. M. Schutte. Master of Science thesis, Case Institute of Technology, May, 1950.
3. "New Methods in Valve Cam Design," by W. M. Dudley. SAE *Quarterly* Transactions, Vol. 2, January, 1948, pp. 19-33.
4. "Cam Design as Related to Valve Train Dynamics," by T. R. Thoren, H. H. Engemann, and D. A. Stoddart. SAE *Quarterly* Transactions, Vol. 6, January, 1952, pp. 1-14.
5. "An Analysis of the Dynamic Forces in a Cam-Driven System," by J. A. Hrones. ASME Transactions, Vol. 70, July, 1948, pp. 473-482.
6. "Vibrations in Valve Mechanisms," by T. Warming. Proceedings of the National Oil and Gas Power Conference, ASME, 1949.
7. "Schwingungen in Schraubenförmigen Ventilfedern," by A. W. Hussmann. *Jahrbuch der Deutschen Luftfahrtforschung,* Part II, 1938, pp. 119-133.
8. "Vibrations in Valve Springs for Internal-Combustion Engines," by W. Marti. *Sulzer Technical Review,* No. 2, 1936. Originally published more fully as "Ventilfeder-Schwingungen," thesis for Doctor of Technical Science, Eidgenossische Technische Hochschule, Zurich, Switzerland, 1936.
9. "Tests on Dynamic Response of Cam-Follower Systems," by D. B. Mitchell. *Mechanical Engineering,* Vol. 72, June, 1950, pp. 467-471.

Designing the Cam Profile for Low Vibration at High Speeds

R. A. Roggenbuck, Ford Motor Co.

Paper presented at SAE Nat'l Passenger-Car, Body, and Materials Mtg., Detroit, March 5, 1953.

A CAM lift curve for an internal-combustion engine is necessarily determined by factors such as maximum lift, tappet size, tappet face stress, and timing of total valve opening. These limitations force the designer to use curves that are not easily expressed mathematically. To reduce these arbitrary cam shapes to a form more easily analyzed, several methods have been developed in the past few years in which the desired lift curve is approximately described by a simple algebraic or trigonometric expression. No doubt the best-known of these is the polynominal method. When the lift curve is thus described, determination of velocity, acceleration, and shock factor is readily made by simple substitution into formulas. These methods certainly put a cam profile into a form in which it can be handled readily, but they cannot be used to predict completely the valve vibration which will be present when a particular cam is used, since there is no information given as to the response of the rest of the valve train.

Control of the motion of a valve by means of forces applied to a cam and transmitted through a tappet, push-rod, and rocker arm can be compared, in an exaggerated way, to control of the motion of a weight hanging at the end of a weak spring with the other end of the spring being supported by hand. The weight does not follow the motion of the hand directly, but goes through a vibratory motion of its own. In a similar manner the valve moves in the same general direction as the cam directs, but it, too, vibrates throughout the valve-open period. As long as the vibratory accelerating forces on the valve do not overcome the valve-spring force, the train holds together and there is no trouble from valve tossing.

A typical overhead-valve mechanism is shown in Fig. 1. It can be considered as consisting of four springs and three masses, which are as shown.

This system can be simplified to that of a single mass and spring, as shown in Fig. 2, and the differential equation of motion of this combination is as shown in this illustration (equation 2-1). When this differential equation is solved for X, it will produce a curve which will represent the motion of the valve when any arbitrary input curve Y is introduced at the cam end of the valve train. A study of such resultant curves was undertaken in the Engineering Research Department at Ford Motor Co., with a view toward more fully understanding the influence of the cam profile in exciting valve train vibrations.

Hand solution of a differential equation such as this requires a tremendous amount of desk-calcu-

THE vibration of overhead-valve trains at high engine speeds can be traced to cam profiles.

The author presents a technique that provides rapid approximate predictions of results with contemplated profiles. This method employs automatic computing machines to make the necessary calculations.

Discussion of this and other papers on "Valve-Gear Problems in Modern Overhead-Valve Engines" starts on page 714.

R. A. ROGGENBUCK is with the Engineering Research Department of Ford Motor Co. At present he is in charge of a group investigating mathematical techniques for solving design problems. He received his B.E.E. degree from Lawrence Institute of Technology in 1940, and upon graduating joined the Production Engineering Department of Ford.

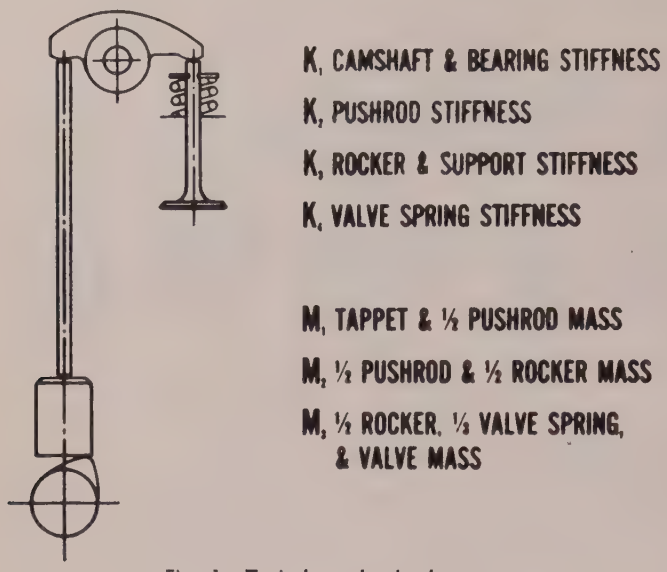
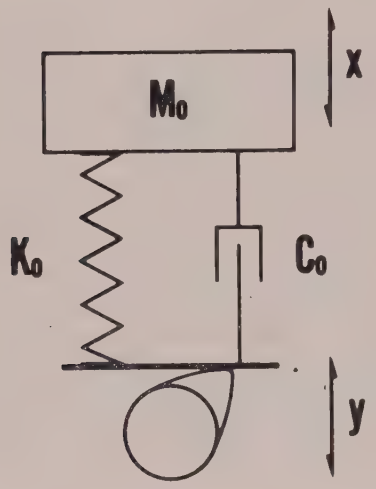


Fig. 1 – Typical overhead-valve gear



$$M_o \ddot{x} + C_o \dot{x} + K_o (x-y) = 0 \quad \text{2-1}$$

$$\ddot{x} + 3378\,\ddot{x} + .1875 \times 10^8 (\dot{x}-\dot{y}) = 0 \quad \text{2-2}$$

Fig. 2 – Simplified diagram of Fig. 1 and differential equations of motion

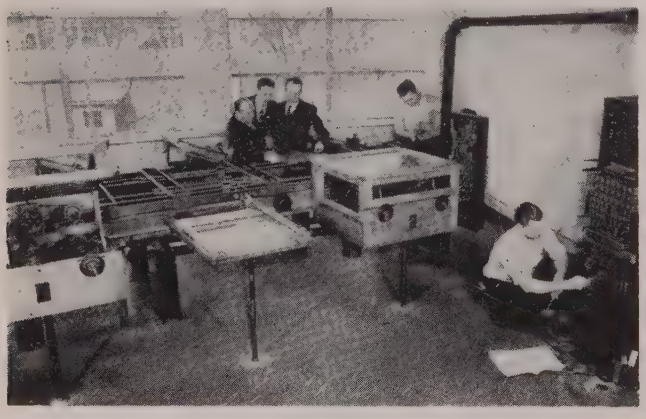


Fig. 3 – Wayne University differential analyzer

lator time. Consequently, we sought out the assistance of the Computation Laboratory at Wayne University and, with the help of their staff, the mechanical differential analyzer was put to work on this problem. A general view of this machine is shown as Fig. 3. The machine is essentially a mechanical solver of differential equations, and is capable of furnishing answers in the form of curves, as desired.

The differential equation in the form in which it was actually put on the machine is as shown in Fig. 2 (equation 2-2). Note that the input function is now the cam acceleration (y with two dots over it) and the output or answer is the valve acceleration (x with two dots over it). The reason for working with acceleration rather than lift is that the effects of changes are much more readily observed when an acceleration base is used.

The valve acceleration diagrams shown here are from only a few of the large number of runs that were made. These are, no doubt, the most interesting from the standpoint of conclusions that can be drawn.

Fig. 4 shows the acceleration curve for an experimental cam that was giving serious trouble at 3000 engine rpm. Note particularly the facts that the rate of rise of acceleration is high, the peak acceleration is high, and the positive acceleration portion of the curve is not very wide. Also on this chart is the valve acceleration diagram for this cam at 3000 engine rpm. The wide sweeps in valve acceleration force obviously overcome the valve-spring force, so the train falls apart and the cam loses control.

Fig. 5 is the diagram for a successful cam, which can be run up to speeds of 4400 engine rpm, and possibly higher, without breakdown. Note that the rate of rise is lower, the peak force is lower, and the width is much greater. The valve acceleration diagram for this cam run at the equivalent of 4260 engine rpm is shown here.

Fig. 6 is a diagram for a cam designed to have about the same lift as that of Fig. 5, but using rectangular sides in all portions of the acceleration pattern. This was set up as a sort of arbitrary standard for comparison. The indicated valve pattern for this cam at 4260 rpm is shown here. Note that the amplitude of the vibrations is about the same as with the cam of Fig. 5.

Sloping the sides of the positive acceleration portion provides the results shown in Figs. 7 through 9. It can be seen that, theoretically at least, side sloping does not have an overwhelming effect on reducing amplitudes, when a constant deceleration pattern is used.

Fig. 10 is a design in which the width of the main acceleration portion is such that at 4260 rpm it sweeps past in an amount of time equal to one-half cycle of valve train natural frequency. The actual lift for this cam would be 0.254 in. The valve acceleration pattern is as shown. Fig. 11 is a

cam whose acceleration width is equivalent to one full cycle at 4260 rpm. The lift for this cam is 0.425 in. The valve acceleration pattern is shown on this chart. Note that the accelerating force merely oscillates slightly around the zero line after so large a disturbance.

Many other cam patterns were run in this study. All of this work was based on acceleration patterns that would give approximately the same lift and overlap characteristics as standard production cams. Changes were made in the following items:

1. Amount of time spent on the ramp.
2. Height and width of rectangular acceleration periods.
3. Height and width of rectangular deceleration periods.
4. Slope of sides of both acceleration and deceleration periods.

Valve acceleration diagrams were made for engine speeds of 2960, 3560, and 4260 rpm, and for valve-gear designs having natural frequencies of 685 (standard), 500, and 790 cps. There were approximately 300 individual runs made and the sheer volume of the information prevents it from being presented at this time.

It is to be noted that the damping coefficient in these solutions was assumed to be constant at 3/16 lb per in. per sec. This was found on the differential analyzer to give a predicted curve of valve acceleration force for a standard cam which followed reasonably closely to an acceleration diagram obtained from an actual engine.

It is not a simple matter to form many hard and fast rules on cam design from these data, but the following general rules are apparent:

1. The most important single criterion is the relationship between maximum running speed and the time for one full cycle of valve-gear natural frequency. The valve train will be more or less tolerant of the form of the acceleration pattern as long as there is no running speed at which the width of the pattern approaches the time occupied by one-half cycle.

2. As a logical conclusion from the above, it is extremely desirable that the natural frequency of the valve train be as high as possible. There is a further note on this later in this paper.

3. Patterns to be avoided are those pictured in Fig. 12. These are bad chiefly because they utilize a high peak force at some portion of the cycle to compensate for loss of it at other portions.

4. The pattern shown in Fig. 5 is good because it meets proper natural frequency requirements in addition to having sloped sides on the acceleration and deceleration periods.

5. The pattern of Fig. 5 has only one disadvantage: The valve opens somewhat later than when a constant-acceleration cam is used. A compromise design, which has been checked on the differential analyzer, has proved to have a sufficiently high

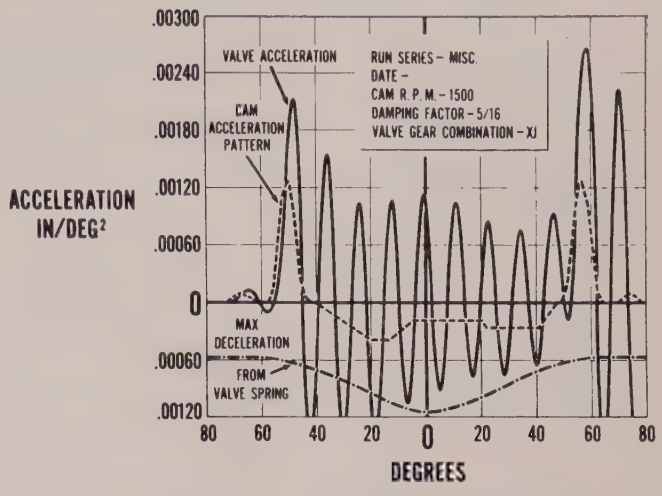


Fig. 4 – Acceleration output chart for cam with poor high-speed characteristics

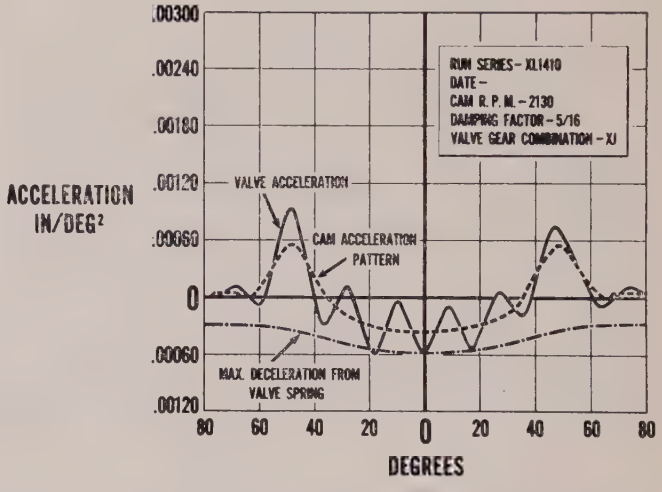


Fig. 5 – Acceleration output chart for cam with much improved high-speed characteristics

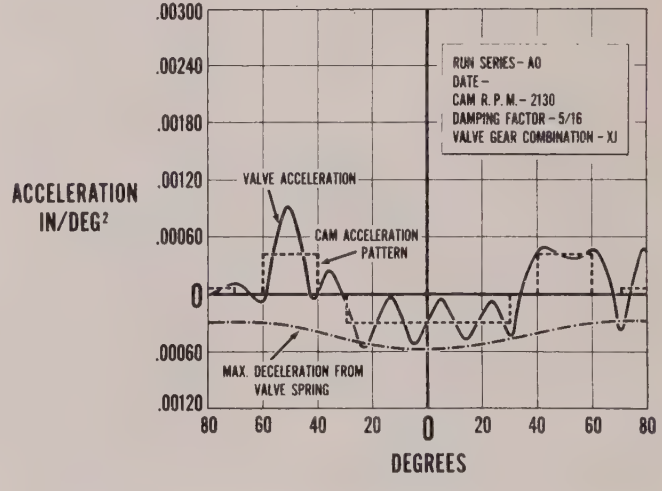


Fig. 6 – Acceleration output chart for constant-acceleration cam

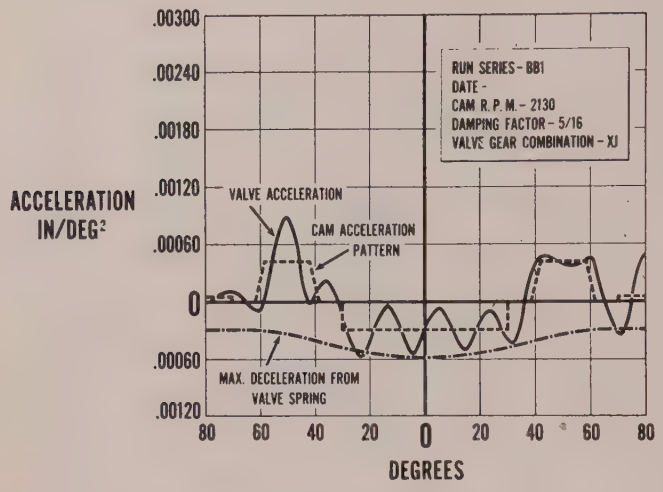


Fig. 7 – Acceleration output chart – positive acceleration portion has sloping sides

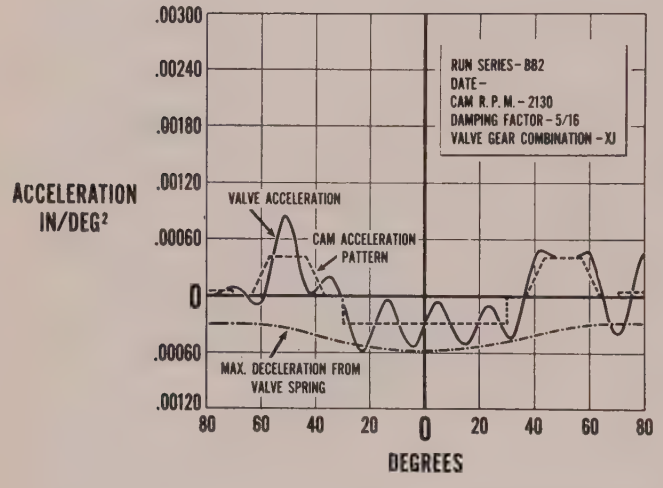


Fig. 8 – Acceleration output chart – positive acceleration portion has sloping sides

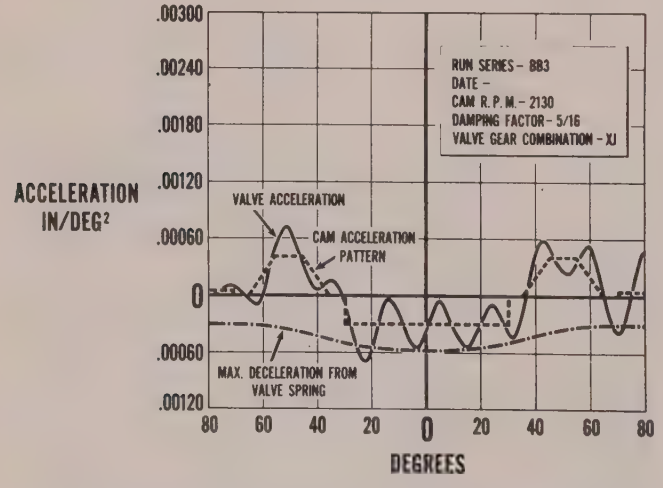


Fig. 9 – Acceleration output chart – effect of sloping the side of acceleration period but maintaining constant deceleration

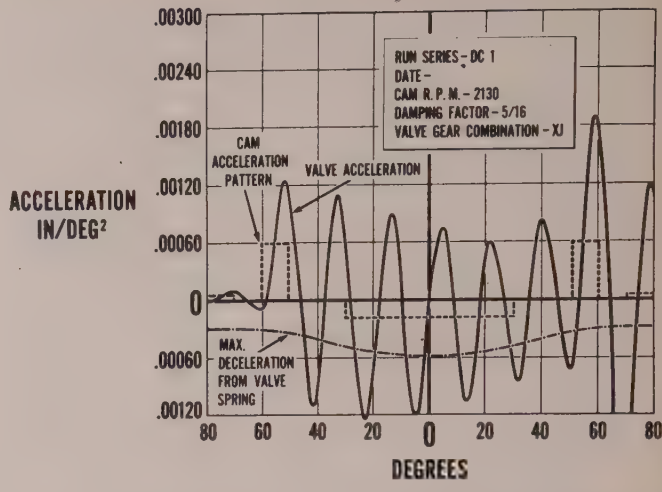


Fig. 10 – Acceleration output chart – cam designed to have width of main acceleration period equal to one-half cycle of valve-gear frequency at 4260 engine rpm. Lift for this cam is 0.254 in.

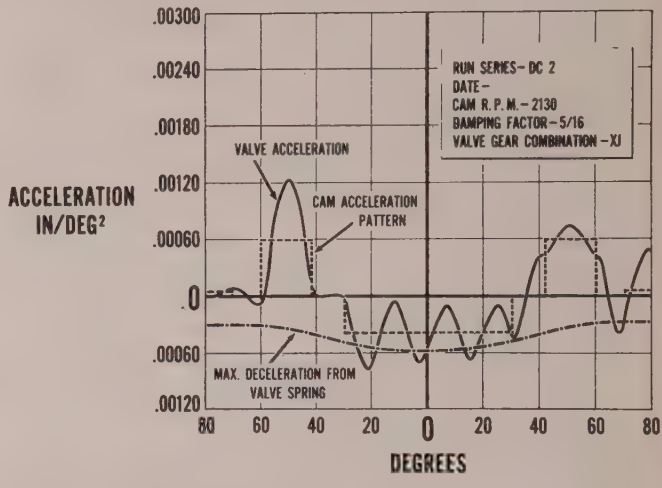


Fig. 11 – Acceleration output chart – cam designed to have width of main acceleration period equal to one full cycle of valve-gear frequency at 4260 engine rpm. Lift for this cam is 0.425 in.

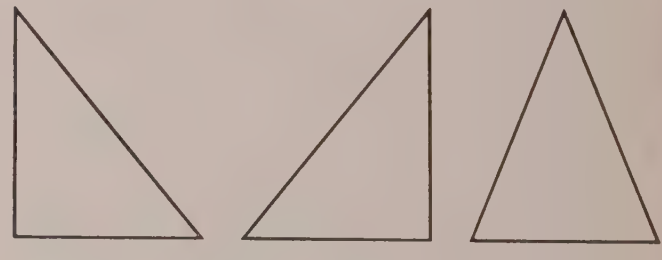


Fig. 12 – Examples of acceleration patterns to be avoided

Table 1 – Effect on Valve-Train Natural Frequency of Indicated Changes in Stiffness or Mass. All Quantities Are Held at "Standard" Except the One Being Investigated

	20% Increase	20% Decrease	Difference
K_1	680	660	20
K_2	700	640	60
K_3	690	650	40
K_4	670	670	0
M_1	670	670	0
M_2	670	680	10
M_3	605	740	135

no-follow speed that it is used in the current Lincoln engine. It is shown as Fig. 13.

6. The valve-spring-force curve represents the maximum deceleration that can be applied to the valve by the valve spring. As running speed is increased, the distance between the theoretical valve acceleration curve and this force curve grows less by a factor proportional to the square of the speed. This means that wide sweeps in the acceleration pattern may be permissible at low engine speeds, but these must become smaller as the speed is raised.

We are expanding this study of cam design in two ways:

1. A more accurate representation of valve gear is being made on an analog machine. This includes a full 3-mass system, loss of input during valve toss, motion of the valve during tossing, and impact in the system when the cam regains control.

2. A thorough study of damping and ways of representing it on an analog is being planned.

Influence of Various Parts on Natural Frequency

It is desirable to keep the natural frequency of a valve train as high as possible. The question naturally arises as to what is the best way to do this. To answer this question for ourselves we set up our own analog equipment to represent the present-production Lincoln valve train. This particular analog uses inductance coils and condensers to represent stiffness and mass and can be readily set up to determine natural frequencies of certain types of vibration problems, including those of valve gear. It is shown in Fig. 14.

Changes in stiffness or mass are readily made by turning dials, so that design studies can be made in a short time.

Table 1 shows the effect upon the natural frequency (670 cps with standard parts) when the changes listed are made. It can be seen that the mass of the valve has by far the greatest effect, with push-rod stiffness, rocker stiffness, camshaft stiffness, and mass at the upper end of the push-rod following in that order. Stiffness of the valve spring and mass of the tappet had no noticeable effect.

This work is a good demonstration of the value of analog methods. To study the characteristics manually of even as few cams as are reported here

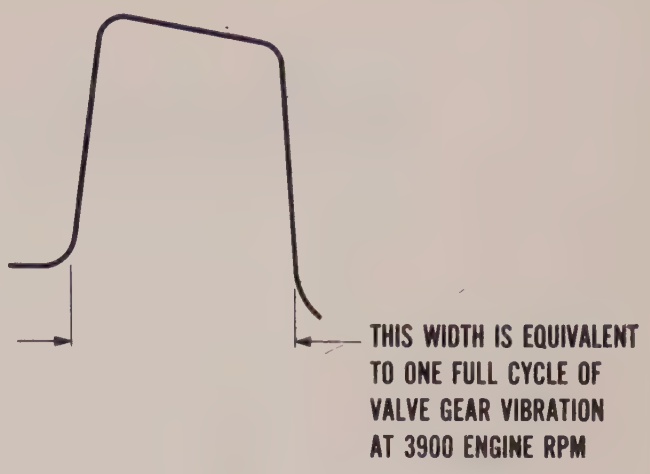


Fig. 13 – Cam acceleration pattern as used in Lincoln engine

would have been a job of several months' duration. Actually to make these cams and try them in engines would have been ridiculous. Yet, when the machine was set up, solutions were made in an average of one-half hour each. The method is so simple to use that it would seem entirely practicable to run an analysis of this nature for any proposed cam, assuming that analog equipment is available to the cam designer.

The author wishes to express his sincere appreciation to Dr. Arvid Jacobson and other members of the staff of the Wayne University Computation Laboratory for their assistance on this project.

Fig. 14 – Ford research vibration analog

Relationship of Valve-Spring and Hydraulic

THE valve spring is a vital component part of the valve gear of an internal-combustion engine. Its prime purpose is to keep the system intact during the periods of negative acceleration that exist during the interval when the cam follower is riding the nose of the cam. However, under conditions of valve gear vibration at high engine speeds, negative accelerations can occur during any portion of the valve lift cycle, meaning that negative accelerations can occur when the valve is almost seated with the spring extended and low on load. Therefore, the success of a valve-spring design is intrinsically related to both the cam design and the resulting valve gear dynamics obtained at high engine speeds.

In the L-head engine, where little valve gear vibration is generally experienced, the loads to be provided by the spring can be established from the theoretical negative acceleration requirements at high speeds. Only a small margin of excess spring load need be provided to take care of the load-loss occurring during periods of spring surge at high speeds. In the case of overhead valve gear and especially where hydraulic lifters are employed, a much greater margin of excess spring load must be provided. Here, in addition to counteracting the negative acceleration force and allowing for spring surge load-loss, additional force must be available to keep the valve gear intact when valve gear vibration occurs. This is

---

THE author shows how the success of a valve-spring design is intrinsically related to both the cam design and the valve gear dynamics obtained at high engine speeds.

Good valve gear dynamics, which is characterized by minimum vibration, he says, minimizes hydraulic lifter pump-up tendency and greatly simplifies the job of making a satisfactory spring design.

He shows that the use of the smooth-acceleration curve is very helpful in producing good valve gear dynamics, and that it is to be recommended over other types.

The author also discusses the use of dual springs and cyclo-pelting and presetting of springs.

Discussion of this and other papers on "Valve Gear Problems in Modern Overhead-Valve Engines" starts on page 714.

The Author

M. C. TURKISH (M '41) is chief engineer, Coil Spring Department, Spring Division of Eaton Mfg. Co. He received his B.S.M.E. from Wayne University in 1938.

Design to Valve Gear Dynamics Lifter Pump-up

M. C. Turkish, Eaton Mfg. Co.

Paper presented at SAE Nat'l Passenger-Car, Body, and Materials Mtg., Detroit, March 5, 1953.

important to avoid excessive loading of component parts and to prevent incipient jump or bounce from occurring, which results in hydraulic lifter pump-up. Pump-up of hydraulic lifters occurs when an incipient jump or bounce occurs during the valve lift cycle. This permits the hydraulic lifter to expand to eliminate all internal lash in the system. At the end of the lift cycle the valve is not fully closed but remains slightly raised off of the valve seat. If the pump-up tendency is severe, the valve will be raised off of the seat during the entire base circle interval, resulting in considerable loss in engine power and reduced valve life due to leakage at the valve seat.

Good valve gear dynamics, characterized by a minimum of vibration, minimizes hydraulic lifter pump-up tendency and greatly simplifies the job of making a satisfactory spring design. For a study of the problem of dynamics one must consider the equivalent vibratory system of a valve gear, as shown in Fig. 1. A valve gear layout is illustrated at A, showing W_v as the effective weight of the vibratory system and K_2 as the effective valve gear rate. The equivalent vibratory system for the first mode of vibration is shown in B. By knowing the magnitude of both W_v and K_2 (both referred to either the valve side or the push-rod side), one can determine the natural valve gear frequency from the following formula when the damping factor is neglected:

$$F = 188\sqrt{\frac{K_2}{W_v}}$$

where:

K_2 = Effective valve gear rate, lb per in.

W_v = Effective valve gear weight (above midpoint of push-rod), lb

A knowledge of this frequency is essential to the study of valve gear dynamics. A simplified method for measuring the natural valve gear frequency directly on an engine is shown in Fig. 2. This incorporates the use of a magnetic velocity transducer mounted on the engine. A small magnet, which is attached to the valve, produces an output voltage proportional to the velocity. A simple electrical differentiating network can be used to obtain an acceleration curve. With the valve gear stationary and the valve in a raised position, a shim of approximately 1/16 in. is inserted between the valve stem and the rocker arm and suddenly removed. With the output voltage from the velocity transducer feeding to a cathode ray oscillograph, a transient velocity or acceleration curve is traced upon the screen. By comparing the frequency of this transient vibration to a 60-cps timing curve, one can obtain the valve gear frequency directly.

An alternate approach to measuring the valve gear frequency is to use the instrumentation shown in Fig. 2 to obtain an actual acceleration curve with the valve gear in operation. By obtaining a relatively low-speed acceleration curve as shown in Fig. 3, one can observe the cyclic valve gear vibration. By counting the number of cycles during the lift interval and comparing it to a theoretical cam curve, or better still to a 60-cps timing curve, the frequency can be determined. For best accuracy only the cycles occurring during the negative acceleration interval should be counted. This is especially true at high engine speeds, when the cycles during the positive acceleration intervals are forced and will give a lower frequency.

In studying valve gear dynamics it is of interest

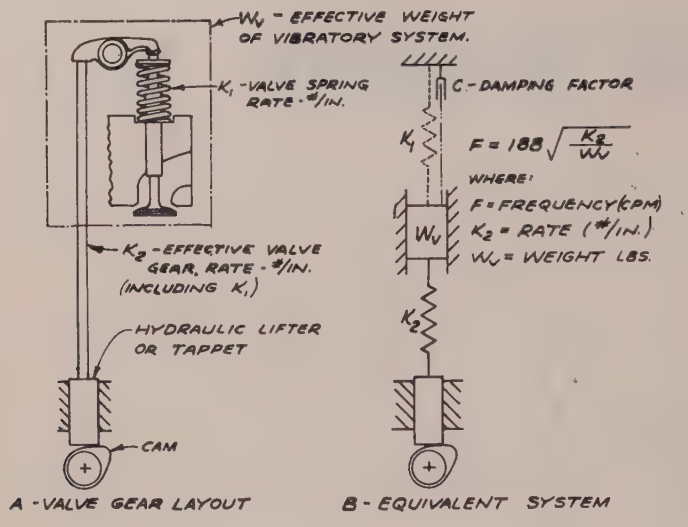


Fig. 1 – Equivalent system of valve gear for first mode of vibration

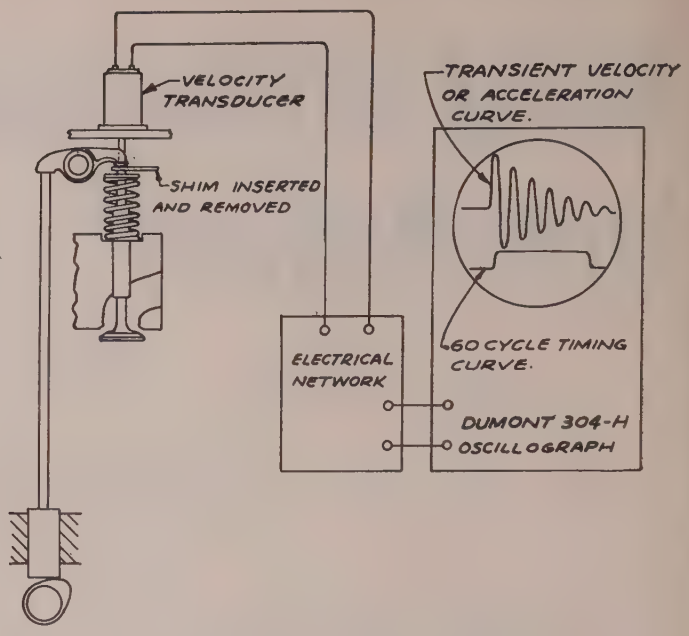


Fig. 2 – Measuring valve gear frequency

to determine the effect of a particular valve gear frequency when used with different types of acceleration curves. Also, the effect of different valve gear frequencies can be determined and how it reflects into the requirements of the valve spring. In Fig. 4 two sets of lift and acceleration curves are presented. The valve life curves are essentially identical. The difference lies in one cam having a smoothly changing acceleration curve throughout the entire valve lift cycle, whereas the other has constant positive acceleration intervals with rapid changes in acceleration. In order to obtain the same lift curve the smooth type of acceleration curve will require a greater positive acceleration interval (27 deg versus 18 deg in this example) and will have a higher maximum acceleration value. Using these two extreme types of acceleration curves at the cam, the response acceleration

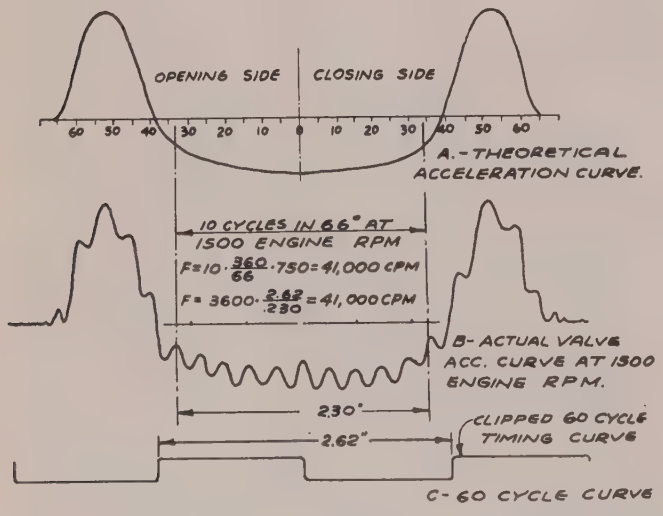


Fig. 3 – Determining valve gear frequency from low-speed acceleration curve

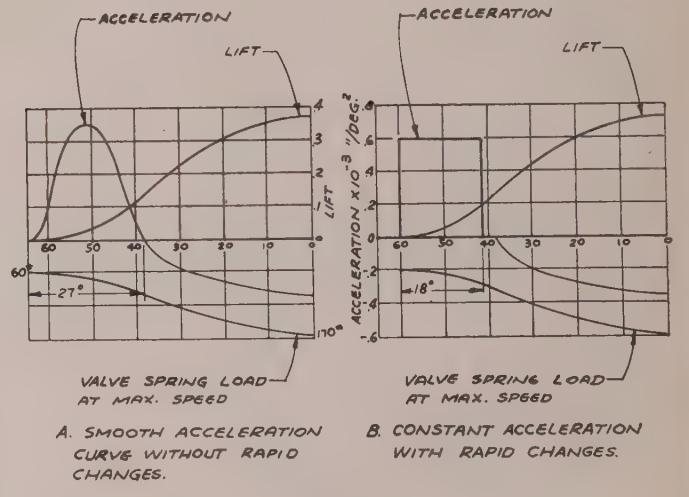


Fig. 4 – Acceleration curve comparison

curves at the valve can be determined analytically. Using a polynominal equation for the smooth-acceleration curve in the form of:

$$Y_c = C_2\theta^2 + C_3\theta^3 + C_4\theta^4 + \cdots$$

the response acceleration will be:

$$Y_v = C_2\theta^2 + C_3\theta^3 + C_4\theta^4 + (2C_2 - 24C_4\phi^4)\cos\psi + 6C_3\phi^3\sin\psi - 2C_2\phi^2 - 6C_3\phi^2\theta - 12C_4\phi^2\theta^2 + 24C_4\phi^4 + \cdots$$

where:

C_2, C_3, C_4 = Constants establishing size of acceleration curves

θ = Cam angle, deg

W_c = Cam speed, rpm

W_n = Valve gear frequency, cpm

$$\phi = 57.3\left(\frac{W_c}{W_n}\right), \text{ deg}$$

$$\psi = 57.3\left(\frac{\theta}{\phi}\right), \text{ deg}$$

Similarly, for the constant-acceleration curve:

$$Y_c = C_1$$
$$Y_v = C_1(1 - \cos\psi)$$

The acceleration response curve is related to the number of full cycles of natural valve gear vibration occurring during the acceleration time interval at the particular operating speed. Defining K as the number of free vibration cycles, response curves can be plotted for various valve gear frequencies. In Fig. 5 acceleration response curves are shown for an engine speed of 4200 rpm using a valve gear frequency of 42,000 cpm. This gives a K value of 1.5 for the smooth-acceleration curve and 1.0 for the constant-acceleration curve. The theoretical response is shown, with the constant-acceleration curve giving better results in this instance. Damping modifies the response curves somewhat, as shown. The magnitude D of the resulting negative acceleration approaches the available spring-load curve. For the smooth type of acceleration curve using this K value of 1.5, as severe a response as can be tolerated is obtained with average spring loads without getting into problems of jump and hydraulic lifter pump-up.

Similarly, acceleration response curves can be shown for valve gear having different frequencies. Response curves at the same engine speed of 4200 rpm are given in Fig. 6 for high-frequency valve gear. With high-frequency valve gear the smooth-acceleration curve gives improved valve gear action. In the case of the constant-acceleration curve a large amount of valve gear vibration occurs even with a valve gear of higher frequency. This vibration occurs whenever a low number of an odd-multiple of half-waves of vibration can occur during the acceleration time interval. In this instance, with $K = 1.5$ the negative acceleration exceeds the spring-load curve, with valve gear jump occurring. Acceleration response curves for a valve gear having a low frequency are shown in Fig. 7. Here the magnitude of D is very high for

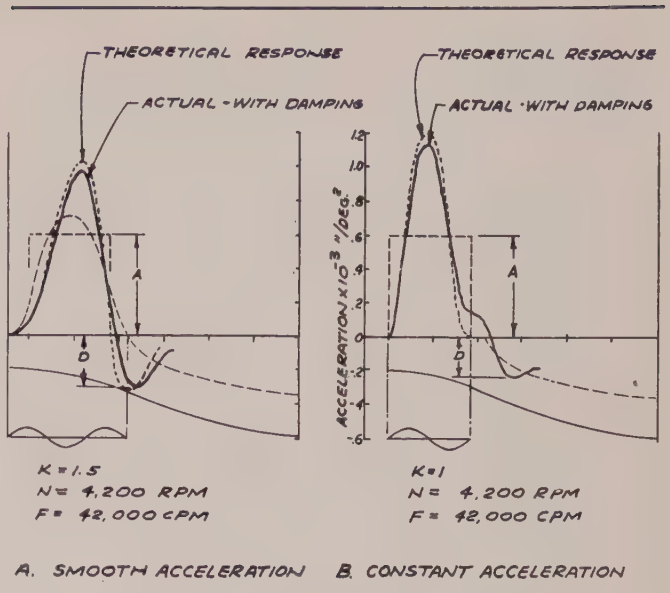


Fig. 5 – Acceleration response curves

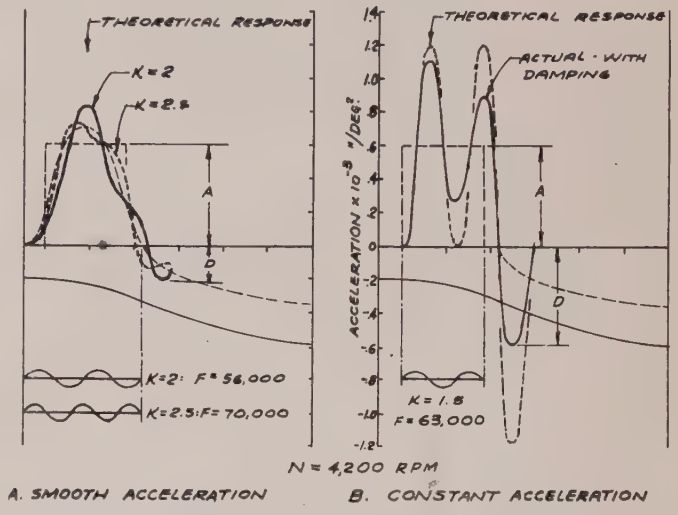


Fig. 6 – Acceleration response curves for valve gear having high frequency

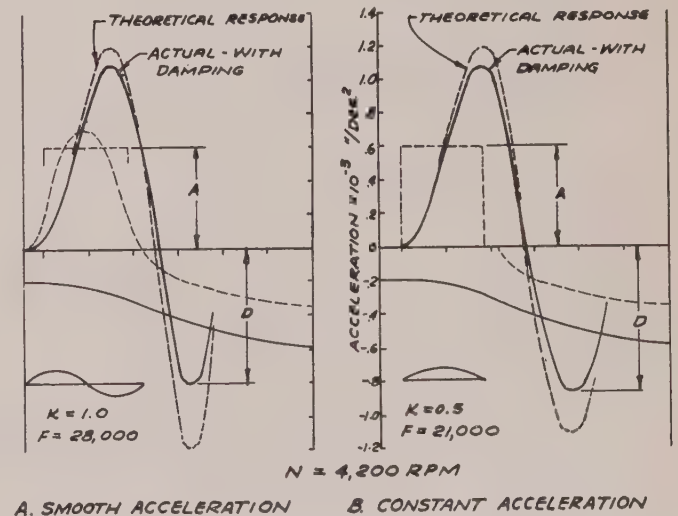


Fig. 7 – Acceleration response curves for valve gear having low frequency

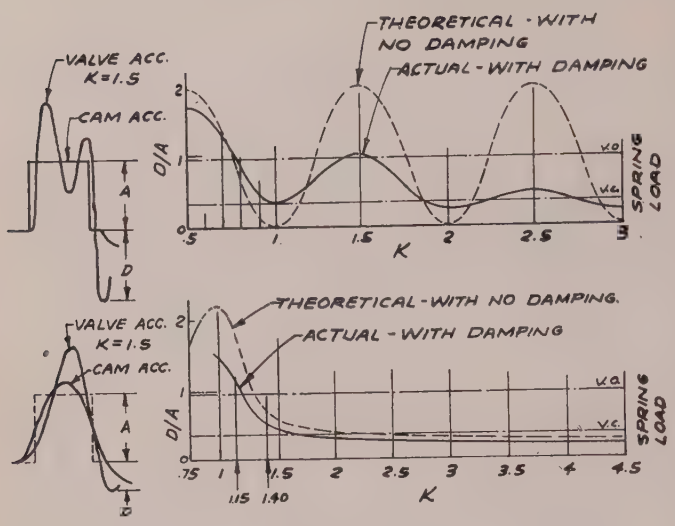


Fig. 8 – Acceleration response amplitude for different valve gear frequencies

N_1 = LOWER CRITICAL ENGINE SPEED
N_2 = UPPER CRITICAL ENGINE SPEED
θ = DEGREES OF CAM ACCELERATION
F = VALVE GEAR FREQUENCY - CPM

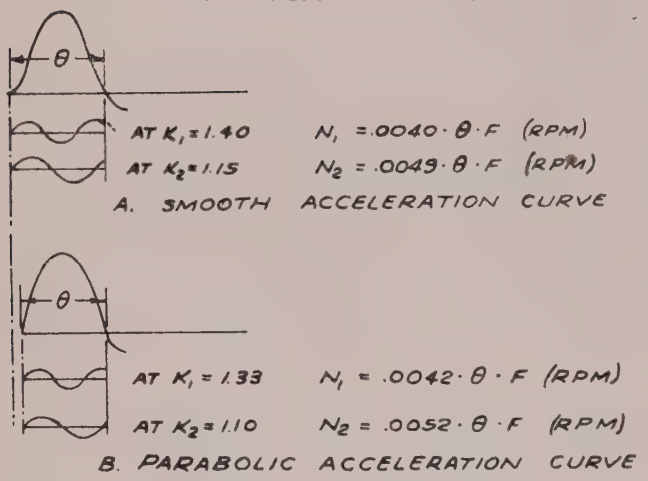


Fig. 9 – Formulas for critical engine speeds

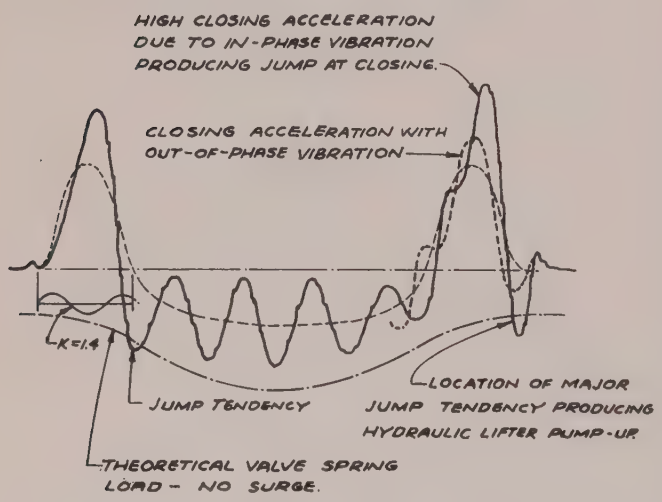


Fig. 10 – Effect of valve gear vibration on closing acceleration

both types of cams. This shows it to be impractical to attempt to operate a valve gear with a K of 1.0 or less with a smooth-acceleration curve and in the region of $K = 0.5$ for the constant-acceleration curve.

A summary of the acceleration response curve amplitude is presented in Fig. 8. The ratio D/A is plotted for various values of K for both types of acceleration curves. This represents a plot of the maximum increase of negative acceleration due to vibration, with and without damping. The magnitude of the K abscissa differs for the two plots to make them more directly comparable. For example, in the comparison made in this report of the two types of acceleration curves, $K = 1.5$ for the smooth-acceleration curve is comparable to $K = 1.0$ for the constant-acceleration curve. This is for the same type of lift curve, the same engine speed, and the same valve gear frequency. The constant-acceleration curve produces a marked variation in response depending upon the value of K. For the smooth-acceleration curve no prominent critical speeds occur with K values of 1.5 or higher. Below this value of K, trouble will be experienced similar to that encountered with the constant acceleration with a K value below 0.8.

From this analysis and from verification with actual engine tests, it is apparent that the use of the smooth-acceleration curve is very beneficial in producing good valve gear dynamics. It also reveals the problems that will be encountered by using the constant-acceleration curve at the cam, especially when operating with K values of 0.5 or 1.5. Therefore, the use of the smooth type of acceleration curve, or some modification of it, is recommended over other types. It was first introduced by J. D. Turlay of the Buick Motor Division in 1935 and has since been continually used in production in the Buick engine. Today, all of the popular V-8 overhead valve gear engines use cam designs similar to or of this type.

In summing up the data just outlined on response curves, it can be stated that critical engine speeds occur when the negative acceleration inertia load exceeds the valve-spring load, even though this may occur only momentarily. For the smooth type of acceleration curve this region occurs when K values are less than 1.50. Selecting a lower critical speed N_1 as an engine speed where valve gear dynamics tend to cause a jump with normal valve spring loads and N_2 as an upper critical speed where valve gear dynamics becomes quite severe and even high valve-spring loads do not remedy hydraulic lifter pump-up, the formulas in Fig. 9 can be simply derived. With a smooth-acceleration curve having a K value of 1.40, the lower critical speed is equal to the product of 0.0040, the degrees of cam acceleration, and the natural valve gear frequency. Selecting a K value of 1.15 for the upper critical engine speed, then the upper critical speed is equal to the product of 0.0049, the degrees

of cam acceleration, and the natural valve gear frequency. Similarly, formulas are developed for the parabolic type of acceleration curve and given in Fig. 9. This type of acceleration curve is quite satisfactory and is comparable in performance to the smooth-acceleration curve, especially for K values above 1.5. It is often preferred in cases where the total cam angle event is limited, due to engine idle characteristics.

The critical speeds derived from the formula in Fig. 9 are intended to serve as a guide in matching cam designs with existing valve gear frequencies. Obviously, the lower critical speed N_1 can be exceeded by using excessively high valve-closed loads and using low acceleration values. However, above this speed, and up to the higher critical speed, other critical speeds will appear, due primarily to the phasing between the vibration ensuing from the opening acceleration with the closing acceleration, as shown in Fig. 10. Since hydraulic lifter pump-up generally occurs due to high negative acceleration at the point of valve closing, critical pump-up speeds will appear when operating with low K values and experiencing in-phase vibration. Other factors will enter, such as differences in the K values of the various valve gear, torsional camshaft vibration, and valve-spring surge.

For high-speed passenger-car engines valve gear frequencies above 40,000 cpm are very desirable, with 50,000 cpm being sought after. With a cam angle of 27 deg, a lower critical speed of 4300 rpm is obtained for the lower frequency and 5400 rpm for the higher frequency. To keep the spring problem as simple as possible, the lower critical speed should be above 4200 rpm. This will permit attaining speeds as high as 4800 to 5000 rpm without hydraulic lifter pump-up by employing suitable valve-spring designs.

Valve-spring surge or vibration is caused by an harmonic excitation of the cam exciting the natural valve-spring frequency. This surge is undesirable since it diminishes the available static spring load intended to keep the valve gear intact and it increases the maximum stress in the spring. It is known to be related to the harmonic number at which it is operated, as shown in Fig. 11. This harmonic number is defined as the spring frequency divided by the cam rpm. The amplitude of spring vibration is directly related to the amplitude of the exciting cam harmonic. An harmonic analysis of a low-speed valve lift curve is shown. It is observed that in this case the ninth order has no amplitude. However, at high speeds with the presence of valve gear vibration, the harmonic excitation is altered, due to the distortion in the lift curve. A typical harmonic analysis of a high-speed valve lift curve, having only little distortion with no actual jump, is shown. Here the ninth order attains appreciable magnitude. One can, therefore, conclude that in general the lower the harmonic number the greater will be the spring surge amplitude. Also, the best way to keep surge to a minimum is to operate at high harmonic orders by using a relatively high valve-spring frequency.

Analogous to the transient vibrations in valve gear, transient vibrations occur in springs at high engine speeds. This is illustrated in Fig. 12. After

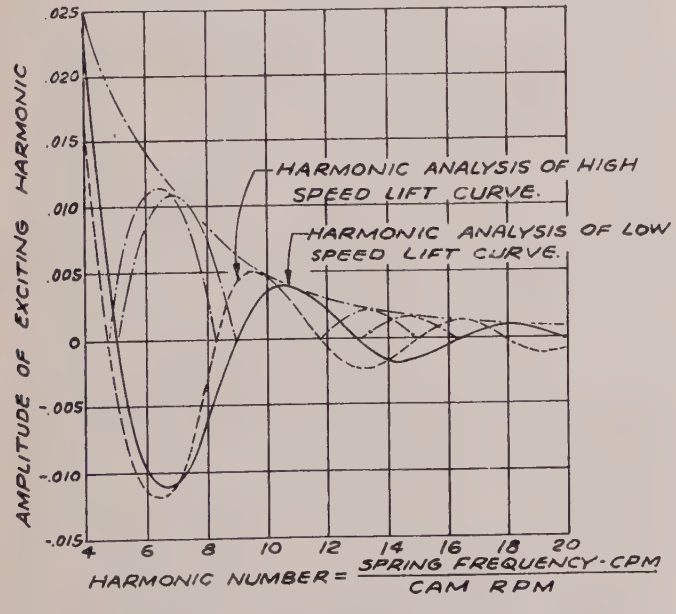


Fig. 11 – Spring vibration excitation versus harmonic number

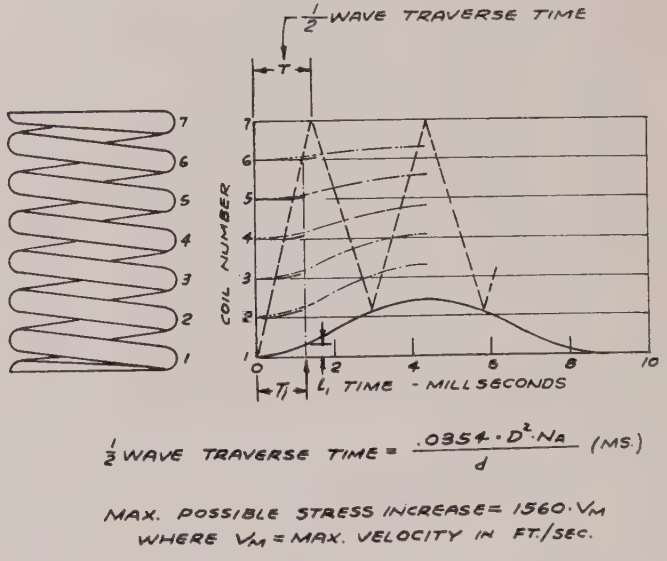


Fig. 12 – Transient vibrations in springs

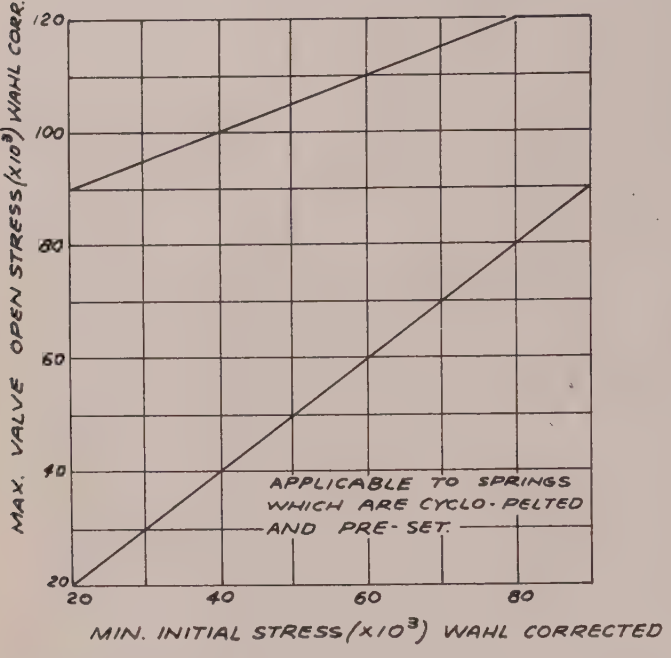


Fig. 13 – Stress diagram for valve springs using pretempered valve-spring carbon wire

$$F = 531\sqrt{\frac{R}{W_A}}$$

$$\text{OR } F = \frac{764{,}000 \times d}{D^2 \times N_A}$$

$$\text{OR } F = \frac{.22 \times \Delta S}{K_W \times L}$$

WHERE:
F = FREQUENCY - CPM
R = RATE #/IN.
W = ACTIVE SPRING WEIGHT
d = WIRE DIA.
D = MEAN SPRING DIA.
N_A = NO. OF ACTIVE COILS
K_W = WAHL FACTOR
L = LIFT
ΔS = NOMINAL OPERATING STRESS RANGE

WHEN USING A SMOOTH ACCELERATION CURVE AND OBTAINING GOOD VALVE GEAR DYNAMICS THE FREQUENCY SHOULD BE HIGH ENOUGH TO OPERATE ABOVE THE 9TH HARMONIC AT MAX. ENGINE SPEED.

WHEN OPERATING WITH POOR VALVE GEAR DYNAMICS THE SPRING FREQUENCY SHOULD BE HIGH ENOUGH TO BE ABOVE THE 11TH HARMONIC AT MAX. ENGINE SPEED.

WHEN OPERATING BETWEEN THE 9TH & 11TH HARMONICS DAMPING COILS HELP TO MINIMIZE SPRING VIBRATION.

WHEN OPERATING BELOW THE 9TH HARMONIC A FRICTION TYPE SPRING DAMPER IS PREFERRED.

Fig. 14 – Formula for valve-spring frequency

the beginning of the lift cycle a time interval T is required for a displacement at one end of the spring to reach the opposite end. In the time interval shown T_1 after the lift L_1 has occurred, the last coil has remained undeflected. This means that the coils that have been deflected will be stressed higher than the theoretical amount if the transient wave did not exist. This transient surge is readily apparent above speeds of 4200 rpm and partially explains why at very high speeds the spring appears to surge continually and does not appear and disappear in relation to the harmonic number.

The use of a valve gear incorporating hydraulic lifters giving good valve gear dynamics permits the use of lower valve-closed spring loads. Since the upper load is usually limited by tappet-face loading, the use of a low valve-closed load permits operating with a high stress range, as shown in Fig. 13. Poor valve gear dynamics necessitates high valve-closed loads with higher minimum initial stress. This limits the stress range to a lower value. The natural valve-spring frequency is directly related to the nominal stress range, as shown in the formula in Fig. 14, and therefore in order to permit operation in the higher stress region, good valve gear dynamics is essential.

When using a smooth-acceleration curve and obtaining good valve gear dynamics, the frequency should be high enough to be above the ninth harmonic at maximum engine speed. If poor valve gear dynamics is being obtained, then the frequency should be high enough to be above the eleventh harmonic at maximum engine speed. Spring damping means are desirable when operating below the eleventh harmonic. Variable pitch damping coils at the fixed end of the spring are adequate when operating between the ninth and eleventh harmonics. Below the ninth harmonic a friction-type damper is preferred, especially when only a single valve spring is used.

The use of dual springs is recommended to help counteract the effect of spring surge. It also provides a safety factor in the event of spring breakage. This is considered very desirable in overhead valve gear, where the piston can strike a valve, causing valve breakage and severe engine damage.

The stress diagram is applicable to springs that are cyclo-pelted and preset to improve endurance and to minimize load-set in operation. When springs are designed to be within the limits of this diagram and are operated without excessive spring surge, the maximum load-set after 500 hr of operation is in the order of 5%. With the present high quality of carbon valve-spring wire and the large improvement in endurance obtained by cyclo-pelting, the load-set rather than the endurance generally limits the stress.

Cyclo-pelting of springs refers to the technique of precision shot-peening springs by rolling them on a plane surface under a stream of steel shot, as shown in Fig. 15. As the spring rolls on the plane surface, all points on the surfaces being shot-peened go through a cycloidal motion. Both the inner and outer surfaces are precisely scanned by

the shot, assuring a uniform peening operation. These springs are carried forward under the shot stream by means of small pins attached to a moving conveyor chain. Laboratory and field results clearly show this method of uniformly shot-peening each spring individually is superior to other popular methods of shot-peening. The mechanics of shot-peening consists in producing a compressive stress on the surface of the spring. It also increases the strength of the surface material, where fatigue failure first tends to occur. This combined action greatly reduces the tendency for spring fracture.

Presetting of springs, which is beneficial in minimizing load-set in operation, consists in compressing a spring beyond the yield point so that an actual reduction in length occurs, as illustrated in Fig. 16. This is best accomplished by compressing the spring solid about two times during the manufacturing operations. This requires that the spring be designed with a solid stress in the range of 125,000 to 150,000 psi (Wahl corrected) to permit proper presetting. A preset spring permits operation in a stress range that is located farther below the yield point. The endurance and load-set characteristics are improved, due to an improvement in the stress pattern in the cross-section of the wire. A comparison of the stress pattern in a wire under torsion for a plain wire and a preset wire is shown in Fig. 16. Presetting introduces beneficial residual stresses with directional qualities, which act under load to decrease the maximum stress to reduce the load-set in operation, and to reduce further the surface stress that helps to improve the endurance of the spring.

The valve gear is a composite mechanical system with the functional characteristics of all the component parts closely interrelated. Performance of the valve spring is dependent upon the cam design, the valve gear dynamics at high speed, and the characteristics of the hydraulic lifter. In like manner the performance of any one of the other three elements is related to the characteristics of the other three acting as controlling parameters. The best approach to obtaining a well-performing overhead valve gear for the high-speed passenger-car engine using hydraulic lifters appears to be to:

1. Use an acceleration curve having a smooth or parabolic type of positive acceleration, which blends in smoothly with the negative acceleration. Rapid changes in acceleration are undesirable and the magnitude of the positive acceleration should be limited to a maximum of five times the negative acceleration.

2. Use a valve gear having proper weight distribution consistent with obtaining maximum rigidity to secure a relatively high valve gear frequency of 40,000 to 50,000 cpm, or higher.

3. Match the cam design to the valve gear to obtain a lower critical speed above 4200 rpm.

4. Use hydraulic lifter designs having low pump-up tendencies or possessing anti-pump-up characteristics to permit using reasonably low valve-closed spring loads.

5. Use high-frequency valve springs and employ

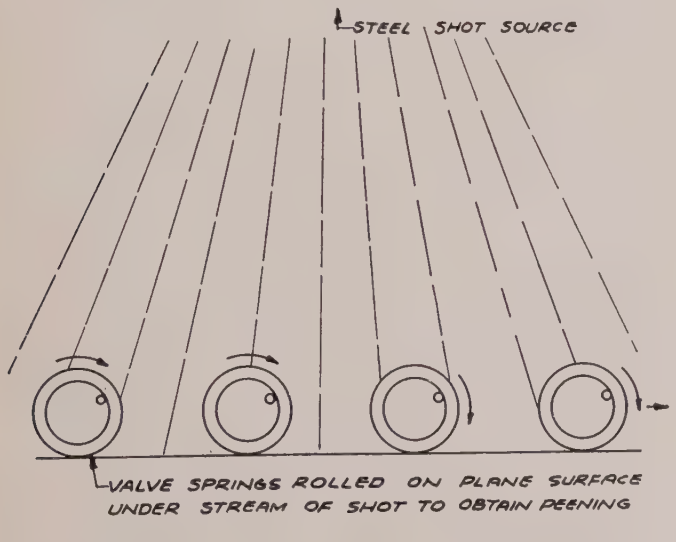


Fig. 15 – Shot-peening of valve springs by cyclo-pelt method

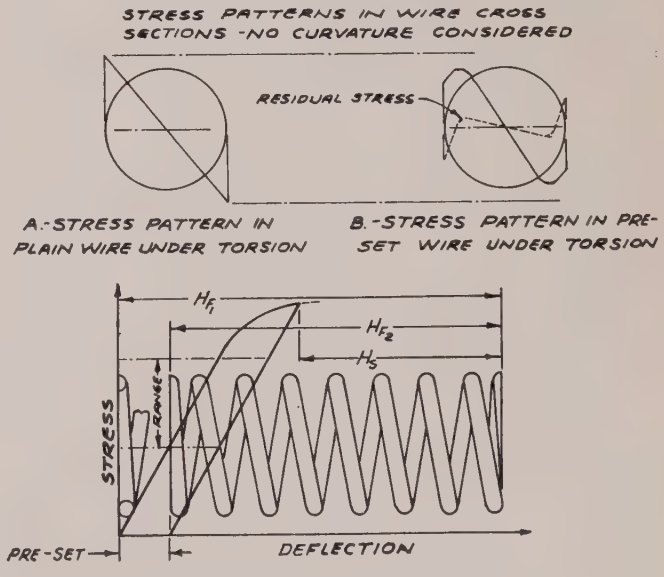


Fig. 16 – Presetting of springs

some spring surge damping method when operating below the eleventh harmonic.

6. Use dual valve springs.

7. Process valve springs by a precision shot-peening technique and preset whenever possible.

Selected Bibliography

1. "Idiosyncrasies of Valve Mechanisms and Their Causes," by F. Jehle and W. R. Spiller. *SAE Journal,* Vol. 24, February, 1929, pp. 133-143.

2. "Electric Telemeter and Valve-Spring Surge," by W. T. Donkin and H. H. Clark. *SAE Journal,* Vol. 24, March, 1929, pp. 315-326.

3. "Vibration Problems in Engineering," by S. Timoshenko. Pub. by Van Nostrand, New York, 1937.

4. "Forced Vibrations and Resonant Frequencies in Machinery," by H. O. Fuchs. *Product Engineering,* Vol. 9, July, 1938, pp. 256-258; August, 1938, pp. 299-302; December, 1938, pp. 479-480.

5. "Poppet Valve Dynamics," by E. H. Olmstead and E. S. Taylor. *Journal of Aeronautical Sciences,* Vol. 6, July, 1939, pp. 370-375.

6. "Graphical Analysis of Surges in Mechanical Springs," by K. J. DeJuhasz. *Journal of Franklin Institute,* Vol. 226, October, 1938, pp. 505-532; November, 1938, pp. 631-644.

7. "Graphical Analysis of Free Vibrations of Helical Springs," by K. J. DeJuhasz. *Journal of Franklin Institute,* Vol. 227, May, 1939, pp. 647-672.

8. "Valve Gear Problems," by M. C. Turkish. *Dumont Oscillographer,* Vol. 8, January/February, 1946.

9. "Valve Gear Design," by M. C. Turkish. Pub. by Eaton Mfg. Co., Detroit, 1946.

10. "Mechanical Vibrations," by W. T. Thomson. Pub. by Prentice-Hall, New York, 1948.

11. "Analysis of Dynamic Forces in Cam-Driven System," by J. A. Hrones. ASME Transactions, Vol. 70, July, 1948, pp. 473-482.

12. "Vibrations in Valve Mechanisms," by T. Warming. ASME Paper No. 49-OPG-2. Presented at 1949 Oil & Gas Power Conference.

Discussion of Valve Symposium Papers

Investigation with Electronic Differential Analyzer

—Clifford H. Allen

Thompson Products, Inc.

THERE is little doubt that the dynamic reaction of the valve gear to the force imposed by the camshaft must form the basis for any effective attempt to improve upon valve motion.

It is obvious that any cam, no matter how poorly designed, will produce "0" amplitude of vibration at "0" engine speed. It is possible, however, to pre-select a second engine speed at which we may attain the condition of "0" amplitude of vibration. This can be done with the polidyne system of cam design developed by Dudley and later expanded by Stoddart.

The speed at which we attempt to arrive at "0" amplitude of vibration is known as "design speed." It is usually selected to occur at 90% maximum engine speed. For instance, if we are working with an engine with which a maximum speed of 5000 rpm is to be considered, we might pick 4500 rpm as design speed. I believe this provision is unique with the polidyne system.

Most of you are familiar with the rather peculiar looking acceleration curve characteristic of a polidyne cam. There is a double peak in the positive portion of the curve. This actually reflects the presence of dynamic factors in the cam design.

If one considers the simplified system consisting of a single mass concentrated at the valve with a weightless spring interposed between the mass and the equivalent cam, it is easy to see that the cam must provide for the motion of the valve. However, the cam must also provide for a controlled deflection of the spring of such magnitude that the deflection load resulting will at all times be exactly equal to the acceleration load required to move the mass in the prescribed manner. Such a system can be described as being in dynamic equilibrium throughout the lift cycle. Theoretically, there will be no exciting forces to induce vibration. This is precisely what occurs at design speed.

The curve showing amplitude of vibration versus engine speed is typical of all cams designed by this method. It has a value of "0" amplitude at "0" engine speed. Amplitude increases very slowly until it reaches a maximum at 70% of design speed. The curve returns to "0" amplitude again at 100% design speed. Further increases in speed result in rapidly increased values of vibration amplitude. At 110% of design speed, the amplitude of vibration is equal to the amplitude at 70% of design speed.

It has been our experience, with very few exceptions, that the maximum amplitude of vibration is well within the static and dynamic deflections in the valve gear system. It is obvious that, if the amplitude of vibration exceeded the static and dynamic deflection of the valve gear system, the valve would lose contact with the cam and valve clatter might occur in the neighborhood of 70% of design speed. The percentage of success in practical applications of the polidyne system of cam design has been gratifyingly high.

In addition to having a sound method to apply, the cam designer must have a practical understanding of the valve gear system, including the influence of the mass and stiffness of various components, such as push-rod, rocker arm, camshaft bearings, and so forth. For this reason, we are currently conducting an investigation of valve gear systems using an electronic differential analyzer.

The first step in this study is to set up differential equations describing the particular valve gear system under consideration. These equations are then rewritten in integral form. They are wired into the computer in integral form. A forcing function representing cam lift in terms of voltage is fed into the computer, which, in turn, draws a curve showing the valve motion that would result with the particular valve gear and camshaft in actual operation.

An imaginary time base is used making it possible to vary engine speed with a twist of a dial. Such factors as mass, stiffness, and damping forces of push-rod, rocker arm, and so forth, can also be controlled with a twist of a dial.

A considerable quantity of data can be obtained in a matter of hours, which would ordinarily require several

months of expensive engine testing – all this without the necessity of building a camshaft.

I feel confident that, with the tremendous amount of technical skill now being devoted to valve motion studies with the help of the latest electronic equipment now available to analyze the results of such studies and with electronic digital computers to take the drudgery out of the mathematics, a relatively simple method may soon evolve that will remove a good deal of the mystery now associated with valve motion problems.

Says Valve Seating Velocity Should Be Considered

– A. W. Hussmann
Pennsylvania State College

MESSRS. Horan and Roggenbuck consider the first positive part of the acceleration curve as the yardstick for determining if a cam is good or bad. They come to similar conclusions. I agree that such an approach at least gives an indication of the no-follow behavior of the valve, which means tells us if the cam follower will jump off the cam or not.

However, this is only one side of the picture, which I admit has confusingly many sides. At least one more vital factor should be considered, that is, the valve seating velocity. While it might be true that operating in a no-follow speed will in general result in unfavorable seating conditions, it is conceivable that even safe cams, designed around the critical design point of Mr. Horan, still vary widely in their seating velocity. There is most likely also a critical speed for designing for low seating velocities, but it has not been discovered yet. Obviously, not just the first part, but the complete acceleration pattern has to be taken into account. I should think this problem is worth another doctor thesis.

Fig. 1 of Mr. Roggenbuck's paper shows the equivalent damping as a dashpot parallel to the flexibility, which I think is about right as long as we don't know better.

However, the fundamental differential equation, shown in the same figure and used for all of his calculations, shows the damping dashpot parallel to the total valve motion. This must give a greatly distorted picture, as is immediately evident if we consider that with such a damping the damping effect would increase with the camshaft speed: The higher the speed, the better the damping. We know that that is not true.

Dr. Barkan has had quite a time to figure out the proper type of damping that would result in a good agreement between theoretical prediction and actual valve behavior. He even invented a new type of friction, which neither Mr. Coulomb nor the theorists of the velocity proportional damping knew of: The magnitude of that friction is proportional to the load; the sign, however, is proportional to the absolute valve velocity. Dr. Barkan had some headache to explain the physical nature of such an animal, but it works.

Some word of caution I wish to direct to Mr. Roggenbuck. In his study he has frequently shown a constant-acceleration cam compared with other, smoother cams. It is true, that the constant-acceleration cam is, by nature, the cam which allows to reach the highest lift with the lowest acceleration and deceleration forces.

It also can be manipulated to result in low gear train vibrations, as Mr. Roggenbuck and incidentally Prof. Warming, years ago, have shown. However, we have learned that it has serious disadvantages.

One of them is higher excitation of valve spring vibrations. In the good old days, when I was struggling with valve dynamics, we thought we had solved the whole problem by learning how valve spring and cam contour could be matched so as to result in low spring vibrations. Today, we are neglecting spring vibrations completely, but we are allowed to do so, only as long as we keep them small. We do this by two means:

1. Use high natural frequency springs, with consequently high stresses. In this way only higher harmonics of the cam lift come into action, which are usually small.

2. Use smooth cam contours for which the Fourier harmonics in question are small.

If we drop the second condition, as Mr. Roggenbuck did for the new Lincoln cam, it might well pay to have a closer look again at the Fourier analysis of his cam and the resultant spring vibrations. The no-follow speed on the differential analyzer might be sufficiently high, but the analyzer does not know that spring vibrations exist. The result can well be that in practice there occur no-follow speeds and valve clatter at the critical spring resonances and even eventually valve spring failures.

Smooth-Acceleration Cam Development

– J. D. Turlay
Buick Motor Division, GMC

THE theoretical approach to valve gear problems is particularly interesting, since many of us do not have the time or the mathematical facility to go through a complete analysis of this sort. Dr. Barkan is to be commended on the close correlation of his theoretical and experimental results. It is to be hoped that his and Mr. Horan's work will lead to simple formulas useful in designing better cam profiles.

In contrast to the theoretical approach, Buick cam design is almost completely empirical. To describe our "cut and try method," or as Mr. Kettering would say, our "method of experimental evaluation," a brief history of the Buick smooth-acceleration cam development will be given. It is probable that a thorough mathematical analysis would have predicted some of the results that we observed.

As engine speeds went up in the first Buick straight-eight engines, the overhead valve gear problem became more troublesome. The 1934 engines, which were equipped with roller lifters and heavy lifter and valve springs, had a clatter speed of about 4300 rpm. A moan in the engine, which was a common field complaint, was traced to the camshaft and lifters. It was felt that a cam profile with smooth and gradual changes in acceleration would reduce the shock loads on the camshaft. Such a camshaft did cure the moan, and so it was used in 1935 production.

With roller lifters the maximum flank acceleration is

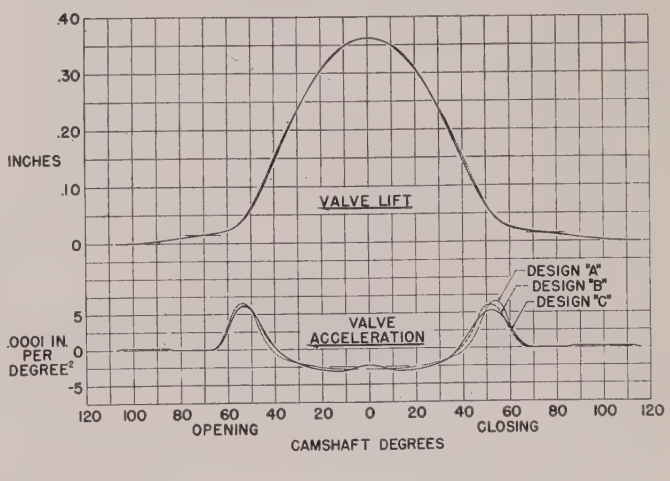


Fig. A – Valve lift and acceleration

limited by the commercial necessity of avoiding a concave flanked cam. With the adoption of flat-faced tappets in 1936, it was believed that a more effective lift diagram could be obtained by increasing the positive accelerations and that the consequent lower negative acceleration would make lighter valve springs feasible, with consequent improved tappet life. A camshaft with the symmetrical smooth-acceleration curve (A in Fig. A) was tested without lifter springs and with 160-lb valve springs. It proved to be a very poor design, and produced violent spring surge and valve bounce at several critical speeds. Sustained running at 3900 rpm invariably resulted in broken valve caps, springs, or valves.

Careful stroboscopic study of the valve action indicated the necessity of a lower positive acceleration, especially on the closing side, even at the cost of a higher negative acceleration. It was felt that, with the closing inflection point higher on the lift curve, the mechanism would have more time to recover from the shock of the closing acceleration before actual valve seating occurred. The second design tested (profile B) was slightly better than A. A third design (C) showed such a marked improvement that it was adopted for 1936 production and used without change until 1948. With this profile valve springs were reduced to 120 lb top load with a clatter speed of 4800 rpm. The difference in performance of the three cams was great, but the valve lift curves differ only slightly. The actual maximum deviation of valve lift curve C from curve A is only 0.008 in.

Although over forty cam profiles have since been made and tested, no great improvement has been obtained over the basic shape of the 1936 design. Strangely enough, the same basic shape has proved to be the best in a number of experimental engines in which the rigidity of the valve mechanism was considerably different, although, as was to be expected, the best results were obtained with the more rigid valve mechanisms.

The acceleration curve that appears desirable is of more or less sinusoidal form, with the closing positive acceleration portion approximately 15% lower than the opening and correspondingly wider. The negative acceleration portion is designed to conform to the spring load curve with a very gradual sweep from the positive acceleration portions of the profile. In some cams a slight reduction in negative acceleration at the cam centerline serves to broaden the lift curve without increasing the valve lift and spring stress. These curves are distorted from a sine wave in a manner that we think makes some allowance for the valve mechanism deflection, but that, in any case, seems to meet with the approval of the valve train.

The computation of the Buick cams is made easy and simple by the use of the increment difference method. The acceleration curve is drawn free hand and modified within the limits dictated by our past experience until the desired lift curve is obtained. A computation of the maximum permissible nose acceleration is useful and is usually made. The experimental profiles are checked in an engine with a stroboscope and, since the adoption of hydraulic tappets, with simple devices to indicate pump-up and pump-down.

The pump-up speed of the 1953 Buick V-8 is in excess of 5300 rpm with valve spring loads of 62 lb closed and 144 lb open. The rigidity and light weight of the mechanism contribute to this result. It has been found possible in experimental engines with cam designs having reduced accelerations to attain speeds of over 6500 rpm with hydraulic tappets and no higher spring loads.

We cannot agree with the statement that spring surge is not important in influencing valve action, although it is true that a proper cam design will reduce spring surge. Buick uses valve springs with a variable pitch, or basket coils to reduce the amplitude of the surge, and dual springs because the effect on the valve mechanism of the independent surge periods is less.

We agree with Mr. Horan in his findings on the tappet face problem, although we have never experienced any great alignment difficulty in production with our flat-faced tappets. The clearance of the tappet in the guide undoubtedly aids in compensating for minor manufacturing errors.

Improperly dressed or broken-down grinding wheels, which sometimes produce slightly stepped or hollow-ground cam surfaces, will ruin the tappets very quickly whether they are flat or crowned, judging from our experience.

Barkan Closure To Discussion

IT is perhaps possible to resolve the apparent disagreement raised by Mr. Turlay over the effect of spring vibrations upon valve motion in the following way: With highly flexible systems, such as was used in this investigation, the linkage vibrations have a much greater significance than spring vibrations. However, as the natural frequency of the linkage is increased, the linkage vibrations become less significant and, as a result, spring vibrations have a proportionately greater influence upon the net valve motion. With a highly rigid system such as is encountered in L-head engines, there is no doubt that the spring surges are of first-order importance at high speeds.

With the low-frequency linkage employed in this investigation, it was observed that, when the linkage jumped off the cam, spring vibrations did at times influence the duration and amplitude of the calculated valve jump, but were not of first-order importance. Footnote 3 of my paper provides additional examples bearing out this conclusion.

It is not believed that unsatisfactory performance of an overhead valve mechanism should be attributed primarily to spring vibrations if Dr. Hussmann's recommendations are followed, and if dual springs are used, as Mr. Turlay suggests.

The effect of spring vibrations might be included in the calculation of valve motion with the aid of methods described in Nos. 7, 8, and 9 of my "Additional References." However, on the basis of the conclusions reached in this investigation, this complication is not ordinarily necessary.

Roggenbuck Closure To Discussion

THE author wishes to thank Dr. Hussmann for his comments. In regard to the matter of the damping terms, I must admit to an error in one of my sketches. The dashpot should be shown connected to ground in Fig. 2. It is certainly true that we know little about the real nature of damping in a valve train—in fact, it is probably the most promising field for study in further improving operation of valve gear. However, I do feel that the damping force should be considered as in some way proportional to the load, since it is probably largely due to hysteresis and friction. This means that the increased inertia loads at the higher speeds should cause an increase in the damping force.

In regard to the question of excitation of the valve spring by harmonics of the cam profile, there is no doubt that the constant-acceleration cam has a stronger set of higher harmonics. However, the force pattern that is impressed on the valve spring is so modified by the response of the whole valve train that it no longer contains these harmonics in the same proportions. Comparison of Figs. 5 and 6 will clarify this point. The two valve acceleration curves are nearly identical, although the cam patterns are vastly different. Since the spring can only be excited by the force that appears at its free end, it is apparent that these cams will have approximately equal harmonic valve-spring excitation characteristics.

AVIATION FUEL ECONOMY AND QUALITY — BROTHERS UNDER THE AIRCRAFT SKIN

H. E. Alquist, Phillips Petroleum Co. **E. A. Droegemueller,** Pratt & Whitney Aircraft Division, UAC **and H. N. Taylor** United Air Lines, Inc.

This paper was presented at the SAE National Aeronautic Meeting, New York, April 23, 1953

UTILIZATION of a large number of new 4-engine aircraft by United States airlines has sharpened interest in reducing fuel consumption during cruise. This interest has manifested itself primarily in several service tests by the airlines of the so-called "manual lean" cruise control procedure in place of cruise operation at automatic lean carburetor settings or at best-power fuel/air ratios with constant spark advance. It is the purpose of this paper to outline some of the requirements for maximum-economy operation and indicate the mutual effort that is necessary on the part of the aircraft operator, engine manufacturer, and petroleum refiner to assure success of this operation.

The question of minimizing aircraft-engine fuel consumption has been covered before[1, 2] and considerable flight experience was developed by Pan American just prior to the war. On the other hand, this is the first time that serious consideration can, or has to, be given to obtaining best economy during cruise on majority of 4-engine aircraft in domestic traffic. First of all, the services of a flight engineer should enable the airline to establish engine operating procedures that may be more involved but that are designed either to reduce fuel consumption or extend engine life over the use of the aircraft with only two crew members. The second reason for the selection of more economical cruise conditions is the realization that major otto-cycle engine design modifications giving better efficiency cannot be expected from manufacturers committed to a large jet-engine defense program. Likewise, it would appear that, since the aircraft piston engine has nearly reached the ultimate in mechanical development, it is a logical time to start "weaning the beast."

The choice of these conditions for maximum economy is controlled to a large extent by the quality of fuel used in the operation. Heretofore the question of availability in time of emergency was the controlling consideration in establishing

AFTER defining the operating conditions for maximum aircraft fuel economy in terms of fuel/air ratios, spark advance, and engine cooling drag, the authors define the characteristics of aviation gasoline necessary for the attainment of these engine conditions.

The Authors

H. E. ALQUIST (J '45) is an aviation technologist with the Research and Development Department of Phillips Petroleum Co. He received his B.S.M.E. degree from North Dakota State College in 1943, and five years later, received his Master's degree in mechanical engineering from Case Institute of Technology.

E. A. DROEGEMUELLER (M '42) is a project engineer with Pratt & Whitney Aircraft, Division of United Aircraft Corp. He is currently working on fuel and lubrication requirements of present and future engines.

H. N. TAYLOR (A '38) is superintendent of powerplant engineering at the Pacific Operations Base of United Air Lines, Inc. He received his B.S.M.E. from Pennsylvania State College. He has been with United since 1943.

[1] See *SAE Journal* (Transactions), Vol. 44, June, 1939, pp. 235-242, 251: "Fuel-Economy Possibilities of Otto-Cycle Aircraft Engines," by D. S. Hersey.

[2] See SAE Transactions, Vol. 53, 1945, pp. 441-449: "Valuation of Better Fuels," by E. A. Ryder.

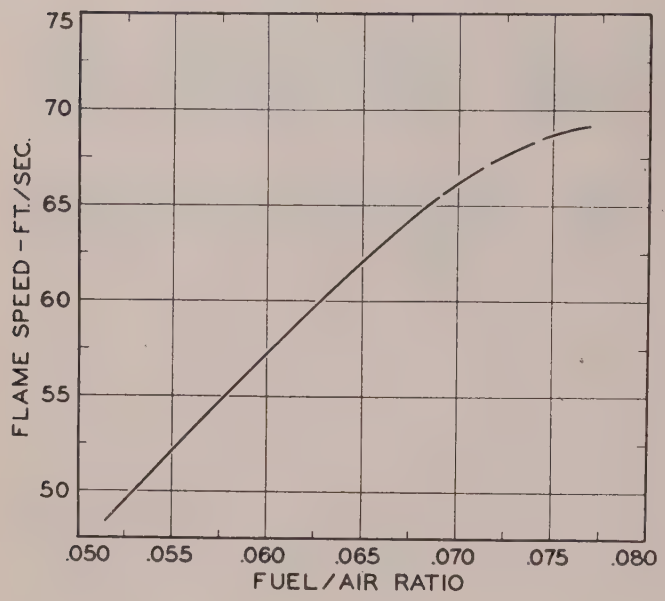


Fig. 1

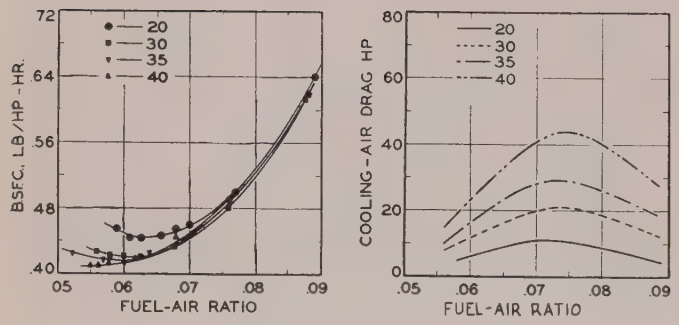


Fig. 2

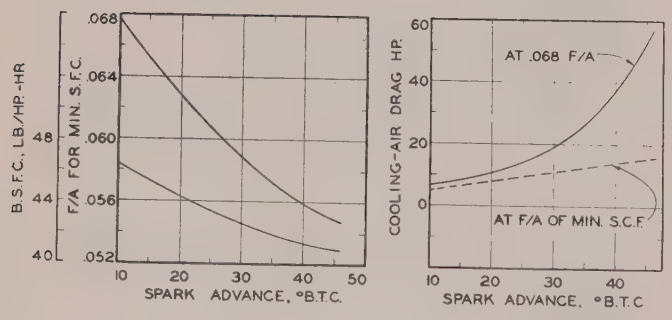


Fig. 3

gasoline quality requirements for both civilian and military aircraft. Aircraft powerplants have generally been developed around, and their power rating established on, a fuel that might be expected to be produced in large scale during an emergency. Availability considerations become less important on commercial avgas in the grades above 100/130 as jet aircraft take on the assignments previously given to piston-engine equipment utilizing avgas of better than 100/130.

The two factors controlling the fuel economy of an aircraft during cruise are: (1) operation at an air-speed giving maximum lift with minimum drag, and (2) operation at engine conditions that allow maximum thrust for a given quantity of fuel but add minimum drag to the airplane. Since the first factor is dependent upon pilot skill and/or schedule requirements, only the second factor is within the scope of this paper. While the effect of operating conditions on economy is fairly well realized, utilization of the optimum cruise conditions for minimum fuel consumption has been subjected to various compromises in order that aircraft reliability would not be jeopardized with the fuels that could be made available in wartime. After the effect of operating conditions on economy are reviewed, compromises that are related either directly or indirectly to fuel quality will be covered.

As indicated earlier, most of the data discussed herein were obtained in the cruise control programs of NACA, CRC, and engine manufacturers under the sponsorship of the military services during the last war. The role of the authors has been one of selecting, summarizing, and bringing up to date, where necessary, the vast amount of data from the literature.

Effect of Design and Operating Conditions on Economy

Probably the engine conditions that most influence fuel consumption, and yet can be changed with no corresponding design modifications, are spark advance and fuel/air ratio. These two must be considered together since utilization of lean fuel/air ratios to decrease fuel consumption is generally not completely effective unless spark advance is also increased. With more time required to complete combustion at lean fuel/air ratios because of the correspondingly lower flame speeds at these mixtures, combustion must be initiated earlier in the cycle. This decrease in flame speed with lean mixtures is illustrated in Fig. 1 for a turbulence condition representative of that in a modern aircraft-engine cylinder. While Hersey[1] shows that advanced spark at cruise cannot be considered competitive with compression ratio in reducing specific fuel consumption per se, it is seen from this figure that some spark advancement is necessary to overcome the deficiency in flame speeds at lean mixtures.

The net result of fuel/air ratio and its corresponding effect on flame speed can be seen in Fig. 2

(reproduced from Farmer and Munger[3]), where it is noticed from the best-economy hooks that for this particular aircooled engine it is necessary to advance the spark setting about 20 crank-angle deg to obtain maximum reductions in specific fuel consumption with lean fuel/air ratios. Another important and practical reason for advancing the spark at lean mixtures is, of course, to avoid burning after the exhaust valve has opened, with its attendant reduction in valve life. Fig. 2 also shows, however, that as spark is advanced to compensate for lower flame speeds, the cooling horsepower loss at any fuel/air ratio increases. To avoid this increased cooling drag loss and to obtain maximum economy, it is essential to operate in cruising spark advance at progressively leaner fuel/air ratios. This is illustrated in Fig. 3 in which the data from Fig. 2 have been cross-plotted. It is seen that operation at advanced spark at the minimum fuel consumption fuel/air ratios does not increase the cooling load appreciably compared to that at constant fuel/air ratio and yet reduces the specific fuel consumption by about 7% for this example.

In connection with this question of cooling load it should be brought out that the maximum miles per pound of fuel are not necessarily attained at the fuel/air ratio of minimum brake specific fuel consumption. This is particularly true for high-altitude advanced-spark operation. Fig. 4 (reproduced from Farmer and Munger[3]) shows a typical curve of head temperature versus fuel/air ratio for a constant-power, constant-cooling condition. At a constant head temperature the cooling curve for the cylinder would, of course, have the same shape. If a best-economy curve at constant head temperature is obtained for this cylinder its shape on a brake horsepower basis would be as shown by the dotted line at the top of Fig. 4. If, however, the fuel flow is divided by the net horsepower available to the airplane after subtracting the cooling load, the best-economy curve would take the shape shown by the upper solid line in Fig. 4. The hump in this curve would only be significant under severe cooling conditions and depends to a large extent on cowling design. For example, analysis of flight test data from a modern 4-engine transport indicates much less difference between the net cooled and brake specific fuel consumption curves at best-power fuel/air ratios, although the minimum net cooled specific fuel consumption still occurs at leaner fuel/air ratios than does the minimum bsfc. The data referred to are shown in Fig. 5 for a transport aircraft operated at 0, 4, 8, 12 and 16 bmep drop from best power at an airspeed and altitude typical of scheduled aircraft in this country. The relatively sharp rise in bsfc at very lean mixtures is due to the fact that

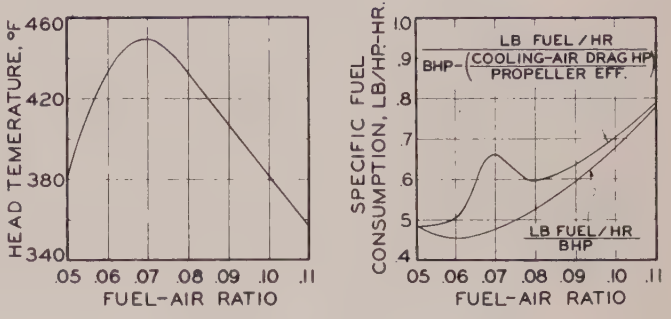


Fig. 4

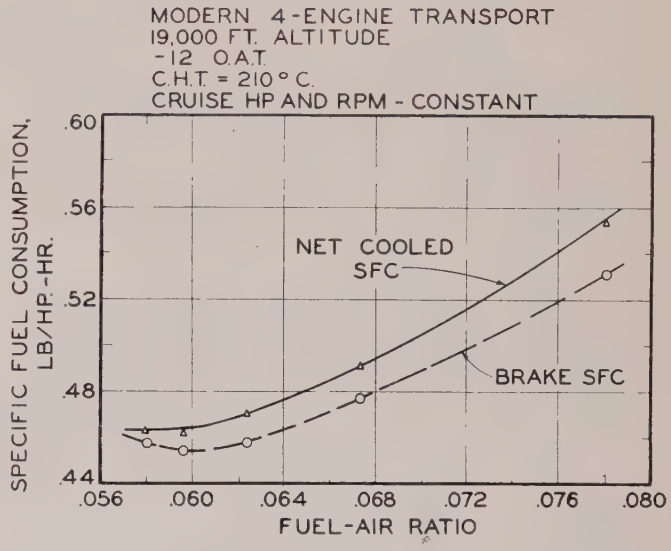


Fig. 5

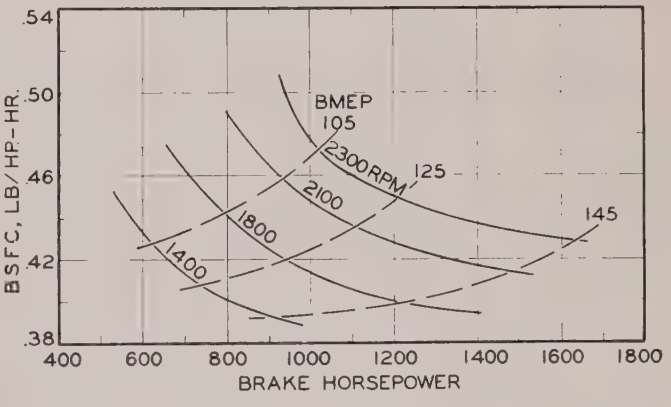


Fig. 6

[3] See NACA ARR E5B01b, February, 1945, "Fuel Consumption Corrected for Cooling Drag of an Aircooled Radial Aircraft Engine at Low Fuel/Air Ratios and with Variable Spark Advance," by J. E. Farmer and M. Munger.

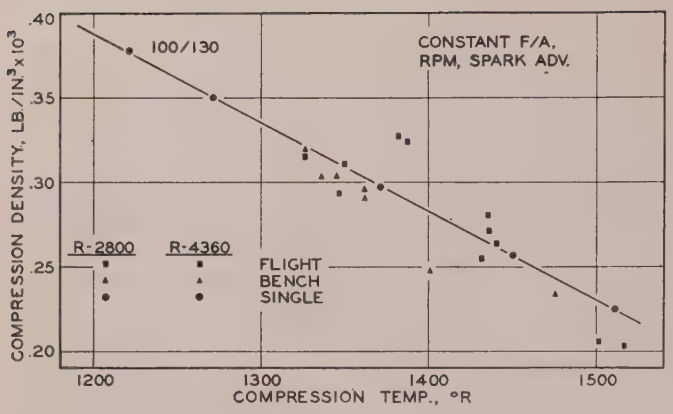


Fig. 7

the spark-advance setting was constant for this fuel/air ratio range and yet flame speeds are rapidly falling off at the leanest conditions. On the other hand, the ability to attain lower cylinder-head temperatures with better cylinder parts life and the concurrent gain in fuel economy emphasizes the value of leaner operation.

Another important factor controlling fuel consumption during cruise is friction horsepower. A typical grid of brake specific fuel consumption versus brake horsepower at various bmep levels is shown in Fig. 6. The obvious reduction in friction horsepower and the corresponding reduction in fuel consumption by obtaining the required power at lower speeds is illustrated at 1100 hp whereby increasing bmep from 125 to 145, bsfc is lowered about 9% for this particular engine.

To summarize this section then, the conditions of operation for maximum miles per pound of fuel, with engines in which one does not expect major design modifications, are at maximum bmep-minimum rpm, at the fuel/air ratio allowing minimum net cooled specific fuel consumption (generally at minimum fuel/air ratio for stable operation) and at advanced spark. The extent to which compromises have been necessary in attaining these conditions by fuel quality limitations can now be considered.

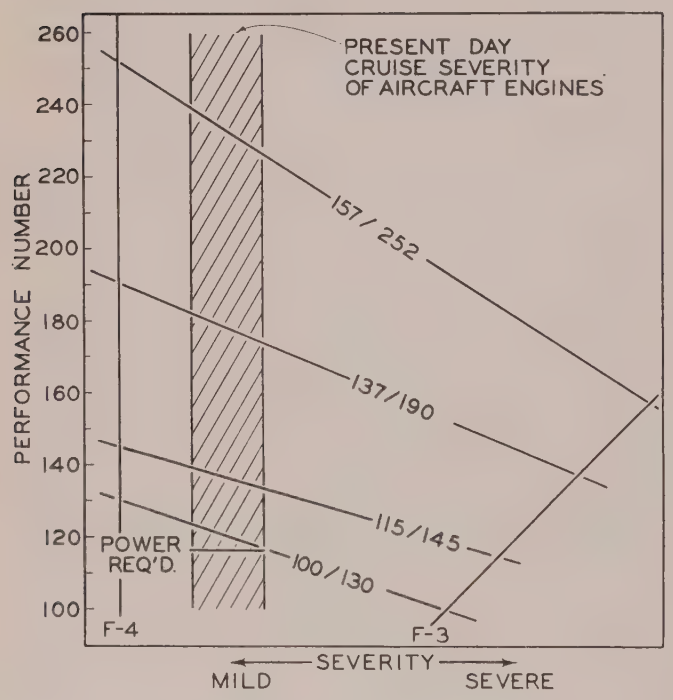


Fig. 8

Effect of Cruise Operating Conditions on Fuel Quality Requirements

Detonation Requirements – Detonation, considered herein to be the result of autoignition of the end-gas or last part of the charge to burn, is generally most prone to occur at the conditions for maximum economy. The reason probably lies in the fact that maximum utilization of the fuel energy is attained when the peak combustion pressure occurs within a few crank-angle degrees after the piston has started down from the top of its stroke; whereas at this condition the end-gas has been subjected to the maximum rate of compression by the piston and flame front working together. Since, with other things being equal, the ratio of chemical chain branching in the end-gases relative to the rate of chain destruction will increase as a function of the rate of temperature rise, the nominal engine conditions allowing maximum rate of compression are most favorable to autoignition of previously unignited gases.

Notwithstanding the above mentioned time factor, the knocking characteristics of a fuel in an aircraft engine have been described with more or less success in terms of calculated functions of the density and temperature of the unburned end-gases just prior to knock.[4] On the basis of these parameters the knock-limited performance of a fuel would appear as a single curve for a range of nominal operating and design variables such as compression ratio and combustion air temperature. An illustration of the application of these parameters is shown in Fig. 7, where the knock-limited performance of 100/130 aviation gasoline is shown as a single curve of compression density versus compression temperature for the R-2800 and R-4360 engines at constant fuel/air ratio, rpm, and spark advance. The ordinate of this curve is a function of knock-limited horsepower output whereas the abscissa is a description of engine severity and, as such, is a function of the nominal engine variables previously mentioned. It will be noted that one curve describes the knock-limited performance regardless of the fact that data were

[4] See NACA TR 655 (1939), "Knocking Characteristics of Fuels in Relation to Maximum Permissible Performance of Aircraft Engines," by A. M. Rothrock and A. E. Biermann.

obtained in flight, bench, and single-cylinder tests.

Although mathematically exact calculations have never been made of the end-gas density and temperature parameters, so that the effect of all nominal variables could be included, it has been possible to deduce the severity of modern aircraft engines with respect to that of the laboratory knock test methods, which guarantee the customer a minimum fuel rating. A somewhat pictorial description of this relationship is shown in Fig. 8. In this figure fuel performance number versus severity, parameters which are functions of knock-limited end-gas density and end-gas temperature, are used to describe the knock-limited performance of a number of fuels. The lines noted F-3 (ASTM Aviation) and F-4 (ASTM Supercharge) designate the severity levels of the laboratory ASTM knock test methods. The cross-hatched area represents the severity range of two widely used aircraft engines operating at conservative cruise conditions. The point at which one of the fuel performance lines crosses the severity regime of the aircraft engine represents the knock-limited performance of the fuel in that engine. If these engines are operated at a given bmep level during cruise their powers can be represented as a horizontal line shown as an example just below the knock-limited curve of 100/130 grade fuel. Once the antiknock characteristics of a fuel required to operate two typical aircraft engines under conservative cruise conditions are established, the fuel antiknock requirements can be established for maximum economy operation.

It was mentioned earlier that the two variables, spark advance and fuel/air ratio, are of considerable interest in this study because not only do they markedly affect economy but they can also be changed with no corresponding design modifications. It has been found from numerous tests that advancing the spark setting to that giving the lowest specific fuel consumption will mean reducing the knock-limited performance of a fuel in an engine. Increasing the spark advance beyond this point increases both specific fuel consumption and the knock-limited horsepower. This relationship is shown in Fig. 9 with the three combustion-pressure versus crank-angle curves at the top of the figure, corresponding to a retarded, optimum, and advanced spark setting, illustrating the point made earlier that when the maximum rate of pressure rise occurs at top center, the specific fuel consumption and knock-limited power will both be at a minimum. At fuel/air ratios of 0.055 to 0.065 for current aircraft engines, it appears that the optimum spark-advance setting or that required for minimum specific fuel consumption, is between 35 and 40 deg btc. An increase in spark advance from 20 deg btc to this maximum-economy value increases end-gas temperature or severity so that a greater premium is placed on fuels of high ASTM Aviation ratings.

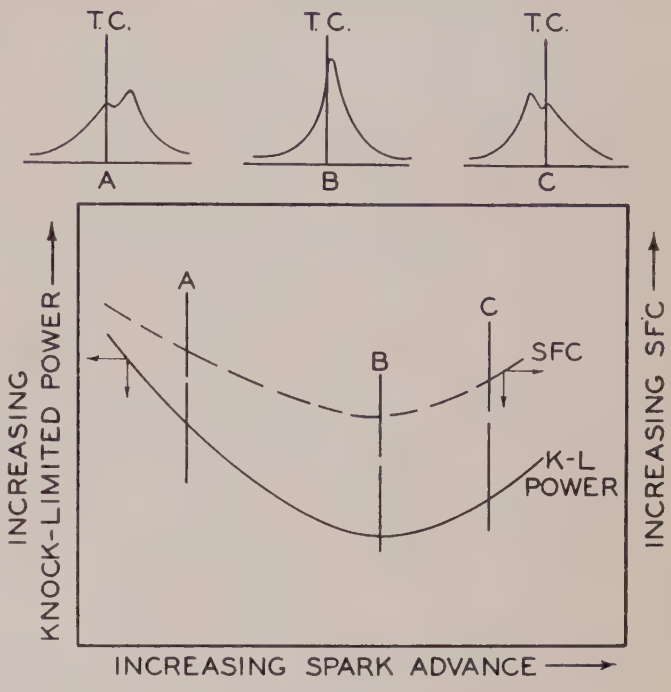


Fig. 9

In a similar manner the fuel/air ratio of minimum brake specific fuel consumption is that at which detonation is most likely to occur. A typical curve of knock-limited horsepower versus fuel/air ratio is shown in Fig. 10 in comparison with typical carburetor settings. For this fuel, at a spark advance of 20 deg btc, minimum knock-limited horsepower is available at the fuel/air ratio at which the minimum brake specific fuel consumption was attained. However, the fact that higher knock-free powers can be attained at fuel/air ratios leaner than 0.060 in combination with the reduced cooling drag in this region indicates that an effort should

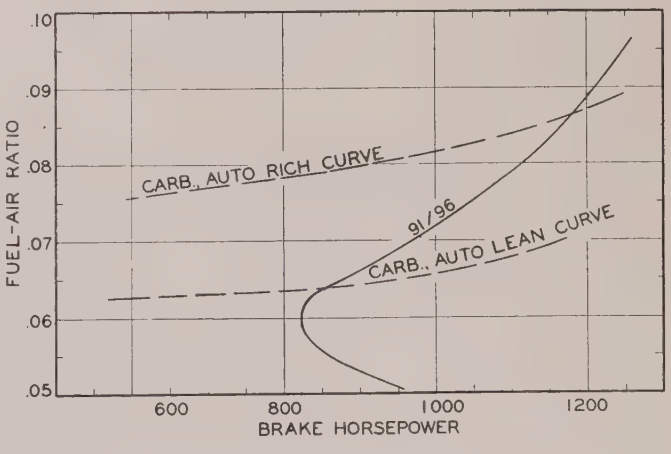


Fig. 10

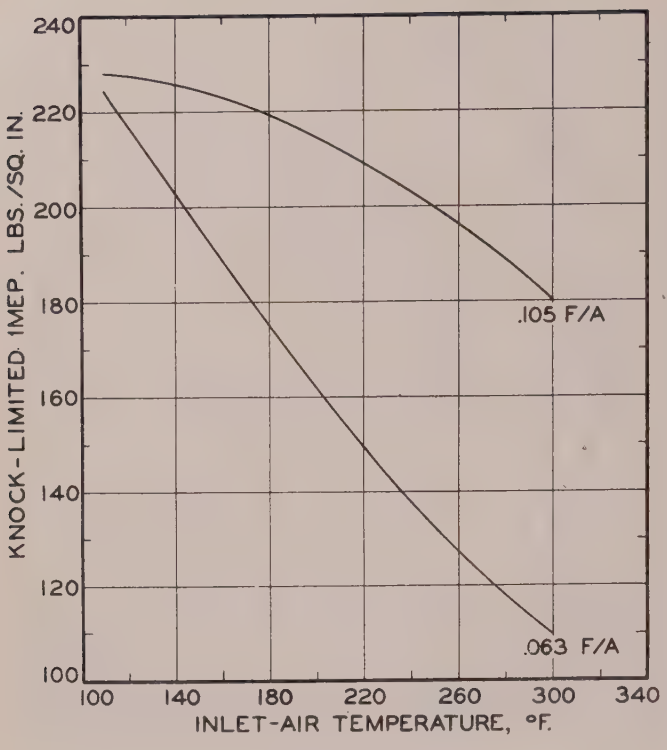


Fig. 11

be made to make this fuel/air ratio range an area of standard cruise operation. It should be emphasized that this suggestion does not carry the inference that fuel antiknock quality is dictated by the rise in knock-limited full-scale engine power at fuel/air ratios leaner than 0.06. On the contrary, the antiknock characteristics of the fuel have to be sufficient to allow passing through the minimum on the curve without encountering detonation, while this combination of higher knock-limited power and cooler temperatures below 0.06 fuel/air ratio merely gives marginal engines greater protection from detonation.

Referring back to Fig. 8 again, it is seen that increasing brake mean effective pressure or horsepower appreciably, at a constant severity level, means that the engine will require a fuel of higher performance number. However, for maximum miles per pound of fuel, it is necessary to stay on the peak of the airplane lift-over-drag curve; thus, higher horsepowers allowed by a fuel of better antiknock characteristics cannot be utilized for long-range cruise where light aircraft weights are involved. At heavier gross weights, of course, a better fuel allows operation at a high-spark-advance manual-lean condition where operation is usually restricted to best-power fuel/air ratios and retarded spark advance. At the lower horsepowers advantage is taken of a fuel of higher performance by increasing bmep at constant brake horsepower and utilizing lower engine speeds to reduce friction horsepower. The advantage of such operation has been shown in Fig. 6 and discussed before. Other than detonation, the limitations to such operation will be lack of supercharger capacity, the lean combustion stability limit as evidenced by engine roughness or backfiring, and engine structural limitations. Cooling losses at higher bmep levels become restrictive at fuel/air ratios of 0.062 and greater, particularly under high blower operation where the temperature rise through the compressor is much higher. A survey of current typical engines indicates that appreciable increase in cruising bmep might be made, provided combustion roughness, detonation, or excessive head temperatures are not encountered at the commonly used cruise horsepowers.

Since increasing cruise bmep means a higher manifold pressure and since the characteristics of the supercharger play such an important role in limiting manual lean operation by dictating the severity of the engine – thus its knocking characteristics – the performance of typical installations should be briefly reviewed. On a typical single-stage 2-speed engine the equivalent inlet air temperature to the engine is more than 100 F higher in high blower at cruising condition than in low blower, even though the ambient temperatures for the altitudes where high blower is required are lower. If utilization of higher bmep levels means using high instead of low blower at a given altitude, a significant increase in engine severity is involved. However, cruise schedules at higher bmep and lower rpm actually mean a reduction in inlet temperature to the combustion chamber, due to lower supercharger rpm, until it is necessary to shift to high blower. For the turbosupercharged engines – generally equipped with intercoolers – the inlet air temperature problem is not as severe; thus, it would appear that higher bmep-lower rpm cruise schedules could be established for these engines more easily than for the geared supercharger types. This would of course involve some increase in intercooler drag.

The reason for our concern about supercharger characteristics, and their possible limitations on cruise power schedules, is illustrated by Fig. 11, where severe depreciation of knock-limited power occurs with high inlet air temperatures at lean fuel/air ratios. These data are similar to those discussed more thoroughly by others.[5] The effect that inlet air temperature exhibits on the knock-limited performance of a fuel has been related to the effect of other variables by a calculated function of end-gas density and temperature.[6] This procedure has been used to obtain Fig. 12, which shows the knock-limited performance of 100/130 and 115/145 grade fuels in a typical single-stage 2-speed engine over a wide range of operating con-

[5] See SAE Transactions, Vol. 52, 1944, pp. 614-620: "Relation of Intake-Charge Cooling to Engine Performance," by E. A. Droegemueller, D. S. Hersey and W. A. Kuhrt.

[6] See NACA TN 2066 (1950), "Correlation of Effects of Fuel/Air Ratio, Compression Ratio, and Inlet Air Temperature on Knock Limits of Aviation Fuels," by L. K. Tower and H. E. Alquist.

ditions. The points at higher end-gas temperature represent data obtained at lean fuel/air ratios, high carburetor air temperatures, and high blower ratios. With these two curves an estimate may be made of the difference in fuel antiknock characteristics required by utilizing maximum-economy operating conditions. Airline take-off power settings are in general very close to the knock-limited curve of the fuel being used. For example, on those engines utilizing 100/130 fuel, the take-off power and severity can be represented by a point in Fig. 12 at approximately 1.7 density and 2000 R end-gas temperature. With conservative operating conditions at cruise, however (20 deg btc spark advance and 0.067 fuel/air ratio), a much greater margin exists between the knock limit and the operating point. For the engine having the 2000 R take-off end-gas temperature, the cruise severity is only about 2100 R and an end-gas density of about 1.0. Actually, of course, the operating points shift about with different ambient air temperatures and altitudes and different models of the same engine, so that the engine is operating in an area in the vicinity of the point. In any event it is evident that some latitude, in so far as detonation is concerned, is available in applying maximum-economy conditions at cruise.

It has been inferred that maximum-economy conditions mean a spark advance of between 30 and 40 instead of 20 deg btc, a bmep from 160 to 165 instead of 140 to 145, and fuel/air ratios of 0.052 to 0.060 instead of 0.065 to 0.075. This change in operating conditions would shift the cruising power end-gas density and temperature from approximately 1.0 and 2100 R, respectively, to about 1.2 and 2200 R. The location of these points with respect to the knock-limited curve for 100/130 fuel is shown in Fig. 13. With the change in operating conditions it would appear that a 100/130 grade fuel would no longer be completely satisfactory for the engine. While this is true, it is interesting to note that the average 1951 production of 100/130 was actually 106/130 and that the average 1951 production of 115/145 was actually 120/146, according to the Bureau of Mines survey of aviation gasoline.[7] This point is emphasized to bring out the fact that, although utilization of maximum-economy cruise conditions means some improvement in the antiknock characteristics over the minimum allowed by present specifications, it at least is not above that which the petroleum industry has been able to supply in the past year or so.

Two qualifications are necessary to the foregoing matching of power settings and knock-limited curves. The first is that an engine manufacturer allows certain cruise power operating conditions on the basis of a guaranteed fuel quality established by a specification. Although it has been indicated above that average production has been above the minimum antiknock specification, the cruise power settings cannot be liberalized until the manufacturer is assured that all fuels going to the engine are as good as the average. Second, the relation between power settings and knock limits discussed above are based on a fuel of constant lead content. There are indications that, for some engines, fuels of the same laboratory knock ratings as those shown in Figs. 12 and 13, but with lower than 4.6 ml tel per gal, would have higher knock-limited performance in the full-scale engine.

To summarize this section on fuel antiknock requirements for maximum cruise economy then, it would appear that little upgrading of present fuel performance numbers above that which was supplied in 1951 is necessary. This conclusion is based on the fact that most aircraft operating today require about 50% take-off power for efficient cruising speeds. At this power level considerable increase in engine severity or end-gas temperature can be made before encountering detonation. Whereas it has been common airline practice to operate their engines at take-off powers very close to the knock limit of the fuel, advantage has not been equally taken of the fuel antiknock characteristics during cruising operation. Generally this has not been the case in military equipment.

Volatility Requirements – Whereas it has been shown that operation at fuel/air ratios leaner than 0.06 has the dual advantage of increasing aircraft miles per pound of fuel by decreasing cooling drag

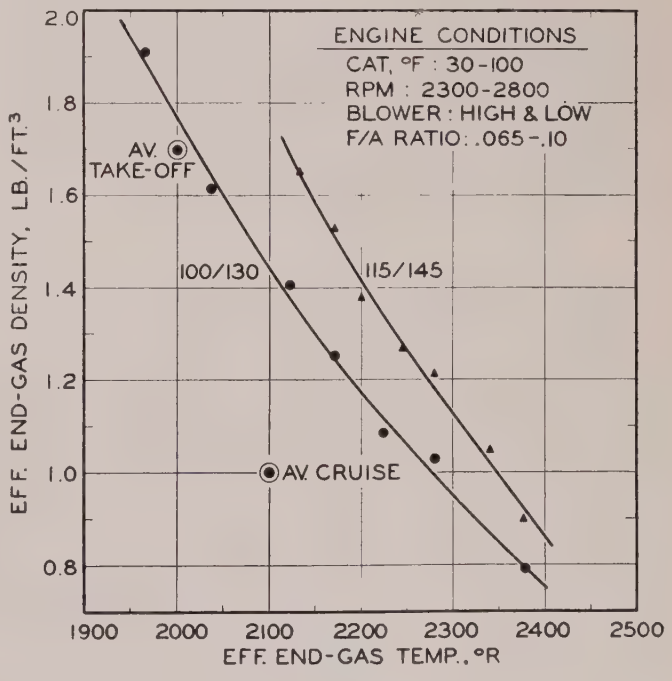


Fig. 12

[7] See U. S. Bureau of Mines, RI 4889 (1952), "National Annual Survey of Aviation Gasoline and Aviation Jet Fuel," by O. C. Blade.

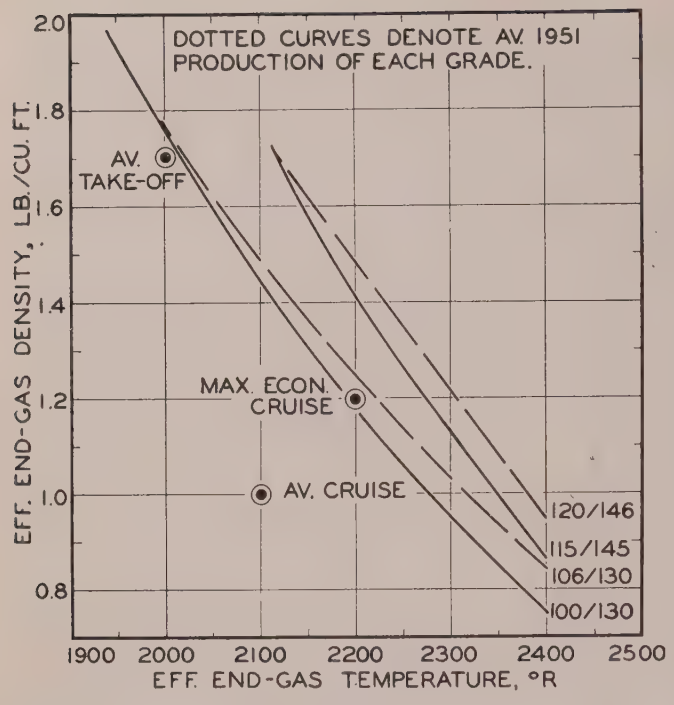


Fig. 13

and also reducing the antiknock requirements for a given power level, such operation places a premium on a homogeneous mixture of fuel vapor and air equally distributed to each cylinder. Such a mixture is particularly difficult to attain at low engine speeds, required for low friction horsepower, since less mechanical mixing of the fuel and air takes place in the blower section and since less heat is added to the charge at lower blower speeds.

At a retarded-spark advance, low bmep operating condition, fuel/air ratio can be varied over wide limits without exceeding cooling restrictions. However, as spark advance and bmep are raised the fuel/air ratio at which excessive head temperatures are exceeded becomes extremely lean (approaching 0.062 fuel/air ratio at certain conditions for some installations). Under these circumstances the fuel/air ratio of lean misfire takes on an entirely new significance and even minor improvements in leaner fuel/air ratios without backfiring are important. Unfortunately, just as the fuel/air ratio of lean misfire is most critical during manual leaning, it is also the operating variable most susceptable to engine conditioning. It is here that the responsibility for a successful venture into manual leaning lies squarely on the shoulders of the aircraft operator. Operation in the vicinity of 0.06 fuel/air ratio and leaner will require stringent maintenance of such equipment as carburetors, ignition harnesses, and spark plugs. Likewise, a more rigid training program will be necessary to marry the flight engineer effectively to his engines.

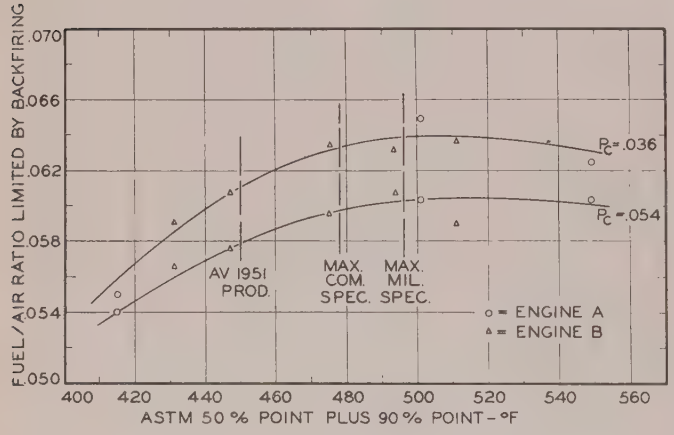


Fig. 14

The operator's role in obtaining smooth engine operation at very lean mixtures has been stressed. Now it might be asked, what are the gasoline requirements for such operation? With respect to volatility, extensive investigations during the last war under the auspices of CRC and the Air Force have shown that lowering the 50 and 90% fuel distillation temperatures permits operation at leaner fuel/air ratios without roughness to a degree dependent upon engine rpm, carburetor air temperature, and cylinder-head temperature. At low speeds and at low carburetor and head temperatures the effect of volatility is, as might be expected, much more pronounced. Fig. 14 is typical of these data and shows the relationship between the fuel/air ratios limited by backfiring and the sum of the ASTM 50 and 90% point for two popular aircraft engines operated at carburetor and head temperatures close to the maximum allowed by structural considerations. Although one of these engines had been tested in flight and the other was tested on the stand, it appears that their lean stability limits are influenced to the same degree over this range of fuel volatility at a given charge density. The two densities selected are indicative of a high and low bmep operating condition and show that increasing the cruise bmep improves lean fuel/air ratio combustion stability as well as improving specific fuel consumption. Vertical lines have been drawn on this figure to represent the maximum 50 plus 90% point allowed by the commercial and military avgas specifications. It can be compared with the average 50 plus 90% point for 1951 production of both civilian and military avgas. It is seen that for these engines there is no significant difference in the fuel/air ratio limited by backfiring between two fuels made to the maximum 50 and 90% points allowed by the two specifications and that volatility is only significant at lower 50 and 90% points. It will be noted that the average 1951 production should be significantly better than the maximum allowed by the specifications from the standpoint of lean misfiring.

The trends shown in Fig. 14 for two carbureted

engines have also been shown for the fuel-injection engines, although not to so marked a degree. Whereas fuel-air distribution in a direct-injection engine between cylinders is usually a function of the air distribution, the shorter vaporization time available in this engine places a premium on a fuel that will evaporate quickly – that is, high tail-end volatility.

Another index of proper distribution of fuel to each cylinder is a small spread in the fuel/air ratios between each cylinder. A small spread in cylinder fuel/air ratio is difficult to attain at rich take-off mixtures, where a great deal of fuel must be vaporized in a short time and at very lean cruise mixtures where there is a lack of heat and where less mechanical mixing is effected due to lower blower speeds. Fig. 15 shows the effect of tail-end volatility on fuel/air ratio spread between cylinders for a take-off and normal rated condition. As might be expected, lowering the tail-end volatility increases the spread in fuel/air ratios between cylinders. It will also be noted in this figure that the average 1951 production avgas would account for very little spread in fuel/air ratios as compared to the least volatile fuels that can be produced to the commercial and military specifications.

Earlier in this paper it was pointed out that utilization of significantly higher bmep cruise schedules on 2-speed single-stage supercharger engines in the field would probably mean greater application of high blower and thus higher inlet air temperatures. Increasing this temperature would undoubtedly reduce both the spread in fuel/air ratios between cylinders and the fuel/air ratio limited by backfiring for any fuel. It is apparent that the low inlet air temperatures required to reduce the tendency to knock are in direct conflict with the high inlet air temperatures required for good distribution.

Finally, it should be noted that fuel volatility has a small but direct effect on fuel economy at lean fuel/air ratios. From the data shown in Fig. 16 it is seen that increasing the tail-end distillation temperatures increases specific fuel consumption to a greater extent with lower inlet air temperatures. This effect is experienced because with heavier fuels burning is not complete in the cylinder and because the maximum rate of combustion pressure rise is not occurring at the top of the stroke. It will be noted from this figure again that the difference in specific fuel consumption between the maximum 90% point allowed by the military and commercial avgas specification is not significant and that the average 1951 military and civilian avgas production should tend to offer lower fuel consumption than either of these maximums.

To summarize this section on the direct effect of fuel volatility on the fuel consumption of aircraft engines and the fuel volatility requirements for the maximum-economy operating conditions, it would

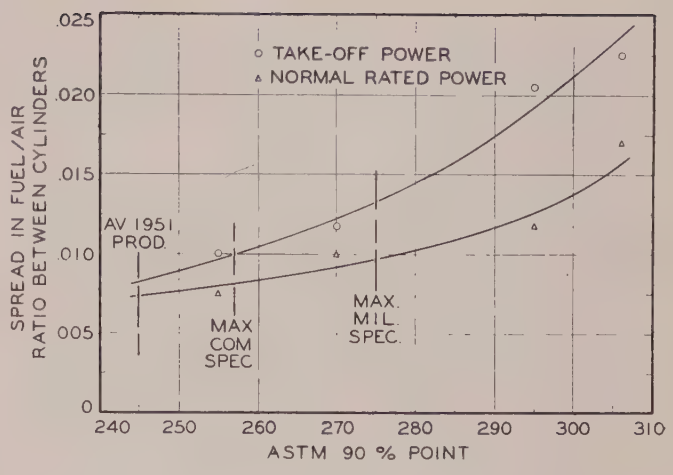


Fig. 15

appear that (a) the difference in performance between two fuels of the lowest volatility allowed by the commercial and military avgas is small, and that (b) the volatility characteristics of the average 1951 production military and civilian avgas are sufficiently better than the lowest volatility allowed by the specification that operation at maximum-economy cruise conditions should be feasible from a fuel quality standpoint. The suggestion that volatility characteristics of present production fuels would not restrict operation at maximum economy is made partly since it has been shown that the operating conditions that promote smooth combustion at ultralean mixtures are those that offer maximum economy. Needless to say, the operator and engine manufacturer would have to be guaranteed this improved volatility before they could plan on it.

Other Fuel Quality Requirements for Maximum Range

In addition to antiknock and volatility characteristics, perhaps the only other fuel factor that significantly influences the ability of an engine to operate at ultralean mixtures is the quantity of noncombustibles in the gasoline. The substitution

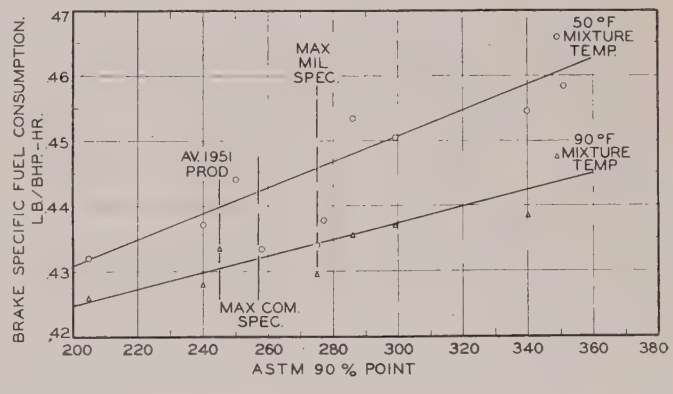


Fig. 16

of internal cooling by lowering the combustion gas temperature, in place of external cooling, can only be successful if the materials in the gasoline, which normally do not burn, will not deposit out on the spark plugs, combustion chamber, and valves. Unfortunately, at the lower combustion temperatures, inherent with ultralean fuel/air ratios, there is a greater tendency for lead oxybromide complexes to adhere to combustion-chamber surfaces as their temperatures approach and go below the melting and vaporization temperatures of the lead compounds. On the other hand, higher bmep operation and advanced spark settings increase the surface temperatures somewhat. In addition, considerable effort is being directed toward the development of improved scavenging agents.

Another fuel characteristic that is not directly related to successful engine operation at maximum-economy cruise conditions, but nevertheless influences aircraft range or payload, is the energy in the fuel. It is mentioned here because, curiously enough, the antiknock and volatility characteristics required for maximum-economy operating conditions are such that the fuel also has a high heat content per pound. This comes about because those paraffinic hydrocarbons offering high lean knock ratings and low 50 and 90% distillation temperatures are also those that offer higher heating value on a weight basis. For example, it has been shown[7] that the average 1951 production 100/130 and 115/145 are not only considerably better than the minimum specified F-3 rating and maximum specified 90% point but also have heat contents of about 19,060 Btu/lb as compared to the 18,700 and 18,800 required by the specification. However, unless manual leaning procedures are utilized, advantage cannot be taken of these heating values because at automatic lean carburetor settings fuel is metered to the engine on essentially a volume basis. Emphasis has been given to gravimetric heat content instead of volumetric heat content because the payload capacity of piston-engine transport aircraft is generally limited by weight restrictions rather than volume capacity.

Summary

An attempt has been made herein to outline the mutual effort necessary on the part of the aircraft operator, engine manufacturer, and refiner to assure satisfactory utilization of maximum-economy cruise operating conditions. While more emphasis has been given to fuel quality considerations in this paper because of the background of the authors, operational and engine design factors are at least equally, if not more, important.

Although it appears that manual lean operation is more critical of fuel quality, particularly antiknock and volatility characteristics, it also seems that significant improvement can be made in cruising economy with the average 1951 production fuels. This is possible because (1) partly through engine design, the present cruise operating points are much further than necessary from the knock-limited powers available with some specification fuels, and (2) because the average production aviation gasolines since the war have been considerably better than the specification requires.

There are some field experience and test data that indicate that the gain in cruising economy available by more severe operating conditions can be attained without jeopardizing engine reliability and at the same time improve the smoothness of engine operation if the average 1951 fuel quality characteristics can be guaranteed and if the operators exert utmost diligence in equipment maintenance and crew training. While the actual gain in economy from tighter cruise control for a given airline would depend upon the particular equipment utilized and the route structure, it would seem that for the aircraft of longer range an improvement of 5 to 7% could be effected over best-power, retarded-spark cruise settings. This would involve no significant engine modification.

DISCUSSION

U. S. Fuels Are Better Than International Ones

—J. T. Hendren
Pan American World Airways, Inc.

IN general we are in accord with the authors' concepts as to the proper way to attain greater fuel economy in cruising operation. For years we have utilized the high bmep, low rpm, manual leaning techniques, but have not explored spark advance any further than a 25-28-deg setting.

A quick survey of the properties of fuel now being delivered worldwide to our company was made, totaling 50 samples of grade 100/130 and 12 samples of grade 108/135, the latter from stations where heavy gas loads are taken on for transoceanic crossings. Although all of these fuels met the commercial specification (ASTM D 910) appreciable percentages were worse than the averages shown in the 1951 survey average (see footnote 7 of the paper). For grade 100/130 60% of the performance numbers were worse, 45% of the 50% evaporated points were worse, 57% of the 90% evaporated points were worse, and 53% of the sum of the 50 and 90% evaporated points were worse. For grade 108/135, 50% exceeded 108 performance number lean, 88% of the 50% evaporated points were worse than the 1951 survey average, 75% of the 90% evaporated points were worse, and 83% of the sum of the 50 and 90% evaporated points were worse.

We conclude from this that on current fuels in the international scene there is less margin on volatility and antiknock value than is shown in the 1951 fuel survey. Unless there is positive assurance that the margins exist in better than 90% of fuels being delivered to a given airline, full advantage cannot be taken of the optimum cruising regimes described. The only way to assure the margins is to rewrite the fuel specification. Desirable as this might be from an operator's standpoint, we are of the opinion that such a step would meet with something less than enthusiastic response on the part of the majority of aviation fuel suppliers.

Index to SAE Transactions

Volume 61—1953

A

Page

ACCIDENTS AND ACCIDENT PREVENTION (Nonaircraft)
Rolling Wheels Gather No Skids, A. C. Gunsaulus... 189
ADDITIVES, OIL
Can All Engine Wear Be Trapped in a Can? R. J. Pocock ... 154
Effect of Heavy-Duty Oils on Engine Wear in Typical Passenger-Car Service, J. L. Palmer... 237
Engineering Possibilities of Synthetic Lubricants, W. A. Zisman... 309
Laboratory and Field Wear Tests Using Radioactive Tracers, H. R. Jackson... 233
Lubricants – The Surface Savers, J. F. Kunc, Jr. and J. P. Hamer... 378
AIR CLEANERS
Can All Engine Wear Be Trapped in a Can? R. J. Pocock ... 154
AIRCRAFT ACCESSORIES AND EQUIPMENT
Development of High Temperature Hydraulic Fluids, J. C. Mosteller and John A. King... 668
Effect of Aerodynamic Heating on Fuel Systems, Julius Jonas ... 320
AIRCRAFT DESIGN AND PERFORMANCE
Can the Airlines Use the Helicopter? F. N. Piasecki and L. S. Wigdortchik... 97
Effect of Aerodynamic Heating on Fuel Systems, Julius Jonas ... 320
Today's Aluminum Aircraft Alloys, J. A. Nock, Jr... 209
AIRCRAFT ENGINES. See Engines, Aircraft
AIRCRAFT, ROTATING WING
Can the Airlines Use the Helicopter? F. N. Piasecki and L. S. Wigdortchik... 97
AIR TRANSPORT
Can the Airlines Use the Helicopter? F. N. Piasecki and L. S. Wigdortchik... 97
ALLEN, CLIFFORD H.
Discussion of papers on "Valve-Gear Problems in Modern Overhead-Valve Engines" by Philip Barkan, Ralph P. Horan, M. C. Turkish, and R. A. Roggenbuck ... 714
ALQUIST, H. E.
Aviation Fuel Economy and Quality – Brothers under the Aircraft Skin (Co-authors E. A. Droegemueller and H. N. Taylor) ... 717
ALUMINUM AND ALLOYS
Coating Steel by the Aldip Process, D. K. Hanink and A. L. Boegehold... 349
Safe Brakes for Passenger Cars, George T. Ladd and Sidney B. Dew... 623
Today's Aluminum Aircraft Alloys, J. A. Nock, Jr... 209
APPEL, W. D.
Discussion of "European Postwar Cars" by Maurice Olley ... 523
AUTOMOBILE ENGINES. See Engines, Passenger Car
AUTOMOBILES. See Passenger Cars

B

BACHMAN, B. B.
Growth and Status of Highway Transport... 114
BARKAN, PHILIP
Calculation of High-Speed Valve Motion With a Flexible Overhead Linkage... 687
BARNARD, D. P.
Discussion of "New Buick V-8 Engine Offers Many Advantages" by V. P. Mathews and J. D. Turlay.. 498
BATTERIES
Automobile Batteries – Selection, Service Life, New Developments, L. E. Wells... 253
BERLYN, MARTIN
Discussion of "Supercharged Diesel Performance versus Intake and Exhaust Conditions" by Lloyd Johnson ... 42
BLACKWOOD, A. J.
Discussion of "New Buick V-8 Engine Offers Many Advantages" by V. P. Mathews and J. D. Turlay.. 500

Page

BLOK, H.
Discussion of "A New Look at the Scoring Phenomena of Gears" by B. W. Kelley... 185
BODIES
Stress Testing Methods in Body Structures, H. C. Johnson ... 14
Wind Effects on Car Stability, W. E. Lay and P. W. Lett, Jr. ... 608
BOEGEHOLD, A. L.
Coating Steel by the Aldip Process (Co-author D. K. Hanink)... 349
BOGHOSSIAN, VICTOR
Discussion of "Wind Effects on Car Stability" by W. E. Lay and P. W. Lett, Jr. ... 619
BOKSENBOM, AARON S.
Automatic Control Systems Satisfying Certain General Criteria on Transient Behavior (Co-author Richard Hood)... 594
BOWDEN, B. G.
Discussion of "European Postwar Cars" by Maurice Olley ... 524
BOWDITCH, F. W.
Discussion of "A Method for Identifying Preignition" by R. F. Winch and F. M. Mayes... 461
BOYLAN, HENRY W.
Discussion of "New Buick V-8 Engine Offers Many Advantages" by V. P. Mathews and J. D. Turlay.. 502
BRAKES
Construction and Operation of Brake Testing Dynamometers ... 329
Rolling Wheels Gather No Skids, A. C. Gunsaulus.. 189
Safe Brakes for Passenger Cars, George T. Ladd and Sidney B. Dew... 623
Temperature Gradients and Heat Stresses in Brake Drums, G. A. G. Fazekas... 279
BUKER, C. B.
New Test for Drawability of Sheet Steel (Co-author J. R. Speer)... 469
BURMAN, PAUL G.
Discussion of "Measuring Rate of Fuel Injection in an Operating Diesel Engine" by R. J. Wehrman, H. R. Mitchell, and W. A. Turunen... 553
BURNS, R. S.
Discussion of "New Test for Drawability of Sheet Steel" by C. B. Buker and J. R. Speer... 476
BUS ENGINES. See Engines, Truck and Bus
BUS MAINTENANCE
Considerations for Commercial Valve Maintenance, J. A. Newton and N. Hoertz... 44

C

CARBURETORS AND CARBURETION
Engines to Digest the Vitamin-Enriched Fuel – Elpeegee, J. E. Glidewell... 131
CARIS, D. F.
Engine-Transmission Relationship for Higher Efficiency (Co-author R. A. Richardson)... 81
Discussion of "New Buick V-8 Engine Offers Many Advantages" by V. P. Mathews and J. D. Turlay.. 495
CHEN, SIMON K.
Discussion of "Engine Cylinder Pressure Measurements" by J. D. McCullough... 570
COLEMAN, WELLS
Discussion of "A New Look at the Scoring Phenomena of Gears" by B. W. Kelley... 185
COMBUSTION CHAMBERS
Combustion-Chamber Deposition and Knock, H. J. Gibson, C. A. Hall, and D. A. Hirschler... 361
New Buick V-8 Engine Offers Many Advantages, V. P. Mathews and J. D. Turlay... 478
Survey of Engine Combustion-Chamber Envelope, C. G. A. Rosen... 260
COMPRESSORS, TURBINE ENGINE
Investigation of Jet Engine Compressor Stall and Other Transient Characteristics, W. J. Kunz, Jr.... 55

Page

CONSTRUCTION AND INDUSTRIAL MACHINERY
Hydraulic Torque Converter – Its Effect on the Power Train, R. M. Schaefer and J. A. Winter 142
COOLING, ENGINE
Contemporary European Aircooled Diesel-Engine Practice, Wayne H. Worthington 422
CORROSION
Today's Aluminum Aircraft Alloys, J. A. Nock, Jr... 209
CORZILIUS, M. W.
Some Factors Affecting Precombustion Reactions in Engines (Co-authors D. R. Diggs and D. L. Pastell) 386
CYLINDERS
Effect of Heavy-Duty Oils on Engine Wear in Typical Passenger-Car Service, J. L. Palmer 237
Engine Wear – Comparison of Radioactive Wear and Field Measurements, F. W. Kavanagh 231
How Engines Wear, J. F. Kunc, Jr., D. S. McArthur, and L. E. Moody 221
Wear Prevention by Alkaline Lubricating Oils, J. C. Ellis and J. A. Edgar 244

D

DENISON, G. H.
Discussion of "Engineering Possibilities of Synthetic Lubricants" by W. A. Zisman 318
DEPOSITS, ENGINE
Combustion-Chamber Deposition and Knock, H. J. Gibson, C. A. Hall, and D. A. Hirschler 361
Effect of Heavy-Duty Oils on Engine Wear in Typical Passenger-Car Service, J. L. Palmer 237
Mode of Formation of Lead Deposits in Gasoline Engines, J. C. Street 442
Spark-Plug Fouling Studies, R. J. Greenshields..... 3
DETONATION
Aviation Fuel Economy and Quality – Brothers under the Aircraft Skin, H. E. Alquist, E. A. Droegemueller, and H. N. Taylor 717
Combustion-Chamber Deposition and Knock, H. J. Gibson, C. A. Hall, and D. A. Hirschler 361
Effect of Combustion Time on Knock in a Spark-Ignition Engine, D. R. Diggs 402
Mode of Formation of Lead Deposits in Gasoline Engines, J. C. Street 442
Some Factors Affecting Precombustion Reactions in Engines, M. W. Corzilius, D. R. Diggs, and D. L. Pastell 386
DEW, SIDNEY B.
Safe Brakes for Passenger Cars (Co-author George T. Ladd) 623
DICKSEE, C. B.
Discussion of "Supercharged Diesel Performance versus Intake and Exhaust Conditions" by Lloyd Johnson 42
DIESEL ENGINES. See Engines, Diesel
DIGGS, D. R.
Effect of Combustion Time on Knock in a Spark-Ignition Engine 402
Some Factors Affecting Precombustion Reactions in Engines (Co-authors M. W. Corzilius and D. L. Pastell)... 386
DROEGEMUELLER, E. A.
Aviation Fuel Economy and Quality – Brothers under the Aircraft Skin (Co-authors H. E. Alquist and H. N. Taylor)..... 717
DUDLEY, DARLE W.
Discussion of "A New Look at the Scoring Phenomena of Gears" by B. W. Kelley 186
DYNAMOMETERS
Construction and Operation of Brake Testing Dynamometers 329

E

EDGAR, J. A.
Wear Prevention by Alkaline Lubricating Oils (Co-author J. C. Ellis) 244
ELLIS, J. C.
Wear Prevention by Alkaline Lubricating Oils (Co-author J. A. Edgar) 244
ENGINES, AIRCRAFT (Jet and Turbine)
Afterburners and Variable Nozzles, Ralph Kress..... 165
Automatic Control Systems Satisfying Certain General Criteria on Transient Behavior, Aaron S. Boksenbom and Richard Hood 594

Page

Instrumentation for Aircraft Gas Turbine Development, R. E. Gorton and B. E. Miller 650
Investigation of Jet Engine Compressor Stall and Other Transient Characteristics, W. J. Kunz, Jr.... 55
ENGINES, AIRCRAFT (Reciprocating)
Aviation Fuel Economy and Quality – Brothers under the Aircraft Skin, H. E. Alquist, E. A. Droegemueller, and H. N. Taylor 717
Spark-Plug Fouling Studies, R. J. Greenshields..... 3
ENGINES, COMPOUND
What's Ahead in Commercial Vehicle Powerplants, C. G. A. Rosen 196
ENGINES, DIESEL
Contemporary European Aircooled Diesel-Engine Practice, Wayne H. Worthington 422
Laboratory and Field Wear Tests Using Radioactive Tracers, H. R. Jackson 233
Measuring Rate of Fuel Injection in an Operating Diesel Engine, R. J. Wehrman, H. R. Mitchell, and W. A. Turunen 542
Procedures Used in Development of Barnes & Reinecke Air Force Diesel Engine, E. W. Spannhake 574
Supercharged Diesel Performance versus Intake and Exhaust Conditions, Lloyd Johnson 34
Survey of Engine Combustion-Chamber Envelope, C. G. A. Rosen 260
Wear Prevention by Alkaline Lubricating Oils, J. C. Ellis and J. A. Edgar 244
What's Ahead in Commercial Vehicle Powerplants, C. G. A. Rosen 196
ENGINES, GENERAL
A Method for Identifying Preignition, R. F. Winch and F. M. Mayes 453
Critical Lubrication Areas in an Engine, Victor G. Raviolo 224
Effect of Heavy-Duty Oils on Engine Wear in Typical Passenger-Car Service, J. L. Palmer 237
Engine Cylinder Pressure Measurements, J. D. McCullough 557
Engine Wear – Comparison of Radioactive Wear and Field Measurements, F. W. Kavanagh 231
How Engines Wear, J. F. Kunc, Jr., D. S. McArthur, and L. E. Moody 221
Laboratory and Field Wear Tests Using Radioactive Tracers, H. R. Jackson 233
Some Factors Affecting Precombustion Reactions in Engines, M. W. Corzilius, D. R. Diggs, and D. L. Pastell 386
Wear Measurements by Physical, Chemical, or Radioactivity Methods, C. C. Moore and W. L. Kent..... 228
Wear Prevention by Alkaline Lubricating Oils, J. C. Ellis and J. A. Edgar 244
ENGINES, JET AND TURBINE (Nonaircraft)
What's Ahead in Commercial Vehicle Powerplants, C. G. A. Rosen 196
ENGINES, PASSENGER CAR
A Method for Identifying Preignition, R. F. Winch and F. M. Mayes 453
Can All Engine Wear Be Trapped in a Can? R. J. Pocock 154
Effect of Heavy-Duty Oils on Engine Wear in Typical Passenger-Car Service, J. L. Palmer 237
Engine-Transmission Relationship for Higher Efficiency, D. F. Caris and R. A. Richardson 81
Engine Wear – Comparison of Radioactive Wear and Field Measurements, F. W. Kavanagh 231
European Postwar Cars, Maurice Olley 503
How Engines Wear, J. F. Kunc, Jr., D. S. McArthur, and L. E. Moody 221
Laboratory and Field Wear Tests Using Radioactive Tracers, H. R. Jackson 233
New Buick V-8 Engine Offers Many Advantages, V. P. Mathews and J. D. Turlay 478
Spark-Plug Fouling Studies, R. J. Greenshields..... 3
Wear Prevention by Alkaline Lubricating Oils, J. C. Ellis and J. A. Edgar 244
ENGINES, TRACTOR
Contemporary European Aircooled Diesel-Engine Practice, Wayne H. Worthington 422
ENGINES, TRUCK AND BUS
Contemporary European Aircooled Diesel-Engine Practice, Wayne H. Worthington 422

Page

ENGINES, TRUCK AND BUS (Continued)
Growth and Status of Highway Transport, B. B. Bachman 114
What's Ahead in Commercial Vehicle Powerplants, C. G. A. Rosen 196

F

FAZEKAS, G. A. G.
Temperature Gradients and Heat Stresses in Brake Drums 279
FINISHES
Coating Steel by the Aldip Process, D. K. Hanink and A. L. Boegehold 349
FLEET OPERATION
Fast Replacement or Heavy Maintenance, Charles C. Hudson 529
Fleet Management: A Job Evaluation, T. L. Preble .. 273
FLEISCHEL, GASTON
Discussion of "European Postwar Cars" by Maurice Olley 523
FOREIGN DESIGN AND OPERATION
Contemporary European Aircooled Diesel-Engine Practice, Wayne H. Worthington 422
European Postwar Cars, Maurice Olley 503
FUELS, AVIATION
Aviation Fuel Economy and Quality – Brothers under the Aircraft Skin, H. E. Alquist, E. A. Droegemueller, and H. N. Taylor 717
Mode of Formation of Lead Deposits in Gasoline Engines, J. C. Street 442
Spark-Plug Fouling Studies, R. J. Greenshields 3
FUELS, MOTOR
Can All Engine Wear Be Trapped in a Can? R. J. Pocock 154
Combustion-Chamber Deposition and Knock, H. J. Gibson, C. A. Hall, and D. A. Hirschler 361
Effect of Combustion Time on Knock in a Spark-Ignition Engine, D. R. Diggs 402
Engines to Digest the Vitamin-Enriched Fuel – Elpeegee, J. E. Glidewell 131
Mode of Formation of Lead Deposits in Gasoline Engines, J. C. Street 442
Some Factors Affecting Precombustion Reactions in Engines, M. W. Corzilius, D. R. Diggs, and D. L. Pastell 386
Spark-Plug Fouling Studies, R. J. Greenshields 3

G

GAS TURBINES. See Engines, Aircraft (Jet and Turbine); Engines, Jet and Turbine (Nonaircraft)
GEARS
A New Look at the Scoring Phenomena of Gears, B. W. Kelley 175
GIBSON, H. J.
Combustion-Chamber Deposition and Knock (Co-authors C. A. Hall and D. A. Hirschler) 361
GIBSON, WILTON B.
Hydraulic Torque Converter – Common-Law Spouse of the Heavy-Duty Truck 409
GLIDEWELL, J. E.
Engines to Digest the Vitamin-Enriched Fuel – Elpeegee 131
GLYNN, F. K.
Discussion of "Automobile Batteries – Selection, Service Life, New Developments" by L. E. Wells 259
Discussion of "Fleet Management: A Job Evaluation" by T. L. Preble 278
GOFFE, J. H.
Discussion of "A Method for Identifying Preignition" by R. F. Winch and F. M. Mayes 465
GORTON, R. E.
Instrumentation for Aircraft Gas Turbine Development (Co-author B. E. Miller) 650
GREENSHIELDS, R. J.
Spark-Plug Fouling Studies 3
GUIDED MISSILES
Automatic Control Systems Satisfying Certain General Criteria on Transient Behavior, Aaron S. Boksenbom and Richard Hood 594
GUNSAULUS, A. C.
Rolling Wheels Gather No Skids 189

H

Page

HALL, C. A.
Combustion-Chamber Deposition and Knock (Co-authors H. J. Gibson and D. A. Hirschler) 361
HAMER, J. P.
Lubricants – The Surface Savers (Co-author J. F. Kunc, Jr.) 378
HANINK, D. K.
Coating Steel by the Aldip Process (Co-author A. L. Boegehold) 349
HANSEN, M.
Titanium Alloy Development (Co-author H. D. Kessler) 640
HEINE, L. J.
Discussion of "Automobile Batteries – Selection, Service Life, New Developments" by L. E. Wells 258
HELICOPTERS. See Aircraft, Rotating Wing
HENDREN, J. T.
Discussion of "Aviation Fuel Economy and Quality – Brothers under the Aircraft Skin" by H. E. Alquist, E. A. Droegemueller, and H. N. Taylor 726
HENNING, WILLIAM W.
Steering of Track-Type Vehicles 25
HERNDON, W. B.
Discussion of "Engine-Transmission Relationships for Higher Efficiency" by D. F. Caris and R. A. Richardson 94
HERON, S. D.
Discussion of "New Buick V-8 Engine Offers Many Advantages" by V. P. Mathews and J. D. Turlay .. 499
HIRSCHLER, D. A.
Combustion-Chamber Deposition and Knock (Co-authors H. J. Gibson and C. A. Hall) 361
HOERTZ, N.
Considerations for Commercial Valve Maintenance (Co-author J. A. Newton) 44
HOOD, RICHARD
Automatic Control Systems Satisfying Certain General Criteria on Transient Behavior (Co-author Aaron S. Boksenbom) 594
HORAN, RALPH P.
Overhead Valve Gear Problems 678
HUDSON, CHARLES C.
Fast Replacement or Heavy Maintenance 529
HUSSMANN, A. W.
Discussion of papers on "Valve-Gear Problems in Modern Overhead-Valve Engines" by Philip Barkan, Ralph P. Horan, M. C. Turkish, and R. A. Roggenbuck 715
HYDRAULIC FLUIDS
Development of High Temperature Hydraulic Fluids, J. C. Mosteller and John A. King 668

I

IGNITION SYSTEMS
Spark-Plug Fouling Studies, R. J. Greenshields 3
INJECTION, FUEL
Engines to Digest the Vitamin-Enriched Fuel – Elpeegee, J. E. Glidewell 131
Measuring Rate of Fuel Injection in an Operating Diesel Engine, R. J. Wehrman, H. R. Mitchell, and W. A. Turunen 542
INSTRUMENTS
Construction and Operation of Brake Testing Dynamometers 329
Engine Cylinder Pressure Measurements, J. D. McCullough 557
Instrumentation for Aircraft Gas Turbine Development, R. E. Gorton and B. E. Miller 650
Investigation of Jet Engine Compressor Stall and Other Transient Characteristics, W. J. Kunz, Jr. .. 55
New Test for Drawability of Sheet Steel, C. B. Buker and J. R. Speer 469
IRON AND STEEL
Coating Steel by the Aldip Process, D. K. Hanink and A. L. Boegehold 349
New Test for Drawability of Sheet Steel, C. B. Buker and J. R. Speer 469
Safe Brakes for Passenger Cars, George T. Ladd and Sidney B. Dew 623

J

JACKSON, H. R.
Laboratory and Field Wear Tests Using Radioactive Tracers 233

Page

JANDASEK, V. J.
Discussion of "Engine-Transmission Relationships for Higher Efficiency" by D. F. Caris and R. A. Richardson 95
JET PROPULSION. See Engines, Aircraft (Jet and Turbine)
JOHNSON, H. C.
Stress Testing Methods in Body Structures..... 14
JOHNSON, LLOYD
Supercharged Diesel Performance versus Intake and Exhaust Conditions 34
JONAS, JULIUS
Effect of Aerodynamic Heating on Fuel Systems.... 320
JONES, B. F.
Discussion of "Automobile Batteries – Selection, Service Life, New Developments" by L. E. Wells..... 259

K

KAMM, W. I. E.
Discussion of "Wind Effects on Car Stability" by W. E. Lay and P. W. Lett, Jr. 620
KAVANAGH, F. W.
Engine Wear – Comparison of Radioactive Wear and Field Measurements 231
KELLEY, B. W.
A New Look at the Scoring Phenomena of Gears.... 175
KENDALL, T. L.
Discussion of "Automobile Batteries – Selection, Service Life, New Developments" by L. E. Wells..... 259
KENT, W. L.
Wear Measurements by Physical, Chemical, or Radioactivity Methods (Co-author C. C. Moore)..... 228
KESSLER, H. D.
Titanium Alloy Development (Co-author M. Hansen)..... 640
KING, JOHN A.
Development of High Temperature Hydraulic Fluids (Co-author J. C. Mosteller)..... 668
KISSAM, W. M.
Discussion of "Fast Replacement or Heavy Maintenance" by Charles C. Hudson..... 539
KLOTH, J. A.
Discussion of "European Postwar Cars" by Maurice Olley 522
KOZIER, T. J.
Discussion of "Wind Effects on Car Stability" by W. E. Lay and P. W. Lett, Jr. 622
KRANC, STANLEY J.
Discussion of "Measuring Rate of Fuel Injection in an Operating Diesel Engine" by R. J. Wehrman, H. R. Mitchell, and W. A. Turunen..... 556
KRESS, RALPH
Afterburners and Variable Nozzles..... 165
KRUGER, FREDERICK B.
Discussion of "Engine Cylinder Pressure Measurements" by J. D. McCullough..... 570
KUNC, J. F., JR.
How Engines Wear (Co-authors D. S. McArthur and L. E. Moody).... 221
Lubricants – The Surface Savers (Co-author J. P. Hamer)..... 378
KUNZ, W. J., JR.
Investigation of Jet Engine Compressor Stall and Other Transient Characteristics..... 55

L

LADD, GEORGE T.
Safe Brakes for Passenger Cars (Co-author Sidney B. Dew)..... 623
LAY, W. E.
Wind Effects on Car Stability (Co-author P. W. Lett, Jr.)..... 608
LETT, P. W., JR.
Wind Effects on Car Stability (Co-author W. E. Lay)..... 608
LIVENGOOD, J. C.
Discussion of "Some Factors Affecting Precombustion Reactions in Engines" by M. W. Corzilius, D. R. Diggs, and D. L. Pastell..... 400
LIVINGSTONE, C. J.
Discussion of "New Buick V-8 Engine Offers Many Advantages" by V. P. Mathews and J. D. Turlay... 501

Page

LUBRICANTS AND LUBRICATION (Aircraft)
Development of High Temperature Hydraulic Fluids, J. C. Mosteller and John A. King..... 668
Engineering Possibilities of Synthetic Lubricants, W. A. Zisman..... 309
Lubricants – The Surface Savers, J. F. Kunc, Jr. and J. P. Hamer..... 378
LUBRICANTS AND LUBRICATION (Nonaircraft)
A New Look at the Scoring Phenomena of Gears, B. W. Kelley..... 175
Can All Engine Wear Be Trapped in a Can? R. J. Pocock 154
Combustion-Chamber Deposition and Knock, H. J. Gibson, C. A. Hall, and D. A. Hirschler..... 361
Critical Lubrication Areas in an Engine, Victor G. Raviolo 224
Effect of Heavy-Duty Oils on Engine Wear in Typical Passenger-Car Service, J. L. Palmer..... 237
Engine Wear – Comparison of Radioactive Wear and Field Measurements, F. W. Kavanagh..... 231
Laboratory and Field Wear Tests Using Radioactive Tracers, H. R. Jackson..... 233
Lubricants – The Surface Savers, J. F. Kunc, Jr. and J. P. Hamer..... 378
Wear Prevention by Alkaline Lubricating Oils, J. C. Ellis and J. A. Edgar..... 244
LUPTON, C. R.
Discussion of "Safe Brakes for Passenger Cars" by George T. Ladd and Sidney B. Dew..... 638

M

MacGREGOR, J. R.
Discussion of "New Buick V-8 Engine Offers Many Advantages" by V. P. Mathews and J. D. Turlay... 498
MANAGEMENT
Fleet Management: A Job Evaluation, T. L. Preble.. 273
MARGOLIN, ELIAS
Discussion of "Engineering Possibilities of Synthetic Lubricants" by W. A. Zisman..... 318
MARTIN, J. J.
Discussion of "Effect of Aerodynamic Heating on Fuel Systems" by Julius Jonas..... 327
MATHEWS, V. P.
New Buick V-8 Engine Offers Many Advantages (Co-author J. D. Turlay)..... 478
MAYES, F. M.
A Method for Identifying Preignition (Co-author R. F. Winch)..... 453
McARTHUR, D. S.
How Engines Wear (Co-authors J. F. Kunc, Jr., and L. E. Moody).... 221
McCULLOUGH, J. D.
Engine Cylinder Pressure Measurements..... 557
McFARLAND, FOREST
Discussion of "Engine-Transmission Relationships for Higher Efficiency" by D. F. Caris and R. A. Richardson 95
McREYNOLDS, L. A.
Discussion of "Some Factors Affecting Precombustion Reactions in Engines" by M. W. Corzilius, D. R. Diggs, and D. L. Pastell..... 398
McWILLIAMS, R. L.
Discussion of "Engine-Transmission Relationships for Higher Efficiency" by D. F. Caris and R. A. Richardson 96
MEAGHER, R.
Discussion of "A Method for Identifying Preignition" by R. F. Winch and F. M. Mayes..... 466
METALS
Titanium Alloy Development, M. Hansen and H. D. Kessler 640
MILITARY VEHICLES
Steering of Track-Type Vehicles, William W. Henning 25
MILLER, B. E.
Instrumentation for Aircraft Gas Turbine Development (Co-author R. E. Gorton)..... 650
MILLER, C. D.
Discussion of "Some Factors Affecting Precombustion Reactions in Engines" by M. W. Corzilius, D. R. Diggs, and D. L. Pastell..... 399
MILLER, E. F.
Discussion of "New Buick V-8 Engine Offers Many Advantages" by V. P. Mathews and J. D. Turlay... 501

Page

MITCHELL, H. R.
Measuring Rate of Fuel Injection in an Operating Diesel Engine (Co-authors R. J. Wehrman and W. A. Turunen).. 542
MODELS
Plastic Prototypes Revolutionize Preparation for Manufacture, Wallace A. Stanley..... 67
MOODY, L. E.
How Engines Wear (Co-authors J. F. Kunc, Jr., and D. S. McArthur).. 221
MOORE, C. C.
Wear Measurements by Physical, Chemical, or Radioactivity Methods (Co-author W. L. Kent)..... 228
MOSTELLER, J. C.
Development of High Temperature Hydraulic Fluids (Co-author John A. King)..... 668
MOXEY, J. G., JR.
Discussion of "Combustion-Chamber Deposition and Knock" by H. J. Gibson, C. A. Hall, and D. A. Hirschler 376
MYERS, P. S.
Discussion of "Engine Cylinder Pressure Measurements" by J. D. McCullough..... 568

N

NEWTON, J. A.
Considerations for Commercial Valve Maintenance (Co-author N. Hoertz)..... 44
NOCK, J. A., JR.
Today's Aluminum Aircraft Alloys..... 209

O

OBERDORFER, P. E.
Discussion of "Some Factors Affecting Precombustion Reactions in Engines" by M. W. Corzilius, D. R. Diggs, and D. L. Pastell..... 398
OGANOWSKI, K.
Discussion of "Coating Steel by the Aldip Process" by D. K. Hanink and A. L. Boegehold..... 360
OGDEN, E. B.
Discussion of "Fast Replacement or Heavy Maintenance" by Charles C. Hudson..... 540
OIL FILTERS
Can All Engine Wear Be Trapped in a Can? R. J. Pocock 154
OLLEY, MAURICE
European Postwar Cars..... 503
OLT, T. F.
Discussion of "Coating Steel by the Aldip Process" by D. K. Hanink and A. L. Boegehold..... 360
OVERCASH, R. L.
Discussion of "Combustion-Chamber Deposition and Knock" by H. J. Gibson, C. A. Hall, and D. A. Hirschler 377

P

PALMER, J. L.
Effect of Heavy-Duty Oils on Engine Wear in Typical Passenger-Car Service..... 237
PASSENGER CAR DESIGN AND PERFORMANCE
Automobile Batteries – Selection, Service Life, New Developments, L. E. Wells..... 253
European Postwar Cars, Maurice Olley..... 503
Plastic Prototypes Revolutionize Preparation for Manufacture, Wallace A. Stanley..... 67
Safe Brakes for Passenger Cars, George T. Ladd and Sidney B. Dew..... 623
Stress Testing Methods in Body Structures, H. C. Johnson 14
Wind Effects on Car Stability, W. E. Lay and P. W. Lett, Jr. 608
PASSENGER CAR ENGINES. See Engines, Passenger Car
PASSENGER CAR MAINTENANCE
Considerations for Commercial Valve Maintenance, J. A. Newton and N. Hoertz..... 44
PASTELL, D. L.
Some Factors Affecting Precombustion Reactions in Engines (Co-authors M. W. Corzilius and D. R. Diggs).... 386
PIASECKI, F. N.
Can the Airlines Use the Helicopter? (Co-author L. S. Wigdortchik)..... 97

Page

PISTONS AND RINGS
Effect of Heavy-Duty Oils on Engine Wear in Typical Passenger-Car Service, J. L. Palmer..... 237
Engine Wear – Comparison of Radioactive Wear and Field Measurements, F. W. Kavanagh..... 231
How Engines Wear, J. F. Kunc, Jr., D. S. McArthur, and L. E. Moody..... 221
Laboratory and Field Wear Tests Using Radioactive Tracers, H. R. Jackson..... 233
Wear Measurements by Physical, Chemical, or Radioactivity Methods, C. C. Moore and W. L. Kent..... 228
Wear Prevention by Alkaline Lubricating Oils, J. C. Ellis and J. A. Edgar..... 244
PLASTICS
Plastic Prototypes Revolutionize Preparation for Manufacture, Wallace A. Stanley..... 67
POCOCK, R. J.
Can All Engine Wear Be Trapped in a Can?..... 154
PREBLE, T. L.
Discussion of "Fast Replacement or Heavy Maintenance" by Charles C. Hudson..... 539
Fleet Management: A Job Evaluation..... 273
PRETZ, P. H.
Discussion of "European Postwar Cars" by Maurice Olley 525
PRODUCTION
New Test for Drawability of Sheet Steel, C. B. Buker and J. R. Speer..... 469
Plastic Prototypes Revolutionize Preparation for Manufacture, Wallace A. Stanley..... 67

R

RADIOACTIVITY
Engine Wear – Comparison of Radioactive Wear and Field Measurements, F. W. Kavanagh..... 231
Laboratory and Field Wear Tests Using Radioactive Tracers, H. R. Jackson..... 233
Wear Measurements by Physical, Chemical, or Radioactivity Methods, C. C. Moore and W. L. Kent..... 228
Wear Prevention by Alkaline Lubricating Oils, J. C. Ellis and J. A. Edgar..... 244
RASSWEILER, G. M.
Discussion of "Engine Cylinder Pressure Measurements" by J. D. McCullough..... 573
RAVIOLO, VICTOR G.
Critical Lubrication Areas in an Engine..... 224
RICHARDSON, R. A.
Engine-Transmission Relationship for Higher Efficiency (Co-author D. F. Caris)..... 81
RIFKIN, ELLIS B.
Discussion of "Mode of Formation of Lead Deposits in Gasoline Engines" by J. C. Street..... 451
ROADS AND STREETS
Growth and Status of Highway Transport, B. B. Bachman 114
ROGGENBUCK, R. A.
Designing the Cam Profile for Low Vibration at High Speeds 701
ROSEN, C. G. A.
Survey of Engine Combustion-Chamber Envelope... 260
What's Ahead in Commercial Vehicle Powerplants.. 196
ROUND, GEORGE
Discussion of "Laboratory and Field Wear Tests Using Radioactive Tracers" by H. R. Jackson..... 251

S

SAMPIETRO, A. C.
Discussion of "European Postwar Cars" by Maurice Olley 524
Discussion of "New Buick V-8 Engine Offers Many Advantages" by V. P. Mathews and J. D. Turlay... 502
SAWYER, R. T.
Discussion of "Supercharged Diesel Performance versus Intake and Exhaust Conditions" by Lloyd Johnson 42
SCHAEFER, R. M.
Hydraulic Torque Converter – Its Effect on the Power Train (Co-author J. A. Winter)..... 142
SCHERGER, T. A.
Discussion of "New Buick V-8 Engine Offers Many Advantages" by V. P. Mathews and J. D. Turlay... 500

Page

SHADE, WILLIAM S.
Discussion of "Measuring Rate of Fuel Injection in an Operating Diesel Engine" by R. J. Wehrman, H. R. Mitchell, and W. A. Turunen..... 556
SPANNHAKE, E. W.
Procedures Used in Development of Barnes & Reinecke Air Force Diesel Engine..... 574
SPARK PLUGS
Mode of Formation of Lead Deposits in Gasoline Engines, J. C. Street..... 442
Spark-Plug Fouling Studies, R. J. Greenshields..... 3
SPEER, J. R.
New Test for Drawability of Sheet Steel (Co-author C. B. Buker)..... 469
STANDARDIZATION
Construction and Operation of Brake Testing Dynamometers 329
STANLEY, WALLACE A.
Plastic Prototypes Revolutionize Preparation for Manufacture 67
STEEL. See Iron and Steel
STEERING
Steering of Track-Type Vehicles, William W. Henning 25
STEIGERWALT, H. G.
Discussion of "Automobile Batteries – Selection, Service Life, New Developments" by L. E. Wells..... 259
STEIN, J. A.
Discussion of "Engine Cylinder Pressure Measurements" by J. D. McCullough..... 572
STEWART, J. P.
Discussion of "How Engines Wear" by J. F. Kunc, Jr., D. S. McArthur, and L. E. Moody..... 251
STREET, J. C.
Mode of Formation of Lead Deposits in Gasoline Engines 442
STRESSES
Hydraulic Torque Converter – Its Effect on the Power Train, R. M. Schaefer and J. A. Winter..... 142
Stress Testing Methods in Body Structures, H. C. Johnson 14
Temperature Gradients and Heat Stresses in Brake Drums, G. A. G. Fazekas..... 279
STURGIS, B. M.
Discussion of "Mode of Formation of Lead Deposits in Gasoline Engines" by J. C. Street..... 450
SUPERCHARGING
Supercharged Diesel Performance versus Intake and Exhaust Conditions, Lloyd Johnson..... 34
Survey of Engine Combustion-Chamber Envelope, C. G. A. Rosen..... 260
What's Ahead in Commercial Vehicle Powerplants, C. G. A. Rosen..... 196

T

TAYLOR, H. N.
Aviation Fuel Economy and Quality – Brothers under the Aircraft Skin (Co-authors H. E. Alquist and E. A. Droegemueller) 717
TIRES AND RIMS
Growth and Status of Highway Transport, B. B. Bachman 114
TOULMIN, H. A.
Discussion of "European Postwar Cars" by Maurice Olley 521
TRACKLAYING VEHICLES
Steering of Track-Type Vehicles, William W. Henning 25
TRACTOR ENGINES. See Engines, Tractor
TRACTORS
Steering of Track-Type Vehicles, William W. Henning 25
TRANSMISSIONS, AUTOMATIC
Engine-Transmission Relationship for Higher Efficiency, D. F. Caris and R. A. Richardson..... 81
Hydraulic Torque Converter – Common-Law Spouse of the Heavy-Duty Truck, Wilton B. Gibson..... 409
Hydraulic Torque Converter – Its Effect on the Power Train, R. M. Schaefer and J. A. Winter.... 142
TRANSMISSIONS, HAND CONTROLLED
Engine-Transmission Relationships for Higher Efficiency, D. F. Caris and R. A. Richardson..... 81
TRUCK DESIGN AND PERFORMANCE
Growth and Status of Highway Transport, B. B. Bachman 114
Hydraulic Torque Converter – Common-Law Spouse of the Heavy-Duty Truck, Wilton B. Gibson..... 409

Page

Hydraulic Torque Converter – Its Effect on the Power Train, R. M. Schaefer and J. A. Winter.... 142
Rolling Wheels Gather No Skids, A. C. Gunsaulus... 189
TRUCK ENGINES. See Engines, Truck and Bus
TRUCK MAINTENANCE
Considerations for Commercial Valve Maintenance, J. A. Newton and N. Hoertz..... 44
Fast Replacement or Heavy Maintenance, Charles C. Hudson 529
Fleet Management: A Job Evaluation, T. L. Preble.. 273
TRUCK TRACTORS AND TRAILERS
Rolling Wheels Gather No Skids, A. C. Gunsaulus... 189
TURKISH, M. C.
Relationship of Valve-Spring Design to Valve Gear Dynamics and Hydraulic Lifter Pump-up..... 706
TURLAY, J. D.
New Buick V-8 Engine Offers Many Advantages (Co-author V. P. Mathews)..... 478
Discussion of papers on "Valve-Gear Problems in Modern Overhead-Valve Engines" by Philip Barkan, Ralph P. Horan, M. C. Turkish, and R. A. Roggenbuck 715
TURUNEN, W. A.
Measuring Rate of Fuel Injection in an Operating Diesel Engine (Co-authors R. J. Wehrman and H. R. Mitchell)... 542

U

UDALE, S. M.
Discussion of "Spark-Plug Fouling Studies" by R. J. Greenshields 13
UYEHARA, O. A.
Discussion of "Engine Cylinder Pressure Measurements" by J. D. McCullough..... 568

V

VALVES AND GEAR
Calculation of High-Speed Valve Motion with a Flexible Overhead Linkage, Philip Barkan..... 687
Considerations for Commercial Valve Maintenance, J. A. Newton and N. Hoertz..... 44
Designing the Cam Profile for Low Vibration at High Speeds, R. A. Roggenbuck..... 701
Overhead Valve Gear Problems, Ralph P. Horan.... 678
Relationship of Valve-Spring Design to Valve Gear Dynamics and Hydraulic Lifter Pump-up, M. C. Turkish 706

W

WEHRMAN, R. J.
Measuring Rate of Fuel Injection in an Operating Diesel Engine (Co-authors H. R. Mitchell and W. A. Turunen)... 542
WELDING
Today's Aluminum Aircraft Alloys, J. A. Nock, Jr... 209
WELLS, L. E.
Automobile Batteries – Selection, Service Life, New Developments 253
WHEELER, J. W.
Discussion of "A Method for Identifying Preignition" by R. F. Winch and F. M. Mayes..... 463
WHITFIELD, M. G.
Discussion of "Coating Steel by the Aldip Process" by D. K. Hanink and A. L. Boegehold..... 359
WIGDORTCHIK, L. S.
Can the Airlines Use the Helicopter? (Co-author F. N. Piasecki)..... 97
WILSON, D. K.
Discussion of "Fast Replacement or Heavy Maintenance" by Charles C. Hudson..... 539
WINCH, R. F.
A Method for Identifying Preignition (Co-author F. M. Mayes)..... 453
WINTER, J. A.
Hydraulic Torque Converter – Its Effect on the Power Train (Co-author R. M. Schaefer)..... 142
WORTHINGTON, WAYNE H.
Contemporary European Aircooled Diesel-Engine Practice 422
WYCZALEK, F. A.
Discussion of "New Buick V-8 Engine Offers Many Advantages" by V. P. Mathews and J. D. Turlay... 495

Z

ZISMAN, W. A.
Engineering Possibilities of Synthetic Lubricants.... 309

DATE DUE

UIC APR 16 '85

MAY 31 REC'D

GAYLORD PRINTE